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Metal-based microchannel heat exchangers: manufacturing and heat transfer testing

Bin Lu
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METAL-BASED MICROCHANNEL HEAT EXCHANGERS:
MANUFACTURING AND HEAT TRANSFER TESTING

A Dissertation

Submitted to the Graduate Faculty of the
Louisiana State University and
Agricultural and Mechanical College
in partial fulfillment of the
requirements for the degree of
Doctor of Philosophy

in

The Department of Mechanical Engineering

by

Bin Lu
B.S., Zhejiang University, 2007
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VITA
ABSTRACT

This dissertation focuses on improving the functionality of metal-based microchannel heat exchangers (MHEs), as well as pushing this technology toward real-world applications. Design optimization was carried out on MHEs for performance maximization. Double-layered microchannel layout was experimentally studied, and a significant reduction on liquid flow pressure drop penalty was achieved. Other than water, another commonly-used coolant, ethylene glycol, was applied to MHEs, and flow and heat transfer characteristics were quantified. Transient Liquid Phase (TLP) bonding was used for joining Cu structures. For further understanding of the MHE heat transfer, a detailed examination was carried out on the TLP bonding interface region. In real applications, an MHE is likely to work with a heat rejection device. Therefore, further study was done on MHEs in the context of a close-loop recirculating-liquid cooling system. An alternative roll molding method suitable for continuous fabrication of metal-based microchannel arrays was studied. This technology may serve as an enabler for large-scale manufacturing of metal-based microchannel devices in an economical fashion.
CHAPTER 1. AN INTRODUCTION ON APPLICATION AND FABRICATION OF MICROCHANNEL HEAT EXCHANGERS

This chapter will lay out the background for this dissertation in terms of potential applications of MHEs and current techniques of microchannel fabrication.

1.1 Introduction

A microchannel heat exchanger (MHE) in general is a solid device with enclosed microchannels embedded within it. Cool (or hot) fluids flow through the microchannels and remove heat from (or add heat into) the solid body of the MHE. Heat transfer between the working fluid and the MHE solid body is achieved through convection occurring between the working fluid and the internal surfaces of microchannels. The working fluids for microchannel devices are more often liquids rather than gases, as convective heat transfer between liquids and solids occurs at a much higher rate than that between gases and solids.

The rate of liquid-solid convective heat transfer can be further enhanced by reducing channel sizes, as demonstrated by Tuckerman and Pease in 1981 [1]. This is the very rationale behind the adoption of MHEs for higher heat transfer performance. In the work of Tuckerman and Pease, several Si-based, high-performance, heat sinks were fabricated by etching microchannels onto Si substrates using KOH. These Si-based MHEs demonstrated a thermal resistance of ~0.1°C/W for a 1cm² area, or removal of heat flux up to 790W/cm², using water as the working fluid. Since the pioneering work of Tuckerman and Pease, liquid flow and heat transfer occurring within MHEs have been intensely studied [2, 3], mainly focusing on Si-based devices [4, 5, 6]. That is because fabrication technologies for Si-based high-aspect-ratio structures are most mature and
most widely available. However, metal-based MHEs possess several key advantages, including improved heat transfer performance due to higher bulk thermal conductivities and increased mechanical robustness and consequently the potential for building MHE devices with low profiles and in different configurations. Heat transfer of Cu-based MHE has been studied by Lee, Garimella and Liu [7]. Techniques for fabrication of Cu- and Al-based MHEs have been studied by Mei et al. [8].

1.2 Applications of MHEs

1.2.1 Cooling of Electronic Modules

![Figure 1-1. A history of electronic chip heat generation in log-log plot. Additional data points were added to those of Chu et al. [9].](image)

According to the well-known Moore’s law [10], performance of microelectronic modules or chips doubles roughly every two years. Accordingly, chip level heat generation also increases exponentially with time. As shown in Fig. 1-1 [9], this has been the case in the last 5 to 6 decades. The heat generated by microelectronic modules needs to be removed in order to keep their operating temperature under its maximum allowed value. As shown in Fig. 1-1, current personal computer central processing units (CPUs)
are capable of generating heat as high as 80W/cm² [11]. Cooling such CPUs pushes the limit of the conventional heat management solution—forced air convection using solid heat sinks and fans. Microchannel heat exchangers provide a potential solution to this challenge, as they are capable of removing highly concentrated heat within a confined area. As compared to Si-based MHEs pioneered by Tuckerman and Pease [1], Al- and Cu- based MHEs can provide an extra boost in heat transfer performance since bulk thermal conductivities of Cu (401W/m-K) and Al (237W/m-K) are significantly higher than that of Si (148W/m-K) [12].

Besides cooling of personal computer CPUs, MHEs are good candidates for data center heat management as well. Data center is a centralized facility for storage, processing, and management of data and information. As the current trends of cloud computing and social networking continue, the demand for data center capacity and performance increases rapidly. Together with this increasing demand, the density of heat generation of data centers also increases rapidly. As thousands of microelectronic processors stack up in a single server rack, air-convection-based heat management solution faces a great challenge [13]. An IBM blade server rack architecture consumes as much as 30kW within a less than 1m² footprint [14]. Average power density of data centers are expected to reach as high as 3000W/m² in the next a few years [15]. Current heat management solution of data center uses air conditioning system which runs around the clock on a 24/7 basis. Studies have been primarily focused on air flow pattern in the accommodating room to better dissipate the heat generated by thousands of processors [13, 16].
Using air conditioning system for the cooling of data centers imposes a double-impact on the environment as well as a double-spending on energy bill, as a significant amount of energy is used by air conditioners to offset the heat generated by those processors. Furthermore, due to the low density and low heat capacity of air, the density of a data center is currently limited by the capacity of air conditioners. An MHE-based, closed-loop, liquid-based cooling system could well overcome these two shortcomings of the current air conditioning system. Such a liquid cooling system would consist of a pump, a large radiator sitting outside of a building, MHEs attached to each processor, and pipelines that connect components together. Such a re-circulating system will be demonstrated in chapter 6. By placing the radiator outside of the accommodating room, heat is directly dissipated into the atmosphere, and thus no need for using extra energy to cool it down. Such a liquid cooling system also allows for a more condensed layout of data center, since liquid typically possesses a much higher density and a higher heat capacity than those of air. For example, water is 857 times denser and 3 times higher in specific heat at 300K as compared to air [12].

In summary, such MHE-based, closed-loop, liquid-based cooling systems could reduce the impact on the environment and help cut electricity budget for operating a data center, meanwhile dramatically improve the efficiency of floor usage in a data center.

1.2.2 Cooling of Plasma-Facing Components in Fusion Reactor

Fusion reaction is widely believed to be the ultimate energy resource, which unlike fossil fuel does not emit CO$_2$ contributing to greenhouse effect. An experimental stellarator, WENDELSTEIN 7-X (W7-X), is now being built in northeastern Germany [17, 18] to demonstrate steady-state fusion operation up to 30 min. This machine consists
of several plasma-facing components which require active cooling to prevent the components from melting. Amongst them, the most demanding component is the High Heat Flux Divertor which requires cooling of $8 \text{ MW/m}^2$ [18]. Removal of such high density of heat greatly challenges conventional liquid cooling technologies.

Metal-based MHEs provide a potential solution to this challenge. As shown in Fig. 2-5 of chapter 2, a $\sim 95000\text{W/m}^2\text{-K}$ surface heat removal capacity was achieved by one of the Cu-based MHEs. This means that the removal of $8\text{MW/m}^2$ would only result in a temperature rise as low as 84K in solids from the base temperature of water as the working liquid. Therefore it is theoretically achievable to maintain the Divertor under 200°C by adopting MHEs as its cooling device, provided that sufficient liquid flow is fed through the MHEs. In addition, mechanical robustness of the metal-based MHEs provides the needed reliability which is a key requirement for fusion reactors.

1.2.3 Cooling of High-Speed and Heavily-Loaded Bearings

Heavily loaded bearings operating at high speeds can generate significant amount of heat. As high as $\sim 7.8\text{kW}$ of heat generation was reported by M. Flouros in a ball bearing testing at a rotational speed of 19000rpm, together with an outer ring temperature of $\sim 195^\circ \text{C}$ [19]. In order for a bearing to continuously run under normal operation, its temperature must be kept under a maximum critical temperature. One reason is because friction between steels increases with increased temperature. Mei et al. have reported significant differences in the coefficient of sliding friction in cases where steel contacting tests were run with and without active cooling by MHE — $\sim 0.025$ vs. $\sim 0.10$ [20]. In their study, testing ring temperature was much lower when MHE cooling was turned on—
~50°C with cooling vs. ~130°C without cooling. Also less frictional damage was observed on the steel surface when MHE cooling was enabled.

In real applications, oil often serves as the working fluid for cooling instead of water-based coolants. A typical engine oil possesses only ~1/4 of the value in thermal conductivity and ~1/2 in specific heat as compared to water [12]. This means use of engine oil as coolant significantly impairs the cooling performance. In order to compensate this effect, use of high-performance, metal-based MHEs becomes desirable, especially for cooling of heavily-loaded high-speed bearings with kilowatts of heat generation.

1.2.4 Summary

In general, MHEs are optimum for cooling applications where intensive heat is generated within a confined area. In other words, MHEs are designed for removal of high heat flux thanks to the microchannel’s inherent nature of high heat transfer rate. In other cases when fluids with poorer heat transport properties must be used as the coolant, the enhanced performance by using MHEs may compensate for the impairment from the coolant.

1.3 MHE Fabrication

1.3.1 Fabrication of Si-Based Microchannel Devices

Early microchannel devices were mostly based on Si [1, 4, 5, 6], since fabrication technologies for Si-based high-aspect-ratio microscale structures are most mature and most widely available. Fabrication of microchannels on a Si wafer combines two procedures—photolithography and anisotropic etching. In the photolithography process, a masking layer like SiO₂ or Si₃N₄ with photolithographic patterns was deposited onto the
Si wafer surface for determining the shapes of microchannels [21]. The subsequent etching process uses an alkaline solution, most commonly KOH, to remove Si from the wafer in order to form the microchannels [1]. The etching rate on the Si wafer is anisotropic. Specifically, the crystallographic plane \{111\} of Si has a much slower etch rate than all other planes [21]. Therefore the Si wafer orientation needs to be taken into consideration when planning microchannel design. Laser can be introduced in the microchannel fabrication before the etching process to allow for different microchannel cross sectional shape [21].

While Si-based MHEs were first deployed for high heat flux cooling, there were several major obstacles to their wide adoption. First, the anisotropic etching process can only work on single-crystal Si wafer for the fabrication of smooth microchannels. This greatly limits the potential applications of the Si-based MHEs, and considerably raises the cost for manufacturing a MHE device. Secondly, due to the brittle nature of Si, mechanical robustness of Si-based MHEs is very limited. This prevents their adoption in rough environments or in critical applications where high reliability is required. In addition, the material Si is not the first choice when it comes to heat transfer applications due to its lower bulk thermal conductivity value, as compared to metals like Cu and Al. Therefore the thermal resistance due to heat conduction within solid Si somewhat impairs the high-heat-transfer-rate nature of microchannels.

1.3.2 Fabrication of Metal-Based Microchannel

Metal-based MHEs hold advantages over Si-based ones in terms of heat transfer performance due to higher thermal conductivities of metals, as well as mechanical robustness and therefore potentials for building devices with low profiles and different
configurations. The following introduces several commonly available fabrication technologies for metal-based microchannel.

a) Micromilling

Milling is a well-established traditional machining technique that is capable of fabricating versatile three-dimensional parts in macroscale, especially with the aid of computer numerical control (CNC) modules integrated into the milling machine. This idea was able to be adopted into the microscale with the aid of ultra-precision CNC machines with ultra-high-speed spindles, which forms the basis for the micromilling technology. Micromilling was used to fabricate different types of miniature parts [22, 23]. Parts fabrication on metallic materials were also studied and achieved [24, 25, 26]. Micromilling is versatile in terms of the variety of patterns it can generate, which includes microchannels.

The major issue with micromilling as a mass production technique comes down to the tool wear problem [26]. In contrast to traditional macroscale mechanical milling, micromilling requires microscale milling tools. Being in the micro realm, these tools are much more sensitive to tool wear. This causes more frequent tool breakage and therefore more disruptive events that break continuous machining process. Tool wear also compromises high degree of precision which is usually required in micromilling. Another problem of micromilling is the occurrence of burrs. Although it can be reduced by choosing smaller milling tools, this in return means higher costs and longer machining time [26].
b) Electrical Discharge Machining (EDM)

EDM is a process that removes electrically conductive material by localized melting/evaporation caused by electrical discharging. It is achieved by applying a high-frequency voltage between an electrode and the work piece. By approaching the electrode to the work piece to a minimum gap, localized electrical discharge occurs and melts/evaporates the material in the immediate vicinity of the discharge. Material removal occurs both from the electrode and the work piece. Both sinker and wire forms of EDM have been used for micro-manufacturing [27, 28], and they are both capable of fabricating microchannels. Sinker EDM is used for microchannel fabrication by applying a thin blade as the electrode. However, fabrication of a microchannel array in the sinker EDM mode would require a sequential series of cuts performed on the work piece, which turns out to be very time-consuming. Generation of microchannels by wire EDM is likewise a serial operation, with sequential wire traversing required in creating each microchannel profile.

Because EDM process removes material without physical contact with the electrode, it enables fabrication of microchannels on hard, brittle and refractory materials [26]. However, the downside of EDM process is that it typically consumes longer time than most other manufacturing techniques. This is intrinsic to the nature of this material removal mechanism.

c) Molding Replication

Molding replication produces microchannels by pressing a mold insert with a series of parallel micro-protrusions into a metal coupon, usually at an elevated temperature. Following the molding step, de-molding retracts the micro-protrusions out
from the molded metal coupon and leaves parallel microchannels on the coupon surface, as shown in Fig. 1-2 [29]. While both micromilling and microEDM produce microchannels in a sequential manner, the compelling advantage of the molding replication technique is the parallel nature of pattern generation and the speed with which the replication process happens. Replication allows for a large number of replicas to be produced repeatedly as long as the mold insert stays intact after de-molding, therefore offering a possibility of a low-cost, high-throughput production of microchannels.

![Figure 1-2. (a) A mold insert fabricated from ICONEL alloy. (b) Microchannels fabricated by molding replication on top of a Cu coupon. Cited from reference [29].](image)

To achieve repeated use of the mold insert, the biggest challenge that the molding replication technique faces is to protect the mold insert from damages caused by chemical/mechanical interactions between the mold insert and the molded metal at elevated molding temperatures. Surface engineering studies have been done to prevent such interactions, largely by depositing a layer of ceramic coatings onto the surface of the mold insert [30, 31, 32].

**1.3.3 Assembly of Metal-Based MHE**

In order to form an enclosed and functional microchannel device, an open microchannel array, like the one shown in Fig. 1-2(b), needs to be bonded to a cover plate
to form an enclosed and leak-tight device. Eutectic bonding with melting temperature depressing intermediate layers is often used to join together the components of metal-based MHEs, because it lowers the bonding temperature and prevents bulk melting of the parts to be joined, promotes interfacial inter-diffusion between the components. By controlling the kinetics of the bonding process, the bonding strength and bonding quality can be adjusted. The Al-Ge system was studied previously [33] and reported to be successful in assembling Al-based MHEs [8, 34]. For Cu-based MHEs, both Al-Cu and Sn-Cu systems have eutectic reactions and can be used for device assembly [26]. They have been studied for assembling Cu-based MHEs [8, 35].

1.4 Summary

MHEs are optimum for cooling applications where concentrated heat was generated within a confined area. As previously stated, MHE is a promising candidate to, but not limited to, cooling of electronic modules, plasma-facing components and heavy duty bearings. Since fabrication technologies for metal-based microchannels have been advancing rapidly and now becoming more mature, metal-based MHEs should be the first choice for practical uses over Si-based ones, thanks to the high thermal conductivity of metals like Cu and Al, increased mechanical robustness, and therefore potential for building devices with low profiles and different configurations.

This dissertation mainly focuses on studies of Cu-based MHEs because of the superior thermal conductivity of Cu. Chapter 2 studies building and testing of low-profile MHEs and the approaches to design optimization. In chapter 3, a commonly-used coolant, ethylene glycol, is applied to MHE, and the corresponding flow resistance and heat transfer performance are quantified. Chapter 4 deals with the interfacial thermal
resistance of the TLP bonding interface region. In chapter 5, double-layer microchannel configuration is deployed and studied. Chapter 6 studies integration of MHEs into a close-loop liquid-recirculation cooling system. Chapter 7 demonstrates an alternative roll molding method for continuous fabrication of metal-based microchannel structures. In the end, chapter 8 gives a summary.
CHAPTER 2.  FABRICATION, ASSEMBLY, AND HEAT TRANSFER TESTING OF LOW-PROFILE COPPER-BASED MICROCHANNEL HEAT EXCHANGERS

2.1  Introduction

The proliferation of personal and large-scale computing devices constitutes one defining feature of twenty-first century societies around the world. This proliferation is enabled by the dramatic increase in the computing power of microelectronic devices over the past half century, with capacity doubling approximately every two years [10]. With the increase in computing power comes an equally dramatic increase in the electronic module level heat generation [9]. The heat generated by electronic modules needs to be removed to ensure reliable operations below a maximum allowable temperature.

Cooling of current high performance electronic modules pushes the limit of the conventional heat management solution, which combines solid-metal, multiple-fin, heat sinks with forced-air convection cooling. As addressed in Chapter 1, microchannel heat exchangers (MHEs) are one of several alternative, liquid-based technologies currently being developed for cooling of next-generation electronic modules. Chapter 1 also discussed the advantages of adopting metal-based MHEs over Si-based ones. This chapter, therefore, focuses on the metal-based MHEs and reports on results of fabrication and bonding of low-profile Cu MHEs with varying geometric dimensions. The heat transfer performance of Cu MHEs is then quantified experimentally. The effect of microchannel geometries on heat transfer is examined in conjunction with preliminary finite element analysis (FEA). The present results demonstrate the potential of using low-profile metal-based MHEs for high heat flux cooling applications.
2.2. Experimental Procedures

2.2.1 MHE Fabrication, Assembly, and Examination

Low profile MHEs were fabricated from Cu110 alloy (99.9% Cu) sheets. In general, one MHE is composed of two Cu sheets: a 41mm×41mm×1.2mm base sheet and a 41mm×41mm×0.5mm cover sheet. A microchannel array with an active area of 41×25mm was fabricated by micro electrical discharge machining (µEDM) on the surface of the base sheet. Prior to µEDM, the Cu base sheet was polished with silicon carbide papers of decreasing grit sizes down to 800. Flat carbon steel sheets with different thicknesses were used as blade electrodes to create microchannels with different widths. A series of parallel cuts were made onto the base sheet surface sequentially to form an open and parallel microchannel array.

To form an enclosed Cu MHE, the Cu base sheet with an open and parallel microchannel array was bonded to a blank Cu cover sheet using Al thin foil intermediate layer bonding. Bonding experiments were carried out on a MTS858 single-axis mechanical testing system interfaced to a vacuum chamber containing two heating stations. The top heating station was connected through a vacuum feedthrough to the hydraulic actuator of the MTS system, through which a pressure can be applied to objects placed in between the two heating stations while heating takes place. Prior to bonding of the as-cut Cu base sheet to the thin Cu cover sheet, the faying surfaces were lightly polished with SiC papers. The cover sheet and base sheet were placed face-to-face on the bottom heating station, with a thin Al foil (Al1100, 99at.%+, ~25μm in thickness) inserted in the middle. After the chamber was evacuated, both heating stations were heated. Bonding of the Cu microchannel sheet to the Cu cover sheet was carried out at a
temperature of ~600°C with an applied pressure of ~3MPa. Bonded Cu MHEs were further connected to cylindrical Cu liquid supply/drain tubes using either Sn soldering or epoxy sealing.

Scanning electron microscopy (SEM) examination of the morphology and geometry of Cu MHEs was carried out on a Hitachi S3600N instrument. For examination of buried bonding interfaces and microchannels, non-contact cutting via a focused Ga⁺ ion beam was conducted on a FEI Quanta3D FEG Dual-Beam focused ion beam (FIB) instrument with a co-focal Schottky field-emission electron column and a high-current focused Ga⁺ ion column. Imaging carried out within the FIB instrument was obtained through either electron-induced secondary electrons (SEs) or ion-induced secondary electrons (ISEs). Imaging with ISEs affords additional contrast due to different crystal grain orientations through the ion channeling contrast mechanism, and is useful for delineating the crystalline microstructure of the examined region [36].

2.2.2. MHE Heat Transfer Testing: Setup, Specimen, and Measurement

Figure 2-1. (a) A cross-sectional SEM image of one portion of a typical low-profile Cu MHE used for heat transfer testing; (b) an optical image of the heat transfer test module for Cu MHE. Numbers shown on the ruler are in mm.
Five Cu MHE specimens with different microchannel dimensions were fabricated to test their heat transfer characteristics in a systematic fashion. Figure 2-1(a) shows a cross-sectional SEM image of a portion of one typical MHE specimen. The total thickness of the Cu MHE is ~1.7mm. This particular Cu MHE is assembled by bonding together a ~1.2mm thick base sheet and a ~500μm thick cover sheet. It is evident from Fig. 2-1(a) that the numerous contacts between the microchannel sidewalls on the base sheet and the cover sheet are well bonded, with no evidence of cross leakage.

To accurately measure the geometric dimensions of the Cu MHEs, the solid spacing between microchannels, $S$, channel height, $H$, and channel width, $W$, are measured from SEM cross-sectional images similar to Fig. 2-1(a) for each and every microchannel in the array. The average channel dimensions are obtained from raw measurements and shown in Table 2-1. Within Table 2-1, the microchannel hydraulic diameter, $D_h$, is defined as four times the channel cross-sectional area divided by the channel perimeter. In order to investigate the influence of microchannel dimensions on heat transfer performance, this series of Cu MHEs is separated into two groups: one group is of similar $S$ and different $D_h$, and the other group is of similar $D_h$ and different $S$. Both groups have similar channel height $H$. $D_h$ varies as channel width $W$ changes.

Table 2-1. Microchannel dimensions of Cu MHEs subjected to heat transfer testing.

<table>
<thead>
<tr>
<th>Cu MHEs</th>
<th>Average dimensions of microchannel arrays</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$N_c$</td>
</tr>
<tr>
<td>No. 1</td>
<td>70</td>
</tr>
<tr>
<td>No. 2</td>
<td>52</td>
</tr>
<tr>
<td>No. 3</td>
<td>52</td>
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<td>No. 4</td>
<td>35</td>
</tr>
<tr>
<td>No. 5</td>
<td>35</td>
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</tbody>
</table>
Figure 2-2. Experimental setup for heat transfer testing: (a) a schematic of the testing apparatus; (b) a schematic of the Cu MHE test module.

Figure 2-1(b) shows the setup of the Cu MHE test module. For heat transfer testing, a 24mm × 26mm × 82mm Cu heater block, containing four round holes for accommodating four cylindrical cartridge heaters, was bonded to the center of the MHE with Sn as the interfacial material. Before bonding, the faying surfaces were polished with SiC abrasive papers down to 800 grit size, followed by de-oxidation using 5% HCl.
solution. A Sn soldering paste, Kester EM907, was uniformly applied onto the surface of the heater block and heated up to melting. Then the MHE was placed onto the top surface of heater block with melted Sn and mechanically pressed to minimize the Sn interlayer thickness. Two Cu tubes with Swagelok® fittings served as the liquid supply and return plena. During the test, one of the tube ends of each tube was closed to form water pass through MHE. A slot was cut by μEDM into each Cu tube, and each end of the Cu MHE was inserted into the slot and sealed by epoxy. The placement of the Cu heater block is such that the entire width of the block is contained within the width of the microchannel array, ~25mm. It is also evident from Fig. 2-1(b) that the length of the Cu heater block does not completely cover the microchannel array, which runs through the entire length of the MHE specimen of 41mm.

Figure 2-2(a) shows the schematic of the testing apparatus designed and built to evaluate the heat transfer characteristics of Cu MHEs. The apparatus consists of three sections: liquid supply section, MHE test module, and data acquisition section. The liquid supply section consists of a holding tank and a diaphragm pump. Original water from local supplier was used as the liquid coolant without deionization or degassing. Water flow rate was adjusted by changing the pump voltage input.

Figure 2-2(b) shows a schematic of the MHE test module, depicting one MHE attached to the 24mm × 26mm × 82mm Cu heater block with Sn bonding. The heater block contains four ~200W cylindrical cartridge heaters. The Sn bonding provides uniform contact between the MHE and the heater block and significantly reduces interface thermal resistance, since the thermal conductivity of Sn, 66.6W/K-m, is much higher than that of typical thermal greases, 0.6~7.5W/K-m [37]. The whole test module
was enclosed in commercial semi-rigid polyvinyl chloride (PVC) foam insulation with part number 9318K78 from McMaster-Carr.com. The four 3/8” diameter high-temperature cartridge heaters with part number 35025K317 from McMaster-Carr.com, were powered by a dc power supply in a constant-voltage mode. During the test, a total of 400W was provided to the four cartridge heaters, which yielded a \( \sim 60\text{W/cm}^2 \) heat flux across the heater block/MHE interface.

K-type thermocouples with part number TJ36-CASS-116U-6 from Omega.com were used for all temperature measurements. Locations of thermocouple placement are shown in Fig. 2-2(b). Two thermocouples were inserted respectively into the liquid supply and drain tubes for measuring the temperatures of the inlet water, \( T_{in} \), and outlet water, \( T_{out} \). A linear array of four holes was drilled into the center of the heating block in a direction perpendicular to the heater block/MHE interface. Four thermocouples were inserted into the drilled holes to obtain, by extrapolation, a measurement of the temperature of the MHE surface contacting the heater block, \( T_{cs} \). The distance between neighboring thermocouples is 6mm, while the first thermocouple, T1, is 3mm away from contact surface. Another thermocouple was placed at the center of MHE surface away from the heater block, and measured the temperature of the MHE free surface, \( T_{fs} \). The entire MHE specimen was covered by PVC insulation. In the data acquisition section, a HP34970 data acquisition/switch unit interfaced to a PC was used to collect thermocouple readings. It should be pointed out that the water inlet/outlet thermocouples are placed within the liquid supply/drain tubes and that the Cu heater block does not extend the full length of the microchannel array. In the ideal case, the \( T_{in} \), and \( T_{out} \) measurements should be made respectively at locations immediately before water enters
under the heater block footprint and immediately after water exits under the block footprint. The present experimental setup therefore carries systematic errors in measurements of $T_{in}$, and $T_{out}$ for evaluation of heat transfer characteristics for the entire microchannel array. Further details on heat transfer characteristics of metal-based microchannel arrays of similar dimensions have been reported previously [29]. In the present chapter, the emphasis is placed on the overall heat transfer performance at the device level.

All data are recorded in steady state, including temperatures and water flow rates. Water flow rates are obtained by measuring the volume of water collected at the exit over a fixed 1 min time period. During this period, temperatures are recorded once every 5 seconds. The final temperature reading is taken as the average of 12 data points. Heat transfer measurements are performed at different volumetric water flow rates, which range from 250 to 2305 ml/min. The difference in outlet and inlet water temperatures ranges from 2 to 22°C.

2.3. Data Analysis

The average microchannel hydraulic diameter of each MHE specimen is defined by $D_h = 4A_c/p$, where the average microchannel cross-sectional area $A_c$ and average perimeter $p$ are obtained from SEM images similar to Fig. 2-1(a) by digital mapping for each and every microchannel using ImageJ software [38]. The steady state rate of heat absorption by water flowing through the MHE specimen, $\dot{q}$, is taken as

$$\dot{q} = \rho C_p \dot{Q} (T_{out} - T_{in}),$$

(2-1)
where $\dot{Q}$ is the water volumetric flow rate, $T_{out}$ and $T_{in}$ are respectively water temperatures measured by the thermocouples placed at the water inlet and outlet, and $\rho$ and $C_p$ are respectively water density and specific heat.

The average velocity of water flowing through the microchannel array is determined from the water volumetric flow rate, the total number of channels in the array $N_c$, and the channel cross-section area,

$$V_{avg} = \frac{\dot{Q}}{N_c A_c}. \quad (2-2)$$

The Reynolds number is then calculated from the average water velocity,

$$Re = \frac{\rho V_{avg} D_h}{\mu}, \quad (2-3)$$

where $\mu$ is the dynamic viscosity of water. All physical properties of water, including density, specific heat, and dynamic viscosity, are evaluated using curve-fitting method at the mean water temperature, defined by $T_m = (T_{out} + T_{in})/2$. The maximum curve-fitting errors as compared to the original data [12] are less than 0.1% for density, 0.01% for specific heat, and 0.5% for dynamic viscosity.

The unit area input admittance, the parameter for quantifying heat transfer performance of the MHE, is given by

$$\bar{Y} = \frac{q}{A_{cs}(T_{wall} - T_m)}, \quad (2-4)$$

where $A_{cs}$ is the nominal area of the contact surface between the MHE specimen and the heater block, $T_{wall}$ is the microchannel wall temperature which is defined as $T_{wall} = (T_{cs} + T_{fs})/2$, where $T_{cs}$ and $T_{fs}$ are respectively temperatures at the MHE specimen’s contact surface with the heater block and the free surface away from the heater block. $T_{cs}$ is obtained from linear extrapolation of the temperature measurements made from the
four thermocouples inserted into the linear array on the heater block body, $T_1$ to $T_4$, i.e.,

$$T_{cs} = \frac{(T_1 + T_2 + T_3 + T_4)}{4} - \frac{12(T_4 - T_1)}{18}$$

based on the present thermocouple locations. Due to the small total thickness of the Cu MHE specimens, the difference between $T_{cs}$ and $T_{fs}$ is within 3°C in all tests performed, suggesting that $T_{wall}$ as defined here offers a reasonable estimate of the true microchannel wall temperature.

The effective convective heat transfer coefficient for the microchannel array is given by

$$h_{eff} = \frac{q}{A_{cw}(T_{wall} - T_m)}, \quad (2-5)$$

where $A_{cw}$ is defined as the microchannel wet area covered by the heater block footprint, which is the microchannel perimeter multiplied by the heater block length multiplied by the total number of channels, i.e., $A_{cw} = N_c \times p \times L$ where $L$ is the length of the Cu heater block, ~26mm.

Thermal conduction through the solid microchannel sidewalls influences the heat transfer effectiveness of the microchannel. To better quantify the influence of the channel-to-channel solid spacing $S$ on heat transfer performance, a free surface efficiency parameter, $\eta_{fs}$, is defined as

$$\eta_{fs} = \frac{T_{fs} - T_m}{T_{cs} - T_m}, \quad (2-6)$$

In the best case, $\eta_{fs}$ approaches 1 and the free surface temperature is approximately the same as the temperature of the contact surface. As $S$ decreases to a point such that thermal conduction through the microchannel sidewalls becomes insufficient, $\eta_{fs}$ will be expected to drop significantly below 1.
2.4. Results and Discussion

![Figure 2-3](image_url)

Figure 2-3. Examples of low-profile Cu MHEs: (a) A partially assembled, low-profile MHE with water jets exiting from the open end; (b) a cross-sectional ISE image of the bonding interface region of the MHE shown in (a). The solid arrow marks the bonding interface region, and the dashed arrows mark the voids formed during the bonding process; (c) a cross-sectional ISE image of another Cu MHE. Focused Ga⁺ ion beam cutting was used to reveal a corner of one buried microchannel.

One example of low-profile, Cu-based MHEs is shown in Fig. 2-3(a), which shows a partially assembled, low-profile MHE with one end attached to a liquid supply tube and the other end open. This particular MHE has an area footprint of 41mm×41mm and a total thickness of ~600μm. It was built from a ~500μm thick base sheet and a ~100μm thick cover sheet through bonding with ~10μm thick Al foil. The base sheet contains an array of 24 parallel and nominally identical microchannels with both width and depth of ~250μm. Once water is supplied from the liquid supply tube, 24 separate micro water jets exit the open end of the MHE. Bonding of the Cu base and cover sheets with an Al intermediate thin foil layer takes advantage of a eutectic reaction between Cu and Al at ~550°C [39]. As the temperature of the Cu/Al/Cu interface reaches ~550°C, the interface region starts to melt. Cu from the base and cover sheets dissolves into the interfacial melt, increases the overall composition of Cu within the melt, and eventually leads to isothermal solidification of the interfacial region. Such a transient liquid phase (TLP) bonding strategy [40] lowers the bonding temperature, and minimizes distortion of
the microscale features to be bonded. Figure 2-3(b) shows a cross-sectional ISE image of the bonding interface of the MHE specimen shown in Fig. 2-3(a), from a region in between two microchannels. In order to minimize ion beam cutting time, the top cover sheet was thinned by mechanical polishing prior to FIB cutting, which proceeded until the buried microchannel was revealed. It is evident from Fig. 2-3(b) that TLP bonding of Cu using an Al thin foil intermediate layer results in a well-formed bonding interface region. Figure 2-3(b) shows that some voids form within the interface region during the bonding process. The bonding interface region possesses a somewhat complex microstructure, which is delineated in Fig. 2-3(b) via the ion channeling contrast [36]. Further structural characterization shows that this complex microstructure is made up of a mixture of Cu terminal solid solution and a Cu-Al intermetallic compound [35]. Such a micro/nano scale two-phase mixture may serve to reduce brittleness of the bonding interface region.

Figure 2-3(c) shows a cross-sectional ISE image of another Cu MHE, in which FIB cutting is used to reveal the region near a corner of one microchannel. It is evident from Fig. 2-3(c) that Cu/Al/Cu TLP bonding yields good quality joints near microchannel corners, leaving no crevices where microchannel sidewalls meet with the top cover sheet. Results shown in Fig. 2-3 demonstrate that the TLP bonding strategy can be used to assemble low profile, “credit-card-like”, Cu MHEs with good structural integrity.

The effective microchannel convective heat transfer coefficient $h_{eff}$, as defined in Eq. (2-5), characterizes liquid-solid convective heat transfer rate at the individual microchannel level. The values of $h_{eff}$ for the group of three Cu MHEs, with similar average hydraulic diameter $D_h$ but with different average channel-to-channel spacing $S$,
are plotted against the Reynolds number $Re$ in Fig. 2-4(a). For another group with similar $S$ but with different $D_h$, the values of $h_{eff}$ are plotted against the Reynolds number $Re$ in Fig. 2-4(b). For all the MHEs in Fig. 2-4(a) and Fig. 2-4(b), $h_{eff}$ increases with increasing $Re$. In Fig. 2-4(a), MHEs with similar $D_h$ values possess similar individual microchannel heat transfer performance $h_{eff}$; Fig. 2-4(b) indicates smaller $D_h$ results in higher values of $h_{eff}$. This suggests that individual microchannel heat transfer performance $h_{eff}$ is determined by hydraulic diameter $D_h$—smaller $D_h$ leads to higher performance [4, 10]. In Fig. 2-4(a), similar $h_{eff}$ values are observed for specimens with different channel-to-channel spacing $S$. That indicates individual microchannel heat transfer performance is largely independent of $S$ in the current range of $S$.

![Figure 2-4](image-url)

Figure 2-4. Effective microchannel convective heat transfer coefficient $h_{eff}$ versus $Re$ for two MHE groups: (a) MHE group with similar average hydraulic diameter $D_h$ but with different average channel-to-channel spacing $S$, (b) MHE group with similar $S$ but different $D_h$.

The unit area input admittance $\bar{Y}$, as defined in Eq. (2-4), characterizes the overall device-level heat transfer performance of the MHE. From the definitions in Eq. (2-4) and in Eq. (2-5), it can be derived that

$$\bar{Y} = h_{eff} \times (A_{cw}/A_{cs}),$$  \hspace{1cm} (2-7)
where $A_{cw}$ is the channel wet area covered by the heater block footprint, and $A_{cs}$ is the nominal area of the contact surface between the MHE specimen and the heater block. Eq. (2-7) indicates that the overall device-level heat transfer performance $\bar{Y}$ is determined by two factors—the individual microchannel heat transfer performance $h_{eff}$ and the ratio between channel wet area $A_{cw}$ and contact surface area $A_{cs}$.

Fig. 2-5(a) shows the results of $\bar{Y}$ for the same MHE group in Fig. 2-4(a). MHEs with smaller values of channel-to-channel spacing $S$ have higher device-level heat transfer performance $\bar{Y}$, regardless of similar individual microchannel heat transfer performance $h_{eff}$ for each MHE demonstrated in Fig. 2-4(a). That is because smaller $S$ enables more number of microchannels to be packed into the fixed microchannel array width of $\approx 25$mm, i.e., more microchannels exist under the Cu heater block footprint, which increases the value of $A_{cw}/A_{cs}$.

Fig. 2-5(b) shows the results of $\bar{Y}$ for the same MHE group in Fig. 2-4(b). Fig. 2-5(b) shows larger differences in $\bar{Y}$ than that in $h_{eff}$ in Fig. 2-4(b). For instance at Reynolds number 1200, the value of $\bar{Y}$ for the MHE with $281 \mu m$ $D_h$ is 148% higher than that for the one with $617 \mu m$ $D_h$, while the value of $h_{eff}$ for the former MHE only exceeds 67% of that for the latter. The enlarged differences in $\bar{Y}$ shown in Fig. 2-5(b) is due to the $A_{cw}/A_{cs}$ factor, as the $A_{cw}/A_{cs}$ value for the former MHE is 4.76, while 3.27 for the latter. In conclusion, small microchannel size not only improves the microchannel-level heat transfer performance, but also increases the value of $A_{cw}/A_{cs}$ at given channel-to-channel spacing $S$, both leading to higher device-level heat transfer performance.
Figure 2-5. Unit area input admittance $\bar{Y}$ versus $Re$ for two MHE groups: (a) MHE group with similar average hydraulic diameter $D_h$ but with different average channel-to-channel spacing $S$, (b) MHE group with similar $S$ but different $D_h$.

The overall cooling capacity of the current series of Cu MHE specimens can be appreciated more intuitively through a simple manipulation of the present heat transfer data shown in Figs. 2-4 and 2-5. For the MHE specimen with $D_h$ and $S$ equal respectively to 281µm and 177µm, the rate of energy removal by water flowing through the microchannel array is $\sim$250W/cm$^2$ at a Reynolds number of $\sim$1200, assuming respectively $T_{cs}$ and $T_{in}$ values of 63°C and 25°C. That is, with a water inlet temperature of 25°C and a Reynolds number of 1200, this particular Cu MHE is capable of dissipating 250W/cm$^2$ of input heat flux while keeping the contact surface temperature at 63°C. For the current personal computer CPUs, the maximum chip-level cooling requirement is 80W/cm$^2$ [11]. For this MHE specimen, a $Re$ value of 1200 corresponds to a total volumetric flow rate of $\sim$1.8liter/min and a total pressure drop across the MHE specimen of $\sim$6.7psi. These values demonstrate that high cooling capacities can be afforded by Cu MHEs at reasonable liquid flow rates and pressure drops.
Figure 2-6. Free surface efficiency versus $Re$ for MHE group with similar average hydraulic diameter $D_h$ but with different average channel-to-channel spacing $S$.

As $S$ decreases, eventually a significant drop in the temperature of the microchannel sidewall away from the heat source is expected due to insufficient thermal conduction through the solid microchannel sidewall. On the other hand, larger $S$ means that less number of microchannels can be packed within a given area footprint. Therefore, an optimum channel-to-channel spacing $S$ should exist for the best overall MHE heat transfer performance. The free surface efficiency $\eta_{fs}$, as defined in Eq. (2-6), is plotted against $Re$ in Fig.2-6 for the group of MHE specimens whose $\bar{Y}$ and $h_{avg}$ data are shown in Figs. 2-5(a) and 2-4(a). Figure 2-6 shows that $\eta_{fs}$ values range from ~75% to over 90% for all three MHE specimens, and show little difference at different $S$ values. The convergence of curves in Fig. 2-6 suggests that the microchannels barely feel the difference in channel-to-channel spacing, and that the current range of channel-to-channel solid spacings is adequate for conducting heat from the contact surface to the free surface of the MHE specimens. The observed high free surface efficiency further
suggests that building multiple-layered MHEs is a feasible approach for further increasing the overall cooling capacity of metal-based MHE devices.

Figure 2-7. A schematic of the 2D heat transfer FEA model implemented using ANSYS.

Further investigation of the optimum channel-to-channel spacing $S$ is conducted through a 2D numerical model, as shown schematically in Fig. 2-7. A single-layer, parallel, periodic microchannel array is modeled through FEA using the commercial package ANSYS 11.0 [41]. As shown in Fig. 2-7, one half of one microchannel + solid sidewall unit is modeled by invoking the periodic boundary condition. A heat flux of 60.9W/cm$^2$, the same as in the experiments, is applied to the bottom side which is the contact surface in the experimental setup. The top surface is insulated, as in the experiments. No heat flux crosses the two side boundaries due to the periodic boundary condition. A constant convective heat transfer coefficient $h$ is assumed for all microchannel internal surfaces. The highest value of $h_{\text{eff}}$ observed in the experiments, 22574W/m$^2$-K, is applied to $h$ in the simulation, along with two other $h$ values of 15000W/m$^2$-K and 30000W/m$^2$-K. A uniform water temperature is assumed within an entire microchannel cross section. A water temperature of 30°C is used in the simulation.
Figure 2-8 shows values of $T_{cs} - T_{water}$ at different channel-to-channel spacings $S$, at the three different assumed $h$ values. All three curves show a similar trend. At large $S$ values, $T_{cs} - T_{water}$ increases with increasing $S$ in an approximately linear fashion. In this linear portion of the curve, $T_{cs} - T_{water}$ increases with increasing $S$ because of the reduction of the total number of channels per unit width. The linearity indicates that the channel-to-channel spacing is adequate for heat conducting from the contact surface side to the free surface side, leading to insignificant temperature drop along the microchannel sidewall. As $S$ decreases, a minimum is observed in the $T_{cs} - T_{water}$ vs. $S$ curve. At even smaller $S$ values, $T_{cs} - T_{water}$ increases with further decreases in $S$, due to inadequate thermal conduction through microchannel sidewalls. The $S$ value corresponding to the minimum in $T_{cs} - T_{water}$ corresponds to an optimum in the overall heat transfer performance of the MHE, balancing the number of microchannels per unit width with adequate thermal conduction through solid microchannel sidewalls. As shown in Fig. 2-8, the optimum $S$ values corresponding to $h$ values of 15000, 22574, and
30000W/m²-K are respectively 45, 55, and 60μm, and are in all cases significantly below the range of current experimental $S$ values. The insensitivity of the experimentally observed $\eta_{fs}$ values with respect $S$ is therefore consistent with the present FEA results.

2.5. Summary

This chapter has demonstrated experimentally that it is feasible to build low profile, credit card like, metallic microchannel heat exchangers through simple fabrication protocols, i.e., microchannel pattern creation in thin sheet metals followed by low to medium temperature bonding and assembly. Cu-based MHEs of different microchannel dimensions are built and experimentally tested for their heat transfer performance. Cooling capacities of ~250W/cm² can be achieved in Cu MHEs at reasonable liquid volumetric flow rates and pressure drops. A simple 2D FEA model adequately captures major factors influencing overall heat transfer performance of metallic MHE devices. The present results show promise of low-profile, metal-based MHEs for high heat flux cooling applications.
CHAPTER 3.  FABRICATION OF METAL-BASED MICROCHANNEL HEAT EXCHANGERS AND QUANTIFICATION OF THEIR LIQUID FLOW AND HEAT TRANSFER CHARACTERISTICS

3.1 Introduction

As addressed in Chapter 1, intense current interests exist in using liquid flow through microchannels and the accompanying liquid-solid heat exchange for applications demanding removal of highly concentrated heat, including next-generation micro-electronic and power-electronic modules. For the design and performance optimization of MHE devices, detailed data on how liquid flow and liquid-solid heat transfer depend on microchannel geometry/morphology and on the physical properties of the liquid medium are needed.

This chapter describes recent research and development activities on fabrication and assembly of metal-based MHEs, as well as experimental quantification of their liquid flow and heat transfer characteristics. Adoption of metal-based MHEs in practical applications necessitates quantification of flow and heat transfer characteristics with application-relevant coolants. Ethylene glycol (EG)/water mixtures are used in many cooling applications in place of pure water, e.g., automotive cooling, because of their extended range of operating temperatures. In such applications, chemical additives are often incorporated into EG/water mixtures. Reliable physical properties of pure EG and pure water are available and are given in Table 3-1 [12]. In flow and heat transfer testing, accurate physical property values for the liquid medium are crucial to the correctness of the final results. Physical properties of liquids can change significantly with temperature, e.g., the value of EG viscosity increases ~30 times as temperature varies from 100 to 0°C.
While physical properties of EG/water mixtures may be calculated and the influence of chemical additives estimated, directly measured property values are preferred due to such dramatic property changes.

Table 3-1. Physical properties of pure EG and pure water.

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<th>T (°C)</th>
<th>ρ (kg/m³)</th>
<th>cₚ (kJ/kg-K)</th>
<th>μ (Pa·s)</th>
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As one further step toward using metal-based MHEs in practical applications, this chapter reports fabrication and assembly of all-Cu MHE prototypes, as well as results of liquid flow and heat transfer testing on all-Cu MHEs using only pure EG and pure water as the liquid medium. Results of heat transfer testing indicate sensitivity of overall heat transfer performance to entrance length effects, which in the case of pure EG, is significantly influenced by its physical properties under actual testing conditions.

In what follows, a brief review of several competing methods for fabricating metal-based microscale structures is given first. The technique of directly replicating metal-based HARMS by compression molding is described, together with previous
efforts on bonding metal-based HARMS to form completely enclosed metallic microchannel devices. Next, detailed descriptions of experimental procedures for fabricating Cu-based MHEs and for quantifying their liquid flow and heat transfer characteristics are given. Following experimental procedures, results and discussion are presented with emphasis on flow and heat transfer characteristics with pure EG as the liquid medium. Finally, a summary is given.

3.2 Fabrication and Assembly of Metal-Based Microchannel Devices

Metallic HARMS are essential building blocks for a variety of metal-based microdevices, including MHEs. Among the techniques for making metallic HARMS, the LiGA (Lithographie, Galvanof ormung, Abformung) technique, based on deep lithography and electrodeposition, holds historical importance [42]. In the traditional LiGA approach, a primary microscale pattern is generated in a polymeric resist by X-ray lithography. The penetrating power of X-rays, usually from an electron synchrotron source, creates vertical structures in the resist ranging from a few hundred μm to a few mm in height. Chemical dissolution of resist in areas exposed to X-rays is followed by metal electrodeposition into developed resist recesses. Dissolution of the remaining resist after electrodeposition yields primary metallic HARMS. The high cost of deep lithography and the slow speed of metal electrodeposition conspire to make primary metallic HARMS too expensive for commercial deployment. Secondary HARMS can be replicated from a primary HARMS by molding replication. The molding process produces numerous secondary microparts from one primary microscale mold insert, and is attractive because it can potentially lower fabrication cost and increase production throughput [43]. Although replication of secondary HARMS by molding using
electrodeposited primary metallic HARMS as mold inserts was proposed over twenty years ago [42], up to 2002 only polymer-based HARMS have been replicated from primary metallic HARMS inserts by compression molding [44] or injection molding [45].

Alternative techniques for fabricating metallic HARMS have been explored. Micromilling (µMIL) with tools fabricated with a focused ion beam technique extends the traditional metal cutting approach to the micro realm [24]. Micro electrical-discharge-machining (µEDM) have been used to machine HARMS out of alloys [28] and electrically conducting ceramics [46]. Techniques derived from the LiGA approach include micro powder injection-molding (µPIM) [47] and micro casting (µCAS) [48]. Serial subtractive techniques such as µMIL and µEDM are time consuming, suffer from tool wear, and may have limitations when fabricating microscale features in close proximity. The µPIM technique requires multiple heat treatment steps, and suffers from incomplete mold filling and part shrinkage during heat treatment [49]. The µCAS technique also requires multiple heat treatment steps, and is a “double lost mold” technique. The disadvantages associated with each of these alternative techniques motivate further searches of effective techniques for fabricating metallic HARMS.

Since 2001, Meng’s group at Louisiana State University Mechanical Engineering has been investigating the feasibility of direct replication of metallic HARMS by compression molding of metals with metallic microscale mold inserts. It was shown that the process of high temperature micromolding of metals is strongly influenced by chemical/mechanical interactions between the molded metal and the mold insert [50, 51]. Conformal deposition of suitable nanostructured ceramic coatings over the entire insert
surface was shown to be an effective means of engineering the chemical, mechanical, and tribological properties of the near-surface region of metallic mold inserts [30, 31, 32].

Besides the need to engineer the chemical and mechanical interactions between the molded metal and the mold insert, the bulk mechanical properties of LiGA fabricated inserts restrict the range of metals and alloys which can be compression molded [52]. To further improve this, microscale mold inserts have been fabricated out of refractory metals such as Ta [53] and refractory alloys such as the Ni-based superalloy Inconel X750™, followed by deposition of conformal coatings for insert surface modification [54]. Using such surface engineered, refractory, microscale mold inserts, direct molding replication of HARMS was successfully demonstrated in softer metals such as Al [51] and harder metals such as Cu, Ni, and NiTi [55, 56]. Some understanding of the mechanics of compression micromolding of metals has been achieved [57, 58, 59, 60]. Compared to the alternative techniques described above, direct replication of metallic HARMS by high temperature compression micromolding is fast and simple, and holds potential advantages in production cost and throughput.

Replicated metallic microchannel structures need to be bonded to other structured or flat metallic plates in order to form enclosed microchannel devices. Transient liquid phase (TLP) bonding of Al plates containing microchannel structures to other Al plates was achieved using Al-Ge eutectic thin film intermediate layers [34]. Tensile strength of Al-Ge eutectic bonding was evaluated as a function of bonding conditions [61]. Mechanisms of using Al-Ge eutectic intermediate layers for bonding Al-based structures were investigated [62]. Bonding and assembly of Cu-based microchannel devices were demonstrated through the use of thin foil Al [63] and Sn [64] intermediate layers.
Combining microscale molding replication and TLP bonding, entirely Al- and Cu-based MHEs were fabricated and assembled. Preliminary heat transfer testing was performed under conditions approximating either a constant heat flux configuration, in which heat flux was supplied via cartridge heaters mounted on one side of the MHE device [65], or a constant solid surface temperature configuration, in which the MHE device was immersed in a constant temperature water bath [29]. In both tests, pure water was used as the liquid medium.

3.3 Experimental Procedures

3.3.1 Molding Replication of Cu-Based Microchannels

Figure 3-1. Scanning electron microscopy overview of an a-Si:N coated, microscale, Inconel mold insert.

Microscale mold inserts were fabricated from Ni-based superalloy Inconel X750™ plates, see for example [29]. As-received Inconel plates were machined to square insert blanks, with an active area of 20.0mm×20.0mm, 3.2mm in height. Insert fabrication from the blanks involved three main steps: micro electrical discharge machining (μEDM) of the active area, electrochemical polishing (ECP) of as-machined
microscale Inconel features, and deposition of a conformal amorphous silicon nitride (a-Si:N) coating over electrochemically polished microscale features. A scanning electron microscopy (SEM) overview of one such a-Si:N coated Inconel insert is shown in Fig. 3-1. The thickness of the a-Si:N coating deposited onto the insert surfaces varied with location, ranging from ~600nm on the top surfaces of the rectangular protrusions to ~300nm on the protrusion sidewall surfaces. Further results on characterizing coatings deposited on microscale mold inserts were reported elsewhere [31, 32].

The active surface of this insert contains 26 parallel rectangular microprotrusions, with a center-to-center spacing of 750±5μm. The average width of all microprotrusions, as measured from SEM micrographs, is 180±5μm. The height of the straight section of the microprotrusions is ~400μm. Due to the combined action of μEDM and ECP, numerous micron and submicron scale features exist on the microprotrusion surfaces. Typical surface roughness values on the microprotrusions, measured via non-contact optical profilometry (OP) using a Wyko NT3300 instrument, ranged from 1-10 μm [29, 65].

Cu 110 (99.9+wt.% Cu) coupons, 42mm×42mm square by 5mm in thickness, were polished with silicon carbide papers of various grit sizes, finished with 1μm diamond suspension, and molded with the a-Si:N coated Inconel insert shown in Fig. 3-1. Compression molding was carried out on a MTS858 single-axis testing system interfaced to a high-vacuum chamber. The vacuum chamber housed two heating stations, one for the Cu coupon and the other for the mold insert. Molding of Cu coupons occurred with both the coupon and the insert heated to 500±10°C. The molding process replicated patterns on the insert into an array of 26 parallel rectangular microchannels on the Cu
coupon, with an average width of 180±5µm and a depth less than 400µm. Further details on mold insert fabrication, coating deposition, and protocols for molding replication have been reported elsewhere [51, 58].

3.3.2 Bonding and Assembly of Cu-Based MHEs

Figure 3-2(a) Overview of a typical pair of Cu coupons, one molded and one blank. An array of parallel microchannels exists between the supply and drain plena on the molded coupon. Numbers on the ruler are in mm. (b) Detailed view of one fluidic plenum on the molded Cu coupon. (c) High magnification view of one typical replicated Cu microchannel.

One liquid supply plenum and one liquid drain plenum were further machined using μEDM onto a blank Cu coupon and a molded Cu coupon to provide connections to replicated microchannel arrays. Flat stainless steel sheets with a thickness of 1100µm were used as blade electrodes for μEDM. Six consecutive cuts on each end of the
microchannel array with partial overlap were performed to create the supply and drain plena. Depth of cuts, ~1.5mm, were kept the same for all cuts on both the supply and drain sides. Surfaces of both the molded coupon and the blank coupon were lightly polished with 1μm diamond suspension after plena cutting with μEDM. Figure 3-2(a) shows a typical pair of molded and blank Cu coupons, with the liquid supply and drain plena already machined into them.

As shown in Fig. 3-2(a), mechanical drilling was used to make two large 4.1mm diameter through-holes and one small 2.0mm diameter through-hole within each plenum. The through-holes were placed symmetrically with respect to the microchannel array. All large through-holes were mechanically tapped from the coupon back side, away from the microchannel array, to allow external liquid connections using plastic adaptors. The use of double inlet and outlet ports enables the placement of pressure taps in the center of the plena and facilitates more uniform flow through the microchannel array. The choice of the 5mm coupon thickness accommodated the placement of taps. The small through-hole within each plenum connected to a small stainless steel tube, which in turn provided connection for a digital manometer and a thermocouple.

Figure 3-2(b) provides a more detailed view of one plenum on the Cu coupon. The plenum has the mottled surface morphology typical of structures cut by μEDM, and provided transitions from the plenum to the microchannels without obstructions at the entrances. Figure 3-2(c) shows a high magnification view of one typical replicated Cu microchannel. The microchannel has vertical sidewalls and sharp sidewall-to-bottom transitions. Micron scale surface roughness is clearly visible on the sidewall and bottom of the microchannel. The bottom roughness appears random, while the sidewall
roughness exhibits clear streaking in the molding direction, resulting from insert motion into Cu during molding. The peak-to-valley roughness $R_z$, as measured via OP, was ~5µm for both bottom and sidewall surfaces of the replicated microchannels. Further details on the surface roughness measurement were reported previously [29, 65].

Bonding experiments were also carried out on the MTS858 system interfaced to the vacuum chamber. Molded and blank Cu coupons, as shown in Fig. 3-2(a), were placed face to face on the bottom heating station with a thin Al foil (Al1100, 99%+, ~10µm in thickness) inserted in the middle. After the chamber was evacuated, both heating stations were heated. Bonding of Cu coupons was carried out at a temperature of ~600°C with an applied pressure of ~3MPa. Further details on the Cu TLP bonding protocol and evaluation of bond quality were given elsewhere [63, 64]. Figure 3-3 shows an optical overview of one bonded Cu MHE prototype, with plastic external fluid adaptors attached and the small through-holes exposed.

![Figure 3-3. Overview of the assembled Cu MHE specimen without the stainless steel tubes inserted into the small through-holes.](image-url)
3.4 Fluid Flow and Heat Transfer Measurements

3.4.1 Experimental Setup

![Schematic of the experimental setup for fluid flow and heat transfer measurements](image)

Figure 3-4. (a) Schematic of the experimental setup for fluid flow and heat transfer measurements. (b) Schematic of the thermocouple fixture. (c) Schematic cross-sectional and plan views of the test section, together with thermocouple locations (shown as black dots).
A testing apparatus was designed and built to evaluate the liquid flow and heat transfer characteristics of the assembled Cu MHE prototype. Figure 3-4(a) shows a schematic of the testing apparatus, which consists of three sections: liquid (pure water or pure EG) supply section, test section, and data acquisition section. The supply section consisted of a pressure-regulated, steel, liquid storage tank. The tank was pressurized by high-pressure, pure (99.9%+) nitrogen gas, and supplied liquid to the MHE specimen at a constant pressure and provided a smooth and stable flow through the microchannels even at low flow rates. A valve placed downstream of the tank exit was used to make fine adjustments to the flow rate. The placement of the liquid outlet tube deeper in the tank as compared to the height of the nitrogen inlet tube ensures that no gas bubble would enter the liquid supply to the testing section. In the case where supply of elevated temperature liquid is desired, two electric tape heaters were attached to the tank exterior and used to heat the liquid within. When EG was used as the liquid medium, nominally pure EG (99%) was placed within the liquid storage tank, with pressurized nitrogen. While EG is hygroscopic, water incorporation into the EG liquid should be minimal with this experimental arrangement.

During heat transfer testing, the Cu MHE prototype was completely immersed in an external water bath contained in a 2000ml glass beaker. Two different inlet liquid conditions, “room temperature (RT)” and “high temperature (HT)”, were applied to conduct the heat transfer performance of the Cu MHE device with pure EG used as the liquid medium. When the inlet EG liquid was RT, the external water bath was heated using a hot plate, continuously stirred, and provided a constant-temperature environment surrounding the entire Cu MHE specimen. When the inlet liquid is HT, the external
water bath was kept at room temperature by continuously flushing RT water into the beaker. Thermocouples, 36-gauge and K-type, were used to measure the inlet and outlet liquid temperatures. Within each plenum, two thermocouples were inserted through the two plastic connecters into the plenum. One additional thermocouple was inserted through the stainless steel tube into the plenum, placed next to the entry/exit of the microchannel array [location B in Fig. 3-4(c)]. Thermocouple insertion into the tube was accomplished with the help of a T-fitting and then sealed with epoxy, as shown schematically in Fig. 3-4(b). To measure the solid wall temperature of the Cu MHE, two small holes were drilled symmetrically from two sides of the Cu device into the body center of the Cu coupon close to the microchannel, parallel to the plane of the microchannel array and perpendicular to the flow direction. Two thermocouples were inserted into these two holes and sealed with epoxy. The distances between the thermocouple tips to the microchannel array were ~1.5mm [location C in Fig. 3-4(c)]. The schematic cross-sectional view of the test section, together with thermocouple locations, is shown in Fig. 3-4(c).

To obtain more accurate liquid flow data, especially considering that the physical properties of EG varies significantly with temperature [12], it was chosen to put the liquid medium under conditions approximating isothermal as closely as possible during liquid flow testing. The Cu MHE prototype was placed in a room-temperature air environment. Room temperature pure water was flown through the device at different pressure drops, as well as pure EG under RT and HT inlet conditions. If a 10W/m²-K heat transfer coefficient $h$ is assumed [12] between room-temperature air and the Cu MHE body, a Biot number of $4.4\times10^{-4}$, defined as $Bi = \frac{h L_c}{k}$, where $k$ is the thermal conductivity of copper.
conductivity of water, will be obtained using the microchannel length as the characteristic length $L_c$. This small Biot number confirms the validity of the “isothermal” condition. Two small stainless steel tubes were inserted directly into the small through-holes in the fluid supply and drain plena of the Cu MHE and sealed with epoxy. The differential pressure across the liquid inlet and outlet was measured with a Dywer digital manometer connected to the two stainless steel tubes, with a measurement sensitivity of 0.1psi (690Pa). The total liquid flow across the microchannel array was obtained by measuring the volume of liquid collected at the exit end over a fixed period of time, immediately after liquid exit from the microchannel device. Liquid was collected into a measuring tube with ~5cm opening in less than five minutes. Weight measurements on static warm water contained within the measuring tube placed in the laboratory ambient showed a typical weight loss of 0.1g in five minutes. Typical volumetric measurements were performed on liquid volumes exceeding 300ml. The measurement error resulting from such weight loss is less than 0.03%, and therefore neglected. The characteristics of liquid flow across the microchannel array were quantified by measuring the total volumetric flow rate versus the pressure drop.

Some degree of nitrogen dissolution into the liquid occurs because of the use of pressurized nitrogen in the fluid supply section. No evidence of gas bubble was found during testing with EG. Occasional gas bubbles were observed during water flow at low flow rates. Thermocouple and pressure readings did not exhibit significant variations during testing once steady state was reached, suggesting that the level of outgassing did not significantly influence the test results.
In the data acquisition section, an Instrunet data acquisition system interfaced to a PC was used to collect the thermocouple readings. All thermocouples were calibrated with a two-point calibration, using a mixed ice/water bath and boiling water as the reference points. Temperature deviations at the two calibration temperatures were ±0.1°C in all cases.

3.4.2 Measurements

Heat transfer measurements were performed with the Cu MHE specimen immersed in the external water bath. Procedures and results of heat transfer measurements using pure water as the liquid medium have been reported previously [29]. Measurements using pure EG as the liquid medium were performed at different pressure drops and volumetric flow rates, which ranged respectively from 3.5×10⁴ to 4.2×10⁵Pa (5 to 60psi) and 0.07 to 0.47liter/min. With HT inlet EG liquid, the external water bath temperature was kept at room temperature via flushing of RT water. With RT inlet EG liquid, the external water bath temperature varied from 60 to 85°C depending on the liquid flow rate within the MHE specimen, because different amounts of heat were being removed through the liquid. The overall EG temperature range for the HT condition is from 58°C to 39°C, and from 24°C to 44°C for the RT condition. For each experimental condition, the external water bath temperature was uniform and constant throughout the experiment.

Liquid within the supply and drain plena undergoes further heating or cooling from the plena surfaces. The two thermocouples inserted through the plastic connectors [location A in Fig. 3-4(c)] are farther from the microchannel array, while the thermocouple inserted through the stainless steel tube [location B in Fig. 3-4(c)] was
placed the closest to the entrance/exit of the microchannel array. Consistently, it was observed that the readings from the B thermocouple to be 2-4°C higher or lower than the readings from the A thermocouples. The readings from the B thermocouples were therefore taken as the liquid inlet and outlet temperatures. In so doing, the difference between liquid outlet and inlet temperatures arise approximately solely from heating/cooling from surfaces of the microchannel array, thus eliminating heating/cooling contributions from the supply and drain plena surfaces. During testing with EG as the liquid medium, the difference between outlet and inlet liquid temperatures ranged from 9.5 to 17.3°C and 4.3 to 7.6°C for RT and HT fluid inlets, respectively. The mean liquid temperature $T_m$, defined as the arithmetic average of the EG outlet temperature $T_o$ and inlet temperature $T_i$, ranged from 30 to 35°C and 42 to 56°C for RT and HT fluid inlets, respectively.

It was difficult to measure the exact solid wall surface temperature, $T_s$, in the current experimental configuration. $T_s$ was therefore estimated from readings on C thermocouples [Fig. 3-4(c)], together with a correction based on an effective heat flux into the microchannel array. In all measurements, actual difference in C thermocouple readings is less than 1.1°C. Further it is believed that this difference in C thermocouple readings is due to a small difference in the actual distances from C thermocouple tips to the bottom or top surface of the microchannel array. $T_s$ was estimated by taking the average reading of the two C thermocouples minus $s q''/k$. In the last expression, $k$ is the thermal conductivity of copper, and $q''$ is an effective heat flux calculated by dividing the total power absorbed by the microchannel array by two times the area of the microchannel array. This effective heat flux was denoted positive when toward the
microchannel array. In the same expression, \( s \) is the average distance from the C thermocouples to the bottom or top surface of the microchannel array, taken as 1.5mm. In this way of estimating \( T_s \), systematic error due to a small difference in C thermocouple readings is compensated. The values of \( T_s \) so estimated ranged from 63 to 68°C and 32 to 43°C for RT and HT inlet liquid cases, respectively.

Table 3-2. Average microchannel dimensions of the assembled Cu MHE specimen, from which the heat transfer data were obtained.

<table>
<thead>
<tr>
<th>#channels</th>
<th>( H ) (( \mu )m)</th>
<th>( W ) (( \mu )m)</th>
<th>( L ) (( \mu )m)</th>
<th>( D_h ) (( \mu )m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>26</td>
<td>350±5</td>
<td>180±5</td>
<td>17470±50</td>
<td>238±5</td>
</tr>
</tbody>
</table>

Final examination of the assembled Cu MHE prototype was accomplished by mechanically cutting it in the direction perpendicular to the microchannel array, after completion of the heat transfer measurements. The cuts were placed within the drain plenum. To fully reveal the cross sectional configurations of the bonded microchannels, the cut piece containing the microchannel array was mechanically polished back close to the microchannel exits. Figure 3-5(a) shows a cross-sectional view of a portion of the bonded Cu microchannel structure. Water was passed from the liquid supply plenum through the microchannel arrays. Microscale water jets exiting the microchannels were observed on the drain plenum side to ensure that each microchannel has remained intact and unblocked during the bonding process. An image of microscale water jets exiting from a microchannel array in an assembled MHE was published previously [61]. The replicated Cu microchannel array contains 26 rectangular microchannels. Figure 3-5(b) shows a higher magnification view of one typical bonded microchannel in the Cu MHE device. To obtain the exact dimensions of bonded Cu microchannels, measurements of microchannel width \( W \) and height \( H \) were performed from micrographs similar to Fig. 3-
5(b) on each and every microchannel for the Cu MHE specimen. Measured microchannel widths and heights were averaged and shown in Table 3-2. Measurements of microchannel length were performed from SEM images obtained from the molded Cu coupon, after the fluid supply and drain plena were created by µEDM but before bonding to the blank Cu coupon. The hydraulic diameter of the microchannel, \( D_h \), is equal to

\[
4A_c / p \quad \text{with} \quad A_c = WH \quad \text{and} \quad p = 2(W + H)
\]

being respectively the microchannel cross-sectional area and perimeter. The average hydraulic diameter for the microchannel array was calculated from the average microchannel dimensions and shown also in Table 3-2.

Figure 3-5. Destructive examination of one Cu MHE: (a) cross-sectional view of a portion of the bonded microchannel structure; (b) higher magnification view of one bonded microchannel. Undulations resulted from the µEDM process.

The experimental uncertainty for each measured parameter was estimated. The uncertainties of microchannel dimension measurements, \( \delta H \), \( \delta W \), and \( \delta L \), are 5µm, 5µm, and 50µm, respectively. Volumetric liquid flow rate was measured by collecting a specific volume of liquid over a fixed time duration. The total volume of liquid collected was greater than 300ml in all cases and the uncertainty of volume reading is 2.5ml. Therefore the resulting uncertainty in measurements of the volumetric flow rate \( \hat{Q} \) is less than 0.8%. The uncertainty for each thermocouple reading is less than 0.25°C, somewhat
higher than that observed during thermocouple calibration. This uncertainty in temperature measurements was used to estimate uncertainties of quantities such as temperature difference between inlet and outlet, $T_o-T_i$, etc.

3.5 Results and Discussion

3.5.1 Data

In order to interpret raw experimental data, accurate values of various physical properties of pure water and pure EG as a function of temperature, such as density $\rho$, dynamic viscosity $\mu$, specific heat $c_p$, and thermal conductivity $k$, need to be known. For pure EG, property values at relevant temperatures were obtained through interpolating 2nd-5th order polynomial fits to data provided in Incropera and De Witt [12]. The fitted $\rho$ is within 1.8% as compared to the original, $\mu$ within 4.8%, $c_p$ within 0.04%, and $k$ within 0.1%. The water properties were obtained in a similar way, and all the fitting errors were within 1%.

3.5.2 Flow Characteristics and Friction Factor

In the devices configuration described earlier, the hydraulic diameters of supply and drain plena are at least one order of magnitude higher than those of the microchannels. The lengths of the flow passages within the supply/drain plena are about one fourth of the length of microchannels. These facts suggest a negligible plena flow resistance as compared to that through the microchannels. Therefore, uniform flow distribution should be expected among the microchannels.

The average velocity of liquid flowing through the microchannels was determined from the volumetric flow rate $\dot{Q}$, the total number of channels in the microchannel array $N_C$ and the channel cross-sectional area $A_c$,
$$V_{\text{avg}} = \frac{\dot{Q}}{N_c A_c} . \quad (3-1)$$

In the present case, $N_c = 26$. In calculating the Reynolds number $Re$ for liquid flowing through the microchannels,

$$Re = \frac{\rho V_{\text{avg}} D_h}{\mu} , \quad (3-2)$$

values for $\mu$ and $\rho$ were taken respectively as those at the mean liquid temperature $T_m = (T_o + T_i)/2$. The uncorrected Darcy friction factor was determined from

$$f_{\text{uncorr}} = \frac{\Delta P_{i-o}}{L \rho V_{\text{avg}}^2} , \quad (3-3)$$

where $\Delta P_{i-o}$ is the experimentally measured pressure drop across the microchannel array and the value of $\rho$ was taken at $T_m$ [66]. This measured pressure drop includes pressure losses when liquid enters the microchannels from the supply plenum and exits the microchannels into the drain plenum. A correction for this pressure drop is therefore necessary to obtain the true friction factor. However, the pressure loss coefficients associated with inlet into and exit from the microchannels depend significantly on the precise entrance/exit geometry, e.g., that shown in Fig. 3-5(b), which is difficult to characterize and not measured in detail. The precise values of these pressure losses are therefore unknown. The choice was therefore made to present measured values of $f_{\text{uncorr}}$, as calculated from Eq. (3-3), which quantifies the resistance to liquid flow through the microchannel array.
3.5.3 Heat Transfer Characteristics and Nusselt Number

Heat transfer characteristics for pure water flowing within Cu-based MHE devices have been reported previously [29]. In this chapter, therefore, the main focus is concentrated on the heat transfer performance of pure EG in Cu MHE devices under various testing conditions. The rate of total heat transfer from the solid Cu body to the liquid flowing through the microchannel array in the case of RT liquid inlet or from the liquid to the solid Cu body in the case of HT liquid inlet was calculated from

\[ \dot{q}_{\text{tot}} = \dot{m} c_p (T_o - T_i). \] (3-4)

The inlet liquid temperature \( T_i \) and outlet fluid temperature \( T_o \) were obtained respectively from the B thermocouples inserted through the stainless steel tubes into the small through-holes connecting to the supply and drain plena. The temperature readings from these thermocouples best represent the actual water temperature entering and exiting the microchannels. In Eq. (3-4), the liquid mass flow rate \( \dot{m} \) was calculated from measured \( \dot{Q} \) values, taking respectively the values of \( c_p \) and \( \rho \) as those corresponding to \( T_m \) in the experiment. The average heat transfer coefficient for a single microchannel was calculated from

\[ h_{\text{avg}} = \frac{\dot{q}_{\text{tot}}}{N_c} \frac{1}{A S \Delta T_{lm}}, \] (3-5)

where \( A_s \) is the surface area of one microchannel. In Eq. (3-5), the log-mean temperature difference is defined as

\[ \Delta T_{lm} = \frac{\Delta T_o - \Delta T_i}{\ln \left( \frac{\Delta T_o}{\Delta T_i} \right)}. \] (3-6)
where $\Delta T_o$ and $\Delta T_i$ are respectively $T_s - T_o$ and $T_s - T_i$, i.e., the differences between the temperature of the solid wall surface and the liquid outlet and inlet temperatures. The complete immersion of the Cu MHE specimen into the external water bath creates a testing condition that approximates a constant surface temperature boundary condition, under which the log-mean temperature difference is the appropriate one to use for the calculation of the average heat transfer coefficient [12]. If $T_s - T_m$ were used in Eq. (3-5) instead, $h_{avg}$ would be underestimated by 0.4 - 4% for the present test conditions. The average Nusselt number for a single microchannel was calculated from $h_{avg}$ as

$$Nu_{avg} = \frac{h_{avg} D_h}{k},$$  

(3-7)

where the value of the liquid thermal conductivity $k$ is taken as that corresponding to $T_m$. The average Nusselt number quantifies the heat transfer efficiency between the solid MHE body and the working liquid flowing through the microchannel array.

### 3.6 Analysis and Discussion

#### 3.6.1 Flow Characteristics and Friction Factor

Values of $f_{uncorr}$ for liquid flow through the Cu MHE specimen, whose dimensions are shown in Table 3-2, are plotted against the Reynolds number in Fig. 3-6. The entire data represent a composite of three sets of flow testing results with 1) RT pure water; 2) RT inlet pure EG; and 3) HT inlet pure EG. Because the dynamic viscosity of pure EG decreases significantly with increasing temperature, increasing the inlet EG temperature led to higher EG flow rate through the Cu MHE device, leading to higher $Re$ values as compared to what can be achieved using RT pure EG. The calculation of error
bars on $f_{uncorr}$ considered measurement uncertainties in microchannel dimensions $W$, $H$, and $L$, water volumetric flow rate $\dot{Q}$, and water pressure drop $\Delta P_{t-o}$, assuming all uncertainties are independent and uncorrelated. The maximum uncertainties for $Re$ and $f$ values extracted for the Cu MHE specimen were $\sim 4\%$ and $\sim 11\%$, respectively [67, 68]. It is evident from Fig. 3-6 that the entire $f_{uncorr}$ vs. $Re$ data exhibit one smooth trend, with measured friction factor decreases monotonically with increasing $Re$. Results of three sets of flow testing merge smoothly into one another, showing good agreements at points of overlap.

![Figure 3-6. Darcy friction factor versus Reynolds number for one Cu MHE. Laminar flow calculations based on fully developed flow and entrance length effects are shown as $f_{ld}$ and $f_{app}$, respectively. Haaland correlation [66] calculated for turbulent flow through smooth and rough pipes are also included.](image)

For fully developed laminar flow in smooth rectangular channels, an approximation to an analytically derived expression for the friction factor is given as
\[ f_{\text{lam,fd}} \ Re \approx 96 \left( 1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5 \right), \] (3-8)

where \( 0 < \alpha = W/H < 1 \) is the channel aspect ratio [69]. For the particular Cu MHE specimen, whose friction factor data are shown in Fig. 3-6, \( \alpha = 0.522 \). Thus in this case \( f_{\text{lam,fd}} \times \text{Re} \approx 61.65 \), according to Eq. (3-8).

Since the microchannels tested here may contain a considerable portion of their length that is in the hydrodynamically developing region for laminar flow, a correlation for an apparent friction factor that includes entrance effects in laminar flow was also used for comparison. Defining a dimensionless entrance length as \( x^+ = L/(D_h \text{Re}) \), an expression for an apparent friction factor \( f_{\text{app}} \), which takes entrance length effects into account, is given by Shah and London as [70]

\[
f_{\text{app}} \ Re = \left[ \left( \frac{3.2}{(x^+)^{0.57}} \right)^2 + \left( f_{\text{lam,fd}} \ Re \right)^2 \right]^{1/2}. \tag{3-9}
\]

In the limit of long channels with \( x^+ \) approaching infinity, \( f_{\text{app}} \) approaches \( f_{\text{lam,fd}} \) and is independent of \( L \). In the short channel limit, \( f_{\text{app}} \) is \( L \) dependent. Values of \( f_{\text{lam,fd}} \) and \( f_{\text{app}} \), as calculated from Eqs. (3-8) and (3-9) using parameters corresponding to the Cu MHE specimen in question, are shown in Fig. 3-6 as a function of the Reynolds number in the range \( 2 < Re < 800 \). It is evident from Fig. 3-6 that, within this \( Re \) range, little difference exists between \( f_{\text{lam,fd}} \) and \( f_{\text{app}} \), implying that entrance length effects do not significantly alter the friction factor for the present channel geometry. It is also evident that reasonable agreement exists between the calculated
laminar flow friction factors and measured values of $f_{uncorr}$ in the range $2 < Re < 350$, with the inference that flow within the microchannels in this range of Reynolds numbers is described reasonably well by laminar flow calculations.

At $Re \sim 2000$, the value of $f_{uncorr}$ is $\sim 0.06$. The Haaland correlations for the friction factor in fully developed turbulent flow within a tube [66]

$$\frac{1}{\sqrt{f}} = -1.8 \log \left( \frac{\varepsilon / D_h}{3.7} \right)^{1.11} + \frac{6.9}{Re}, \quad (3-10)$$

are also plotted in Fig. 3-6 for $Re > 1800$. In Eq. (3-10), $\varepsilon$ is the roughness parameter. Two cases are plotted, corresponding respectively to values of $\varepsilon$ being 0 and 5µm. The latter $\varepsilon$ value corresponds to the average value of $R_z$ measured from the microchannel bottom and sidewall surfaces of the present set of Cu MHE specimens replicated from the same mold insert. In the turbulent flow regime, the hydrodynamic entrance length effects on friction factor are neglected in this analysis. Values of $f_{uncorr}$, obtained from RT pure water flowing through the Cu MHE device, appear to be closer to the Haaland correlation values calculated for 5µm roughness, suggesting that the presence of surface roughness within the microchannels contributes to increases in friction factor in the turbulent flow regime. The present data on liquid flow in microchannels expand on the usual studies with water as the liquid medium, and are consistent with previous results reported in the literature [29, 68, 71].

3.6.2 Heat Transfer Characteristics and Nusselt Number

Values of the average Nusselt number, $Nu_{avg}$, calculated from Eqs. (3-5)-(3-7), are plotted against the Reynolds number in Fig. 3-7, for both RT and HT inlet EG cases.
Nusselt number values for pure water flowing in Cu-based microchannels were reported previously [7, 29]. The calculation of error bars on $N_{\text{u}_{\text{avg}}}$ considered measurement uncertainties on temperatures $T_s$, $T_o$, and $T_i$, EG volumetric flow rate $\dot{Q}$, and microchannel dimensions $W$, $H$, and $L$, again assuming all uncertainties are independent and uncorrelated [69, 70]. The maximum uncertainties for $Nu$ values extracted for the Cu MHE specimen were ~16% and ~21% for RT and HT inlet EG cases, respectively. Under both testing conditions, Fig. 3-7 shows that measured $Re$ numbers fall within the laminar flow regime. $Nu$ values for the RT inlet EG condition appear to increase at a faster rate with increasing Re as compared to the HT inlet EG case, leading to $Nu$ values from the RT inlet EG condition exceeding those from the HT inlet EG condition at $Re > 60$.

![Figure 3-7. Nusselt number vs. Reynolds number for pure EG flowing through the Cu MHE under RT EG inlet and HT EG inlet conditions.](image)
For laminar flow, estimates for the hydrodynamic entry length, $L_h$, and the thermal entrance length, $L_{th}$, are known: $L_h \approx 0.05D_h \text{Re}$ and $L_{th} \approx 0.05D_h \text{RePr}$ [12], where the Prandtl number, $Pr = c_p \mu / k$, characterizes the ratio between momentum diffusivity and thermal diffusivity. Based on the present testing conditions for pure EG, it is calculated that $L_h$ ranges from 0.3 to 2.1mm and $L_{th}$ ranges from 23.8 to 146mm. This means that the hydrodynamic entrance length covers up to 12% of the total length of the microchannel, while the thermal entrance length exceeds the total microchannel length under all testing conditions for EG. Because the dynamic viscosity of EG decreases significantly with increasing temperature, the Prandtl numbers for pure EG are significantly higher under the RT inlet test condition as compared to those under the HT inlet test condition. The increase in $Pr$ implies a more pronounced influence of thermal entrance length effects on overall heat transfer under the RT inlet EG testing condition which would result in higher Nu values in the RT inlet EG case, as shown in Fig. 3-7.

Known Nusselt number correlations are compared to the heat transfer data obtained using pure EG as the working fluid under each testing condition, as shown in Fig 3-8. For laminar flow, the Hausen correlation [12], valid for a constant surface temperature and thermally developing flow, is given by

$$Nu_{Hausen} = 3.66 + \frac{0.0668(D_h / L) \text{RePr}}{1 + 0.04[(D_h / L) \text{RePr}]^{2/3}},$$

(3-11)

The difference between the correlation for circular ducts attributed to Hausen [12] and the tabulated one for rectangular ducts [72] is within 6%. For the sake of simplicity, the
Hausen correlation is adopted here to compare with the present experimental observations.

Figure 3-8. (a) Nusselt number vs. Reynolds number for pure EG flowing through the Cu MHE under RT EG inlet condition. Calculated Sieder-Tate and Hausen correlation values are included. (b) Nusselt number vs. Reynolds number for pure EG flowing through the Cu MHE under HT EG inlet condition. Calculated Sieder-Tate and Hausen correlation values are included.

The laminar regime Sieder-Tate correlation [12], valid for a constant surface temperature and combined hydrodynamically and thermally developing flow, is given by

\[
Nu_{Sieder-Tate} = 1.86 \left( \frac{Re \Pr}{L/D_h} \right)^{1/3} \left( \frac{\mu}{\mu_s} \right)^{0.14}.
\]  

(3-12)

All quantities in Eqs. (3-11) and (3-12) are evaluated at the mean temperature \( T_m \) except the value of \( \mu_s \), which is evaluated at the solid wall temperature \( T_s \). Nusselt numbers calculated from the Hausen and Sieder-Tate correlations, as expressed in Eqs. (3-11) and (3-12), as well as the corresponding experimental data, are respectively plotted in Fig. 3-8(a) for the case of RT inlet EG and in Fig. 3-8(b) for the case of HT inlet EG. Figure 3-8(a) shows that calculated values for the Hausen and Sieder-Tate correlations, as well as the experimental \( Nu_{avg} \) values, increase with increasing \( Re \). The calculated correlation
values fall within the range of the experimental data. The experimental data exhibit higher rate of $Nu_{avg}$ increase with increasing $Re$. Figure 3-8(b) shows similar results for the case of HT inlet EG. The experimental $Nu_{avg}$ values and the corresponding Hausen and Sieder-Tate correlation values are again close, with the experimental data increasing at a faster rate with increasing $Re$.

Table 3-3. EG properties at inlet and outlet temperatures under RT and HT conditions.

<table>
<thead>
<tr>
<th>RT condition</th>
<th>HT condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>flow rate=0.47L/min</td>
<td>flow rate=0.40L/min</td>
</tr>
<tr>
<td>$\mu$ (10$^{-3}$Pa·s)</td>
<td>$\mu$ (10$^{-3}$Pa·s)</td>
</tr>
<tr>
<td>Pr</td>
<td>Pr</td>
</tr>
<tr>
<td>$T_i = 24.5^\circ$C</td>
<td>$T_i = 58.3^\circ$C</td>
</tr>
<tr>
<td>17.5</td>
<td>5.7</td>
</tr>
<tr>
<td>167.7</td>
<td>56.1</td>
</tr>
<tr>
<td>$T_o = 36.0^\circ$C</td>
<td>$T_o = 53.8^\circ$C</td>
</tr>
<tr>
<td>10.7</td>
<td>8.6</td>
</tr>
<tr>
<td>102.8</td>
<td>61.8</td>
</tr>
</tbody>
</table>

Several factors may contribute toward the observed differences between the experimental $Nu$ values and calculated correlation values. First, the correlations assume the condition of constant wall temperature $T_s$, which may be violated in the present case if heat conduction in the flow direction through the Cu channel walls cannot be neglected. Second, it should be noted that the physical properties of EG differ significantly under RT or HT testing conditions, as shown in Table 3-3. Variations in $\mu$ and $Pr$ values are much more pronounced under the RT inlet EG condition. The $Nu$ values in the HT inlet EG case, Fig. 3-8(b), appear to agree better with the correlation values, suggesting that significant variations in EG physical properties may contribute to the apparent differences between experimental and calculated correlation values. For example, error in the choice of the mean fluid temperature $T_m$ leads to selection of improper $\mu$ and $Pr$ values, which is exaggerated under the RT inlet EG condition.
It is of interest to compare cooling performance of EG vs. that of water. For the RT inlet EG case, at a flow rate of ~0.47liter/min, the rate of heat removal due to EG flowing within the microchannel array was ~245W, and the corresponding $h$ value was ~15700W/m²K. In comparison, at a water flow rate of ~0.40liter/min, the heat removal rate and $h$ value are respectively ~315W and ~51300W/m²K. Actual applications may demand the use of EG/water mixtures as the fluid medium, the heat transfer performance of which awaits to be quantified in the Cu-based MHE context.

3.7 Summary

All-Cu MHE prototypes were fabricated by combining direct molding replication and solid state bonding. Liquid flow and heat transfer testing of the Cu MHE prototypes was conducted with pure water and pure EG as the liquid medium. Good agreement between data on flow characteristics, as expressed in the dimensionless Darcy friction factor, was obtained from RT pure water and RT and HT pure EG. Experimentally measured average Nusselt numbers for pure EG in the laminar flow regime are found to be in general agreement with known laminar flow correlations. The observed differences in the average Nusselt number for EG under RT inlet and HT inlet testing conditions are rationalized by the influence of the Prandtl number on thermal entrance length effects, as higher Prandtl number under RT inlet condition enhances the thermal entrance length effect and therefore results in higher Nusselt numbers.

NOMENCLATURE

$A$ & Area (m$^2$) \\
$A_c$ & Microchannel cross-sectional area
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bi</td>
<td>Biot number</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat (J/kg·K)</td>
</tr>
<tr>
<td>$D_h$</td>
<td>Microchannel hydraulic diameter (µm)</td>
</tr>
<tr>
<td>$f$</td>
<td>Darcy friction factor</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient (W/m²·K)</td>
</tr>
<tr>
<td>$H$</td>
<td>Channel height (µm)</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity (W/m·K)</td>
</tr>
<tr>
<td>$L$</td>
<td>Channel length (µm)</td>
</tr>
<tr>
<td>$L_c$</td>
<td>Characteristic length (µm)</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate of liquid (kg/s)</td>
</tr>
<tr>
<td>$N_c$</td>
<td>Number of channels</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>$\Delta P$</td>
<td>Differential pressure (Pa)</td>
</tr>
<tr>
<td>$p$</td>
<td>Microchannel perimeter (µm)</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>Volumetric flow rate (m³/s)</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>Power (W)</td>
</tr>
<tr>
<td>$q''$</td>
<td>Effective heat flux (W/m²)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$Rz$</td>
<td>Peak-to-valley roughness (µm)</td>
</tr>
</tbody>
</table>
$s$ Distance of C thermocouple to microchannel

$T$ Temperature (°C)

$V$ Velocity of water (m/s)

$W$ Channel width (µm)

$x^+$ Dimensionless entrance length

**Greek Symbols**

$\alpha$ Microchannel aspect ratio

$\varepsilon$ Roughness parameter (µm)

$\rho$ Density (kg/m³)

$\mu$ Viscosity (N·s/m²)

**Subscripts**

$app$ Apparent

$avg$ Average

$fd$ Fully developed

$h$ Hydrodynamic

$i$ Inlet

$lam$ Laminar

$lm$ Log-mean

$m$ Mean

$o$ Outlet
<table>
<thead>
<tr>
<th>s</th>
<th>Surface</th>
</tr>
</thead>
<tbody>
<tr>
<td>th</td>
<td>Thermal</td>
</tr>
<tr>
<td>tot</td>
<td>Total</td>
</tr>
<tr>
<td>uncorr</td>
<td>Uncorrected</td>
</tr>
</tbody>
</table>
CHAPTER 4. QUANTIFICATION OF THERMAL RESISTANCE OF TRANSIENT-LIQUID-PHASE BONDED CU/AL/CU INTERFACES FOR ASSEMBLY OF CU-BASED MICROCHANNEL HEAT EXCHANGERS

4.1. Introduction

The transient liquid phase (TLP) bonding [40] approach has been used for making enclosed, Cu-based microchannel structures. The Cu base metal containing a parallel array of microchannels was TLP bonded at temperatures of ~600°C to a Cu sheet metal capping layer with a thin elemental Al foil as the intermediate bonding layer [35]. Figure 4-1 shows a cross-sectional scanning electron microscopy (SEM) image of a portion of one Cu-based MHE. As evidenced in Fig. 4-1, such a fabrication sequence is capable of producing MHEs of low profile, with total thicknesses of 1mm or below.

![Figure 4-1. A cross-sectional SE image of a portion of one Cu-based, low-profile MHE.](image)

Figure 4-1 shows the equilibrium Al-Cu phase diagram [73]. Joining a Cu base containing a microchannel array to a thin Cu capping layer through Cu/Al/Cu TLP bonding begins with interfacial melting when the eutectic temperature of 546°C is exceeded, followed by subsequent re-solidification and homogenization [35, 40]. These
liquid- and solid- phase processes associated with interfacial melting/re-solidification/homogenization produce a heterogeneous material layer surrounding the original bonding interface location, which may impact the heat transfer performance of the bonded MHE.

Figure 4-2. The equilibrium Al-Cu phase diagram.

Figure 4-3. Schematics of two different heat transfer scenarios from an external heat source to liquid flowing within a Cu MHE.
As one example, Fig. 4-3 illustrates schematically two different heat transfer scenarios from a high-temperature source external to a Cu MHE to liquid flowing through an array of rectangular microchannels within the MHE. In one case, heat flows from the capping layer side and passes through the bonding interface region before reaching the microchannel internal surfaces. In the other case, heat flows from the base side and reaches three of the four microchannel internal surfaces before encountering the bonding interface region. Preliminary heat transfer testing was conducted in these two scenarios, and a significant difference in heat transfer performance was noted. One possible rational for the measured performance difference lies in that, in one configuration, major heat flow within the solid needs to cross the TLP bonding interface region, which presents an additional thermal resistance as compared to bulk Cu.

As another example, MHEs containing multiple layers of microchannel arrays have the potential for increased heat removal capacity. Figure 4-4 shows a cross-sectional image of a portion of one Cu-based, double-layered, MHE, TLP bonded with Al intermediate layers. When MHEs are fabricated in configurations with microchannel...
arrays in multiple layers, heat flow necessarily crosses at least one TLP bonding interface region. Quantification of the thermal properties of the Cu/Al/Cu TLP bonding interface region is therefore necessary for modeling the heat transfer performance of such multilayered MHEs.

This chapter reports the result of a quantitative evaluation of the thermal resistance of bonding interface regions formed through TLP bonding of Cu with elemental Al intermediate layers. Cu/Al/Cu sandwich specimens formed by TLP bonding were prepared at different temperatures. Structure of the Cu/Al/Cu TLP bonding interface region changes with changing TLP bonding protocol, and evolves from being a single phase Cu-Al intermetallic compound, $\gamma_1$-Al$_4$Cu$_9$, to an eutectoid mixture of $\gamma_1$-Al$_4$Cu$_9$ and the Cu terminal solid solution, (Cu). In order to obtain the mass density and specific heat of $\gamma_1$-Al$_4$Cu$_9$, measurements were carried out on two bulk, single-phase, $\gamma_1$-Al$_4$Cu$_9$ samples. An effective thermal conductivity of the bonding interface region was probed with a transient flash method and obtained through data analysis using a three-layer-model [74, 75]. Overall thermal resistance of the bonding interface region was presented, based on measured effective interfacial thermal conductivity values. The present results provide guidance to the development of bonding protocols for Cu-based MHEs with optimized heat transfer performance.

4.2. Experimental Procedures

4.2.1 Sample Preparation

Figure 4-5 illustrates the process of Cu/Al/Cu test specimen preparation. Cu 110 (99.9+ wt. % Cu) coupons, ~26mm×~26mm by ~2mm in thickness, were polished with silicon carbide papers down to a grit size of 1200. A 38μm thick Al 1100 (99.99 wt. %
Al) foil was placed in between two polished Cu coupons. Then the Cu/Al/Cu assembly was heated up in vacuum to a bonding temperature under a compression force of ~100N, amounting to a pressure of ~0.15MPa. The Cu/Al/Cu assembly was held at the bonding temperature for 10min, and then allowed to cool to room temperature in ~4 hours. Ten individual Cu/Al/Cu specimen coupons were prepared at three different bonding temperatures of 560°C, 580°C, and 600°C, and sorted into three groups: group560, group580, and group600. A 12.7mm diameter cylindrical disc was cut from each bonded Cu/Al/Cu coupon using wire electro-discharging-machining (EDM) for thermal property measurements. Remaining portions of the specimen coupons were used for imaging of the TLP bonding interface regions as well as composition analysis.

![Diagram of TLP bonding process]

Figure 4-5. A schematic of the procedure for preparing TLP bonded Cu/Al/Cu test specimens.

### 4.2.2 Structural and Compositional Characterization of the TLP Bonded Interface Region

An FEI Quanta3D FEG Dual-Beam FIB instrument was employed to characterize the interface regions of TLP bonded Cu/Al/Cu specimens. The FIB instrument combined a Ga⁺ ion beam column and a Schottky field-emission electron beam column, both with acceleration voltages up to 30kV. Imaging within the FIB instrument can be obtained
with electron-induced secondary electrons (SEs) or ion-induced secondary electrons (ISEs). Crystalline grains are oriented differently with respect to the incident Ga\(^+\) ion beam direction, giving rise to different degree of Ga\(^+\) ion channeling. This difference in ion channeling in turn gives rise to contrast differences from different crystalline grains in ISE imaging [36]. The FIB instrument further incorporated an electron/ion beam catalyzed Pt deposition system using Pt-containing organometallic gaseous sources and an OmniProbe microprobe/manipulator for specimen attachment and lift-out.

A piece, \(\sim 1\text{mm} \times \sim 5\text{mm} \times \sim 4\text{mm}\) in size, was cut from each and every specimen coupon using a Struers Accutom-5 precision cutting machine. The cut surfaces were then mechanically polished to less than \(3\mu\text{m}\) in roughness, followed by chemical etching with a \(\sim 5\%\) iron nitrate solution. After chemical etching, the crystalline grain boundaries were clearly visible in an optical microscope. Specimen examination in the FIB instrument began with a light surface etching with a \(30\text{kV}\) Ga\(^+\) ion beam, followed by ISE imaging. Cross-sectional images of the bonding interface region were obtained from each and every sample in this manner.

Cross-sectional transmission electron microscopy (TEM) specimens were prepared in-situ the FIB instrument. Thin specimen slices from the bonding interface region were cut by Ga\(^+\) ion beam, lifted out through Pt attachment to the tip of an OmniProbe W microprobe, transferred to a special TEM specimen grid, and thinned to electron transparency by glancing angle Ga\(^+\) ion beams of decreasing voltages and currents [35].

Specimen characterization by TEM and X-ray energy dispersive spectroscopy (EDS) was carried out on a JEOL JEM2010 instrument operated at 200kV, equipped with
an EDAX EDS system employing a Super Ultra Thin Window Sapphire Si(Li) Detecting Unit with an active area of 30mm2. Electron images and selected area diffraction patterns (SADPs) obtained at various locations along bonding interface regions were obtained. To accurately index the diffraction patterns and obtain relevant lattice parameters, a series of SADPs from a single crystal Si specimen were obtained at the same operating voltage and the same nominal camera length, and compared to the SADPs to be analyzed. Spot mode compositional analysis at various locations within the bonding interface region was accomplished with the EDAX system. Analysis of EDS spectra was carried out with factory-supplied EDAX Genesis2000 software package [35].

4.2.3 Single-Phase $\gamma_1$-$\text{Al}_4\text{Cu}_9$ Bulk Specimen Preparation and Measurement

The presence of $\gamma_1$-$\text{Al}_4\text{Cu}_9$ was detected at the Cu/Al/Cu bonding interface region. To analyze flash data quantitatively, values for mass density, specific heat, and thermal conductivity of the $\gamma_1$ phase need to be known. Not able to locate such data in the literature, single-phase, bulk, Al$_4$Cu$_9$ specimens were prepared and measured.

Two bulk specimens were prepared, with total weights of 1.0g and 1.5g, respectively. Elemental Al (Alfa Aesar, 99.45% Al) and Cu (McMaster-Carr, 99.9% Cu) foils, were used as the starting materials. The elemental Al and Cu masses for one specimen were 0.159g Al and 0.841g Cu, and 0.239g Al and 1.261g Cu for the other specimen. These mass combinations corresponded to the Al/Cu atomic number ratio of 4/9. The elemental Al and Cu foils were shredded into small pieces, well mixed in quartz crucibles, and heated up in a vacuum furnace until melting. The 1.5g specimen was cooled slowly in-furnace, while the 1.0g specimen was quenched. The 1.0g quenched sample and the 1.5g slow-cooled sample will be designated as sample01 and sample02 in
this chapter, respectively. Subsequent annealing in a vacuum furnace was carried out on both samples at 550°C for 24hr. To confirm the final structure of the two samples, θ-20 X-ray diffraction (XRD) patterns were obtained from both samples on a Siemens D5000 instrument.

Measurements of mass density, ρ, of the two bulk specimens were carried out on an Ohaus Adventurer AR2140 Balance with readability of 0.0001g. First, the sample weights were measured on the balance. Then for the volume measurement, Archimedes principle was employed on the analytical balance: a glass beaker filled with water was placed on the balance; a sample attached to a filament was immersed into the water without touching the beaker; the weight change read from the balance is the weight of the water that has the same volume of the sample. This volume was then calculated from the weight change divided by the known water density. The ρ values of bulk γ1 samples were then obtained from sample weights and volumes. The typical error associated with mass density measurements is ~±1%.

Measurements of specific heat, $c_p$, and thermal conductivity, $k$, of the two bulk specimens were carried out on a Quantum Design Physical Property Measurement System (PPMS). For $c_p$ measurements, two cubic pieces, of ~1.5mm×1.5mm×1.5mm in size, were cut from the bulk samples. During the measurement, a known amount of heat was absorbed by the specimen, and its corresponding temperature rise was detected. For $k$ measurements, two long square bars, of ~1.7mm×1.7mm×7mm in size, were cut out from the bulk samples. In this measurement, a known heat flow was input along a square bar, and temperatures at two ends of the bar were then measured at steady state. Values of $k$ were calculated from values of input heat flow and measured temperatures, together
with specimen dimensions. Typical errors associated with \( c_p \) and \( k \) measurements on the PPMS instrument are \( \pm 2\% \) and \( \pm 5\% \), respectively.

### 4.2.4 Transient Flash Measurements

Principles of transient flash measurements have been described previously [74, 75, 76]. Flash measurements were carried out on a Netzsch LFA447 Nanoflash instrument. During the measurement, a transient light pulse, \( \sim 300\mu s \) in duration, was directed onto the front side surface of a cylindrical disk specimen, as shown schematically in Fig. 4-6(a). The temperature of its backside surface was measured with an infrared temperature detector and recorded as a function of time.

Flash measurements were first conducted with the Nanoflash instrument on a homogeneous cylindrical disc of Cu 110 (99.9+wt.%). In this case, the flash data were analyzed using a single-layer model [76]. Flash measurements on TLP bonded Cu/Al/Cu specimens were also carried out. Ten separate measurements were conducted on each TLP-bonded Cu/Al/Cu cylindrical disk specimen. Specimens from the three groups were all measured in the same manner. Recorded temperature-time data from flash...
measurements on TLP bonded Cu/Al/Cu specimens were analyzed with a three-layer model [74, 75], the software implementation of which was supplied as a part of the Nanoflash instrument.

### 4.2.5 Sample Annealing and Measurement

Four samples from group560 were further annealed at the temperature of 580°C in a vacuum furnace. During annealing, samples were rapidly heated up to 580°C, held for a certain amount of time, and then slowly cooled down in furnace. Heating up and cooling down took ~6min and ~2hr, respectively. Each sample was annealed three times: held at 580°C for 3 min, 2hr, and 48hr. After each annealing, interface region was further imaged, interlayer thickness was further measured, and additional interfacial thermal property measurements were carried out for each and every sample from group560, following the procedures described in sections 2.2 and 2.4.

### 4.3 Analysis and Results

#### 4.3.1 Imaging of TLP Bonded Interface Regions

Figure 4-7 shows typical ISE images of Cu/Al/Cu TLP bonding interface regions from samples of group560, group580, and group600. These images indicate that the two Cu coupons were well-bonded with neither gaps nor excessive voids present within the bonding interface region. Therefore, the effect of thermal contact resistance is ignored in subsequent analysis. Each sample group shows a different type of structure within the bonding interface region. Additional TEM imaging, SADP, and EDS studies showed that bonding interface regions from group560 samples consisted of only the $\gamma_1$-$\text{Al}_4\text{Cu}_9$ phase; that bonding interface regions from group600 samples contained a eutectoid mixture of $\gamma_1$ and the Cu terminal solid solution, (Cu); and that bonding interface regions from
group580 samples contained a layered structure, with one $\gamma_1$ layer sandwiched between two $\gamma_1/(Cu)$ eutectoid mixture layers. Further details of the TEM/SADP/EDS experiments were reported elsewhere [35]. The relevant layer thicknesses were measured from ISE images similar to those shown in Fig. 4-7 for each and every specimen.

![Figure 4-7](image_url)

Figure 4-7. Typical cross-sectional ISE images of bonding interface regions of TLP bonded Cu/Al/Cu specimens: (a) from group560, (b) from group580, and (c) from group600.

4.3.2 Measurements on Bulk $\gamma_1$-Al$_4$Cu$_9$ Samples

Figures 4-8(a) and 4-8(b) show respectively XRD patterns obtained from the two bulk Al$_4$Cu$_9$ specimens, sample01 and sample02, after annealing at 550°C. All the observed diffraction peaks were indexed to those derived from a single cubic structure, with lattice parameters of 8.70±0.06 Å and 8.68±0.03 Å for sample01 and sample02, respectively [77]. The measured lattice parameters are consistent with the literature value.
for that of the $\gamma_1$-Al$_4$Cu$_9$ phase, 8.707Å [78]. The XRD patterns shown in Fig. 4-8 indicate that the two bulk specimens consist of a single $\gamma_1$-Al$_4$Cu$_9$ phase.

![XRD patterns](image)

Figure 4-8. XRD patterns obtained from bulk $\gamma_1$ specimens after annealing at 550°C: (a) sample01; (b) sample02.

Table 4-1 lists the physical properties of sample01 and sample02, measured at 20°C. The measured mass density is consistent, to ~1%, with the bulk density of the $\gamma_1$-Al$_4$Cu$_9$ phase, 6.84g/cm$^3$, calculated from the known crystal structure and lattice parameter of $\gamma_1$ [78]. Values of $c_p$ obtained from sample01 and sample02 are consistent with each other to within the typical measurement error. The difference in $k$ values obtained from sample01 and sample02 appears to be larger than what would be expected from typical measurement errors, and may stem from differences in grain size and grain orientation between the two specimens. The average of $\rho$ and $c_p$ values for the $\gamma_1$-Al$_4$Cu$_9$
phase was also calculated and shown in Table 4-1. These average values were used in the subsequent analysis of flash measurements.

4.3.3 Transient Flash Measurements

Flash measurements were first conducted with the Nanoflash instrument on a homogeneous cylindrical disc of Cu 110 (99.9+wt.%), with a total thickness of 4.35mm. In this case, the flash data were analyzed using a single-layer model [76]. In this model, heat transfer occurs along the z-direction within the cylindrical disc in a one-dimensional (1D) adiabatic manner (see Fig. 4-6(a)). The equation governing the heat transfer process is the 1D thermal conduction equation,

\[
\frac{\partial^2 \theta(z,t)}{\partial z^2} = \frac{1}{\alpha} \frac{\partial \theta(z,t)}{\partial t},
\]

(4-1)

where \( \theta \) is the temperature rise and \( \alpha = k/(\rho c_p) \) is the thermal diffusivity. Taking the coordinate as shown in Fig. 4-6(a), the boundary condition at the front surface of the disc is

\[-k \frac{\partial \theta(z,t)}{\partial z} = w(t) \text{ at } z = 0,
\]

(4-2)

where \( w(t) \) is heat flux induced by the transient light pulse. The boundary condition at the back surface of the disc is

\[-k \frac{\partial \theta(z,t)}{\partial z} = 0 \text{ at } z = d.
\]

(4-3)

Figure 4-6(b) shows one typical raw data set obtained from the pure Cu disc. The measured disc back surface temperature change was recorded as a function of time. The back surface temperature rises from the initial value to a plateau value in about 50ms after the instant the light pulse was incident on the disc front surface, denoted by the
single-headed arrow in Fig. 4-6(b). The duration of the light pulse is ~300μs, which is negligible as compared to the temperature response time at the disc back surface. Therefore, \( w(t) \) can be approximated by a Dirac delta function response, \( w(t) = W \delta(t) \) where \( W \) is a constant. The initial condition for the present problem is

\[
\theta(z, t = 0) = 0. \tag{4-4}
\]

Solving the governing equation subject to the boundary and initial conditions, one obtains the following solution,

\[
V(t) = 1 + 2 \sum_{n=1}^{\infty} (-1)^n \exp \left( -\frac{\alpha n^2 \pi^2}{d^2} t \right), \tag{4-5}
\]

where \( V(t) \) is the normalized temperature rise at the disc back surface, \( V(t) = \theta(t) / \theta_{\text{max}} \), and \( \theta_{\text{max}} \) is the maximum temperature rise at the back surface when steady state is reached. When \( V = 0.5 \), Eq. (4-5) yields \( \alpha \pi^2 t_{1/2} / d^2 = 1.37 \), with the parameter \( t_{1/2} \) defined as the time lapse from the instant the light pulse was incident on the disc front surface to the moment when \( \theta \) reaches one half of \( \theta_{\text{max}} \). Therefore,

\[
\alpha = \frac{1.37 \pi^2}{d^2} t_{1/2} = 0.1388 d^2 / t_{1/2}. \tag{4-6}
\]

In actual measurements, values of \( t_{1/2} \) were extracted from raw temperature-time data, as indicated in Fig. 4-6(b). The value of \( \alpha \) was then obtained from Eq. (4-6). The value of thermal conductivity was then calculated from values of density and specific heat, together with that of thermal diffusivity: \( k = \rho c_p \alpha \). Ten separate light pulse measurements performed on the Cu disc yielded a thermal conductivity value of 401±3W/m-K at 20°C, using bulk density and specific heat values of \( c_p = 385 \text{J/kg-K} \) and \( \rho = 8.9 \times 10^3 \text{kg/m}^3 \). This measured thermal conductivity value compares favorably with the
literature value for $k$ of pure Cu, 401W/m-K [12], thereby confirming the correctness of the flash measurements.

Recorded temperature-time data from flash measurements on TLP bonded Cu/Al/Cu specimens were analyzed with a three-layer model [74, 75]. The total thickness of Cu/Al/Cu specimens was ~4mm in all cases. The specimen structure was set to consist of three layers: one pure Cu layer, followed by a TLP bonding interface region layer, and another pure Cu layer. Relevant physical properties of the two pure Cu layers were taken as their respective bulk values: mass density, $\rho=8.9\times10^3$kg/m$^3$; specific heat, $c_p=385$J/kg-K; and thermal conductivity $k=401$W/m-K [78]. Ignoring the internal structure of the bonding interface region and taking it as a homogeneous layer, its effective thermal conductivity can be calculated from the raw flash data analyzed according to the three-layer model, if values of layer thickness $t$, $\rho$, and $c_p$ for each of the three layers are known, together with $k$ values for the two pure Cu layers [74, 75].

In order to use the three-layer model to obtain the effective thermal conductivity of the TLP bonded interface region, effective values of specific heat and density of the bonding interlayer region need to be known. Referring to the equilibrium Al-Cu phase diagram shown in Fig. 4-2, the phases relevant to structural development within the Cu/Al/Cu TLP bonding interface region are the $\gamma_1$ phase, the (Cu) phase, and the high-temperature $\beta$ phase. The $\beta$ phase forms at the $\gamma_1/(\text{Cu})$ interfaces, and undergoes an eutectoid decomposition of $\beta \rightarrow \gamma_1 + (\text{Cu})$ as the specimen temperature decreases below the eutectoid temperature of 567°C. The eutectoid decomposition of $\beta$ is responsible for forming the eutectoid $\gamma_1/(\text{Cu})$ mixtures within the Cu/Al/Cu bonding interface region [35]. Assuming local thermodynamic equilibrium, the atomic composition of Cu in the $\gamma_1$
phase, $C_{\gamma_1}$, and the Cu composition in the (Cu) phase, $C_{(Cu)}$, were taken respectively to be those corresponding to the relevant $\gamma_1$ and (Cu) phase boundaries, 69% and 80% [73]. Likewise, the average Cu composition within the $\gamma_1/(Cu)$ eutectoid mixture, $C_{eu}$, is taken as the equilibrium eutectoid composition, 76%.

The specific heat $c_p$ of the Cu terminal solid solution has been measured previously [79]. The mass density of (Cu) was calculated based on previously published lattice parameter value $a_{(Cu)}=3.6650\text{Å}$ [80], the face-centered cubic lattice structure, and $C_{(Cu)}=80\%$, i.e.,

$$\rho_{(Cu)} = \frac{4(A_{Cu} \times C_{(Cu)} + A_{Al} \times (1-C_{(Cu)}))}{a_{(Cu)}^3 \times N_A}. \quad (4-7)$$

In Eq. (4-7), $A$ is the molar mass and $N_A$ is the Avogadro’s number. Since the eutectoid mixture consists of the $\gamma_1$ phase and the (Cu) phase, the phase percentages, $P$, in the eutectoid mixture were calculated using the lever rule based on the equilibrium Al-Cu phase diagram [73],

$$\begin{cases}
P_{\gamma_1} = \frac{C_{(Cu)} - C_{eu}}{C_{(Cu)} - C_{\gamma_1}} , \\
P_{(Cu)} = \frac{C_{eu} - C_{\gamma_1}}{C_{(Cu)} - C_{\gamma_1}} .
\end{cases} \quad (4-8)$$

With the calculated phase percentages $P_{\gamma_1}$ and $P_{(Cu)}$, the volume percentage, $V\%$, and mass percentage, $M\%$, of each phase in the eutectoid mixture can be obtained as well,

$$\begin{cases}
V\%_{\gamma_1} = \frac{P_{\gamma_1}O_{\gamma_1}}{P_{\gamma_1}O_{\gamma_1} + P_{(Cu)}O_{(Cu)}}, \\
V\%_{\gamma_1} + V\%_{(Cu)} = 1,
\end{cases} \quad (4-9)$$

and
\[
M\%_{\gamma_1} = \frac{V_{\%\gamma_1} \rho_{\gamma_1}}{V_{\%\gamma_1} \rho_{\gamma_1} + V_{\%(Cu)} \rho_{(Cu)}},
\]
\[
M\%_{\gamma_1} + M\%_{(Cu)} = 1
\]  

(4-10)

In Eq. (4-9), \(Q_{\gamma_1}\) and \(Q_{(Cu)}\) are the average volume per atom for \(\gamma_1\) and (Cu), calculated from the known crystal structures, lattice parameters, and compositions of \(\gamma_1\) and (Cu) [78, 80]. Using the known properties of \(\gamma_1\) and (Cu), the \(c_p\) and \(\rho\) values of the eutectoid mixture were then calculated using a linear rule of mixture assumption,

\[
\begin{align*}
\rho_{eu} &= V_{\%\gamma_1} \times \rho_{\gamma_1} + V_{\%(Cu)} \times \rho_{(Cu)} \\
c_{peu} &= M\%_{\gamma_1} \times c_{p\gamma_1} + M\%_{(Cu)} \times c_{p(Cu)}.
\end{align*}
\]

(4-11)

Table 4-2. Values of \(c_p\) and \(\rho\) at 20°C for the \(\gamma_1\) phase, the (Cu) phase, and eutectoid \(\gamma_1/(Cu)\) mixture.

<table>
<thead>
<tr>
<th></th>
<th>(\gamma_1)</th>
<th>(Cu)</th>
<th>Eutectoid mixture</th>
</tr>
</thead>
<tbody>
<tr>
<td>(c_p) (J/kg-K)</td>
<td>461</td>
<td>450</td>
<td>454</td>
</tr>
<tr>
<td>(\rho) (g/cm³)</td>
<td>6.92</td>
<td>7.60</td>
<td>7.34</td>
</tr>
</tbody>
</table>

Table 4-2 lists \(c_p\) and \(\rho\) values for the \(\gamma_1\) phase (presently measured), the (Cu) phase (previously known, [79, 80]), and the eutectoid \(\gamma_1/(Cu)\) mixture (calculated). For the three-layered eutectoid/\(\gamma_1\)/eutectoid bonding interface region, shown in the middle image in Fig. 4-7, the effective \(c_p\) and \(\rho\) values for the entire interlayer were calculated based on the thickness of each sub-layer,

\[
\begin{align*}
\rho_{interlayer} &= V_{\%\gamma_1} \times \rho_{\gamma_1} + V_{\%_{eu}} \times \rho_{eu} \\
M\%_{\gamma_1} &= \frac{V_{\%\gamma_1} \rho_{\gamma_1}}{V_{\%\gamma_1} \rho_{\gamma_1} + V_{\%_{eu}} \rho_{eu}} \\
M\%_{eu} &= 1 - M\%_{\gamma_1} \\
c_{pinterlayer} &= M\%_{\gamma_1} \times c_{p\gamma_1} + M\%_{eu} \times c_{peu}
\end{align*}
\]

(4-12)
Table 4-3 lists imaged structure thicknesses within the entire bonding interface region, together with effective $c_p$ and $\rho$ values, for all 10 Cu/Al/Cu samples from the three specimen groups.

Analysis of the data from room-temperature, 20°C, flash measurements with the three-layer model yields a value for the effective thermal diffusivity of the entire bonding interface region, $\alpha$. The effective thermal conductivity, $k$, was then calculated from $k=\alpha \rho c_p$, taking effective $\rho$ and $c_p$ values from Table 4-3. Figure 4-9 shows values of effective thermal conductivity, extracted from the four samples within group 560, which contain only the $\gamma_1$-$\text{Al}_4\text{Cu}_9$ phase within the entire bonding interface region. The extracted $k$ values therefore represent an indirect measurement of thermal conductivity of the $\gamma_1$ phase. Error bars on the four data points represent standard deviations from ten independent measurements on each specimen. Figure 4-9 shows significant scatter in extracted $k$ values, larger than what can be expected from random measurement errors, as
indicated by the error bars on the four data points. The observed scatter in $k$ values, obtained from samples within the same group, may result from two factors. First, thicknesses of bonding interface regions were measured through FIB cross sectioning at a specific location, and may deviate from the thickness averaged over the entire specimen. Second, even with nominally the same bonding protocol, the structure and composition of the bonding interface region may vary from sample to sample. As a comparison, $k$ values measured from the bulk $\gamma_1$ specimens, sample01 and sample02, are also shown in Fig. 4-9. Error bars on bulk thermal conductivity measurements are representative of the typical precision of the PPMS instrument. From Fig. 4-9, it is evident that a substantial overlap exists between thermal conductivity values directly measured from bulk $\gamma_1$ specimens and those deduced from flash measurements on bonding interface regions consisting of a single $\gamma_1$ phase. This agreement lends additional confidence to effective thermal conductivity values of the bonding interface region deduced from flash measurements conducted on TLP bonded Cu/Al/Cu specimens through data analysis according to the three-layer model.

Figure 4-9. $\gamma_1$ thermal conductivity values extracted from flash measurements on Cu/$\gamma_1$-interlayer/Cu specimens in comparison with those measured directly from bulk $\gamma_1$ specimens.
Figure 4-10. (a) Effective thermal conductivity values of Cu/Al/Cu bonding interface regions vs. the bonding temperature; (b) effective thermal resistance of bonding interface regions vs. the bonding temperature, calculated with $A=1\text{mm}^2$. The solid lines are trend lines connecting data averages.

Figure 4-10(a) shows effective thermal conductivity values deduced from flash measurements on all ten TLP bonded Cu/Al/Cu samples as a function of the bonding temperature. Significant scatter, beyond what can be expected from random measurement errors, again exists within the same sample group prepared with nominally the same bonding protocol. Even with this scatter, Fig. 4-10(a) suggests that effective $k$ value of the bonding interface region increases as the bonding temperature increases from $560^\circ\text{C}$ to $580^\circ\text{C}$ or $600^\circ\text{C}$. Effective thermal resistance of the bonding interface region is
the parameter that controls the influence of bonding on overall heat transfer performance. The effective thermal resistance, \( \eta \), is calculated from \( \eta = t/kA \). In this expression, \( t \) is the thickness of the bonding interface region and \( A \) is a unit area. Figure 4-10(b) shows \( \eta \) values in units of K/W, calculated for \( A=1\text{mm}^2 \), for bonding interface regions of various Cu/Al/Cu samples formed at different bonding temperatures.

4.3.4 Influence of Annealing on TLP Bonded Interfacial Layer

![Cross-sectional ISE images of bonding interface regions](image)

Figure 4-11. Cross-sectional ISE images of bonding interface regions of one group sample after a series of annealing at 580°C for different durations: (a) as-bonded; (b) 3min; (c) 2hr; (d) 48hr.

The four Cu/Al/Cu samples from group sample were further annealed in vacuum at 580°C. Each sample underwent a series of annealing treatments followed by subsequent measurements, as described in Section 2.5. Figure 4-11 shows four cross-sectional
images of the bonding interface region, obtained from one sample (560-2) after a series of annealing. As the annealing process proceeds, the original bonding interface region, which consists of only the $\gamma_1$-Al$_4$Cu$_9$ phase (Fig. 4-11(a)), evolves into a three-layered eutectoid/$\gamma_1$/eutectoid structure with the eutectoid reaction and the subsequent eutectoid decomposition proceeding from the original $\gamma_1$/Cu interface locations (Fig. 4-11(b), annealing 580°C 3min). The eutectoid layers thicken at the expense of the $\gamma_1$ phase, which is eventually consumed (Fig. 4-11(c), annealing 580°C 2hrs). The $\gamma_1$/Cu eutectoid mixture eventually transforms to the Cu terminal solid solution (Fig. 4-11(d), annealing 580°C 48hrs). Structure thicknesses within the bonding interface region were measured from cross-sectional images, such as those shown in Fig. 4-11, on all annealed specimens. Flash measurements were conducted on annealed specimens as well. Figure 4-12(a) shows values of effective thermal conductivity of the bonding interface region, deduced from flash measurements, against the annealing time at 580°C. Data in Fig. 4-12(a) show that, as the bonding interface region undergoes the structural transformation from the $\gamma_1$ phase to the (Cu) phase, its effective thermal conductivity increases significantly. Data in Fig. 4-12(b) show the influence of annealing on the effective thermal resistance of the bonding interface region, again calculated for $A=1\text{mm}^2$. Despite the increase in thermal conductivity, the effective thermal resistance appears to exhibit a slight increase. This is because the increase in effective thermal conductivity is negated by a concomitant increase in the overall thickness of the bonding interface region due to annealing.
Figure 4-12. Thermal properties of bonding interface regions of group560 samples vs. annealing time at 580°C: (a) effective thermal conductivity values. The solid lines connect data points obtained from the same sample; (b) effective thermal resistance calculated with $A=1\text{mm}^2$. The solid line is a trend line connecting data averages.
4.4. Discussion

Values of the effective thermal conductivity of as-bonded Cu/Al/Cu interface regions, deduced from flash measurements through the three-layer model together with measured and calculated properties of the $\gamma_1$-Al$_4$Cu$_9$ and (Cu) phases, ranged from 1/5 to 1/10 of that for pure Cu. The present measurements therefore indicate that the Cu/Al/Cu TLP bonding interface region constitutes a significant barrier to heat flow, and therefore impairs the performance of Cu MHEs TLP-bonded with elemental Al intermediate layers. As shown in Fig. 4-10(a), higher bonding temperatures increase the effective thermal conductivity of the bonding interface region. A counter-balancing effect from increasing the bonding temperature, however, is the increase of the total thickness of the bonding interface region. In terms of heat transfer performance, the present results shown in Fig. 4-10(b) indicate that, overall, higher bonding temperatures do indeed reduce the thermal resistance of the bonding interface. In Fig. 4-10(b), one sample (600-3) shows much lower effective interfacial thermal resistance, ~0.5K/W, due to the fact that this particular sample had a much lower total thickness of the bonding interface region as compared to other samples in group600, as shown in Table 4-3.

From the cross-sectional images of the bonding interface region, shown in Fig. 4-11, it is evident that the following structural evolution of the bonding interface region took place during the annealing process: $\gamma_1$ → three-layer eutectoid/$\gamma_1$/eutectoid → $\gamma_1$/(Cu) eutectoid mixture → (Cu). In addition, a longer annealing time yielded a wider bonding interfacial region. Figure 4-12(a) indicates that annealing in general increases the effective thermal conductivity of the bonding interface region, as the thermal conductivity of (Cu) roughly doubles that of $\gamma_1$. The trade-off for annealing comes in the
increase in total thickness of the bonding interface region. This trade-off leads to an overall unfavorable effect due to annealing, in that it leads to a slight increase in the effective thermal resistance of the bonding interface region, as indicated in Fig. 4-12(b). The present data suggest that decreasing the total thickness of the bonding interface region will lead to improvements in the thermal resistance of the bonding interface.

### 4.5 Concluding Remarks

A quantitative evaluation of the thermal resistance of TLP bonded Cu/Al/Cu interface regions has been carried out. The structural development and evolution of TLP bonded Cu/Al/Cu interface regions have been studied through site-selective cross-sectional imaging, as well as structure and composition characterization with TEM/SADP/EDS. The effective thermal conductivity of bonding interface regions has been deduced from transient flash measurements on TLP bonded Cu/Al/Cu sandwich specimens, in combination with measurements on bulk Cu-Al intermetallic compound specimens.

Values of effective thermal conductivity of the single-phase $\gamma_1\text{-Al}_4\text{Cu}_9$ layer were deduced, and shown to be consistent with thermal conductivity measurements on bulk $\gamma_1\text{-Al}_4\text{Cu}_9$ specimens. Values of effective thermal conductivities of as-bonded Cu/Al/Cu interface regions were obtained as a function of the bonding temperature. Together with cross-sectional imaging, an effective thermal resistance of the bonding interface region is calculated. It is observed that higher bonding temperatures reduce this interfacial thermal resistance.

Additional annealing of as-bonded Cu/Al/Cu specimens was conducted. While annealing increases the effective thermal conductivity of the bonding interface region, it
does not reduce its overall thermal resistance. To further optimize the heat transfer performance of TLP bonded MHEs without sacrificing their mechanical robustness, future efforts should be directed toward reducing the total thickness of the bonding interface region while keeping the same structural integrity.
CHAPTER 5. EXPERIMENTAL INVESTIGATION OF CU-BASED, DOUBLE-LAYERED, MICROCHANNEL HEAT EXCHANGERS

5.1. Introduction

As demonstrated in chapter 2, building low-profile Cu-based MHEs was achieved, and those MHEs were characterized in terms of heat transfer performance. In this chapter, another layer of microchannel array is added to an MHE. Testings of heat transfer and pressure drop are performed and compared to another single-layered MHEs which serves as a benchmark.

Figure 5-1. Cross-sectional SEM images of a portion of (a) one Cu-based, single-layered MHE and (b) another Cu-based, double-layered MHE.

(a)

(b)
Figure 5-1(a) illustrates a Cu-based MHE in which an array of parallel microchannels is contained in one plane, in a single-layered configuration. Figure 5-1(b) shows another Cu-based MHE in which two arrays of parallel microchannels are located in two separate but parallel planes, in a double-layered configuration.

The main reason why liquid flow within microchannels delivers high heat transfer performance is because the heat transfer rate between the internal microchannel surfaces and the working liquid increases as the size of the channel cross section decreases. Smaller sized channels lead to higher heat transfer coefficient, $h$, which characterizes the rate of liquid-solid convective heat transfer. In fact, $h$ is inverse proportional to the size of a channel, as demonstrated in the following equation [12],

$$h = k N_u / D_h .$$

In Eq. (5-1), the channel hydraulic diameter, $D_h$, is defined as $D_h = 4 A_c / P$ where $A_c$ and $P$ are respectively the area and perimeter of the channel cross section. For a circular channel, $D_h$ is equal to the channel diameter. The parameters $k$ and $N_u$ are respectively is thermal conductivity of the working liquid and the Nusselt number. The dimensionless parameter $N_u$ can be interpreted as a ratio between heat convection and heat conduction that take place within the working liquid. In a circular tube under fully developed laminar flow conditions, $N_u = 3.66$ if the tube inner surface temperature is held constant, and $N_u = 4.36$ if a uniform surface heat flux is provided [12].

Counter balancing the increased heat transfer rate, smaller channels carry severe penalties in terms of an increased resistance to liquid flow. Again taking a circular channel as an example, under fully developed laminar flow conditions, the liquid pressure drop, $\Delta P$, along a channel of length $L$ is expressed as [12]
\[ \Delta P = \frac{128\mu \dot{V}L}{\pi D^4}. \] 

(5-2)

In Eq. (5-2), \( \mu \) is the liquid viscosity and \( \dot{V} \) is the volumetric liquid flow rate through the circular channel of diameter \( D \). It is noted that the pressure penalty is related to the channel diameter to the fourth power provided that other conditions remain the same.

The requirement of larger and more powerful pumps often poses a hindrance to adoption of liquid flow within microchannels for higher heat transfer performance. The increased resistance to liquid flow is especially limiting in applications with pumping capacity restrictions. This factor has been a significant roadblock that prevents more wide-spread usage of MHEs.

Using Cu-based double-layered MHEs as an example, this chapter shows that multiple-layered MHEs can be built and offer a potential solution to the microchannel flow resistance problem mentioned above. As a comparison, Cu-based, single-layered MHEs with nominally the same microchannel dimensions were also fabricated. Liquid flow and heat transfer measurements were conducted on both types of MHEs. The results show that double-layered MHEs reduce, by more than half, the liquid pressure drop of single-layered MHEs while keeping the device-level heat transfer performance largely unchanged.

Additional applications for double-layered MHEs can be conceived. As a demonstration, a Cu-based, double-layered, liquid-liquid counter-flow, MHE prototype was built and tested. During testing, hot water and cold water were fed separately through the two different layers of the microchannel device. Heat transfer occurred between the two liquids flowing through microchannel arrays on separate layers. This
experiment demonstrates multiple-layered MHEs as potential candidates for compact, high-performance, liquid-liquid counter-flow heat exchangers.

5.2. Experimental Procedures

5.2.1 MHE Fabrication

Double-layered MHEs were fabricated from Cu 110 (99.9% wt. Cu) sheets. The fabrication starts with three Cu sheet metal pieces with the same footprint of 41.8mm×41.8mm but different thicknesses. A thin sheet, 0.25mm in thickness, was sandwiched between two 3.2mm-thick Cu sheets. A microchannel array with an active area of 41.8mm×28mm was cut by sinking-mode micro-electrical-discharge-machining (μEDM) onto one surface of each 3.2mm-thick Cu sheet. Prior to μEDM, the 3.2mm-thick Cu sheets were polished with silicon carbide papers of decreasing grit sizes down to 800. During the μEDM process, a carbon steel blade with a thickness of ~180μm was used as an electrode to perform 46 parallel and periodic cuts across the Cu sheet, resulting in an open and parallel microchannel array with a pitch of ~600μm.

To form an enclosed microchannel device, transient liquid phase (TLP) bonding [40] of the Cu sheets was achieved using elemental Al thin foil intermediate bonding layers. Before bonding, all faying surfaces were polished with SiC abrasive papers down to 800 grit size, followed by a surface etch using a 5% HCl solution. The two 3.2mm-thick Cu sheets were placed with the two open microchannel arrays facing each other, with the 0.25mm-thick sheet sandwiched in between. An elemental Al foil (25μm in thickness, 99 at.% Al) was placed at each interface between the 3.2mm-thick Cu sheet and the 0.25mm-thick sheet. Bonding was carried out on an MTS858 single-axis hydraulic testing system which was interfaced to a vacuum chamber. An upper heating
station was attached to the hydraulic actuator while a lower heating station was placed at the bottom of the vacuum chamber. During the bonding process, the chamber was evacuated to \(~1\times10^{-5}\)Torr, with both heating stations held at the same temperature. The entire Cu/Al/Cu/Al/Cu specimen assembly was placed in between the two heating stations, heated to ~600°C, and held for ~10min under an applied pressure of ~3MPa. Further descriptions of the Cu/Al/Cu TLP bonding process have been given elsewhere [35]. An illustration of double-layered, Cu-based MHEs is given in Figure 5-2(a), which shows a cross-sectional scanning electron microscopy (SEM) image of a portion of the two parallel microchannel arrays contained in two separate and parallel planes. After bonding, the Cu MHE was further connected to liquid supply/drain tubes using Sn soldering, as shown in Fig. 5-2(b).

![Figure 5-2. The Cu-based, double-layered MHE: (a) a SEM image of a portion of the cross section, (b) the assembled testing module with connection fittings. The MHE was Sn-soldered to the supply/drain tubes.](image)

The single-layered, Cu-based MHE was fabricated and assembled in a similar manner. It consists of two Cu sheets with dimensions of 41.8mm×41.8mm×3.2mm. An open and parallel microchannel array, with a total channel number of 46, was cut onto a
surface of one Cu sheet, while the other Cu sheet served as the mating cover. For TLP-bonding, an elemental Al foil (25μm in thickness, 99 at.% Al) was inserted between the two Cu sheets. The TLP bonding and subsequent connection to liquid supply/drain tubes were carried out following the same procedure described for double-layered MHEs.

![Diagram of MHE](image)

Figure 5-3. The Cu-based, liquid-to-liquid, counter-flow MHE: (a) a cross-sectional schematic, (b) the assembled device with fittings.

The double-layered, liquid-liquid counter-flow MHE was built differently. As shown schematically in Fig. 5-3(a), it consists of three Cu coupons with the same footprint of 56mm×60mm but different thicknesses. A middle coupon, of 1.6mm in thickness, was sandwiched by two 9.5mm-thick coupons. An open and parallel microchannel array with an active area of 36mm×46mm was cut by sinking-mode μEDM onto each side of the middle coupon. The dimensions of each microchannel were 46mm in length, ~210μm in width, and ~650μm depth. The channel-to-channel pitch was ~600μm, with 60 parallel microchannels incorporated within the active area on each side. Two plena were machined onto each 9.5mm-thick coupon to accommodate liquid flow in and out of the microchannel arrays. A threaded hole was drilled into each plenum and connected with a Swagelock® fitting. An elemental Al foil (38μm in thickness, 99 at.%
Al) was placed at each coupon interface, and the entire Cu/Al/Cu/Al/Cu specimen assembly was TLP-bonded following the procedure described above. After bonding, the effective length of the microchannels in use was 40mm, instead of the full channel length of 46mm, because of a 3mm overlap between the microchannel array and the plenum at each end. Figure 5-3(b) shows the assembled device. To ensure zero liquid leakage in this connection scheme, the coupon thickness of 9.5mm was chosen to give the threaded-hole connections a sufficient length of thread engagement.

5.2.2 Liquid Flow and Heat Transfer Measurements

5.2.2.1 Liquid Flow Measurements

Figure 5-4. A schematic of experimental setup for the pressure measurement.

Measurements of pressure drop as liquid flows through the MHE device were conducted separately from heat transfer testing. The experimental setup for the pressure drop measurements is shown schematically in Fig. 5-4. A diaphragm pump with a flow dampener was used to pump room-temperature tap water into the MHE. The flow dampener acts to smooth out the flow pulse generated by the pump. A Dywer® digital manometer with a measurement sensitivity of 0.01psi (69Pa) was connected to the MHE with two three-way fittings at the inlet and outlet of the MHE. The volumetric water flow rate through the MHE was obtained by measuring the volume of water collected at the exit over a fixed period of time. Collected volume was measured in a 500mL
measuring cylinder. During testing, the water flow rate was increased step by step and then decreased in the same manner, with all data points recorded at steady state.

5.2.2.2 Heat Transfer Testing of Single- and Double-Layered MHEs

Figure 5-5. A schematic of experimental setup for the heat transfer testing on the single- and double-layered MHEs.

A schematic of the experimental setup used for heat transfer testing of single- and double-layered MHEs is shown in Fig. 5-5. A diaphragm pump supplied water from a storage tank and fed it through a flow damper to the MHE. Water flow rates were measured in the same manner described in Section 2.2.1. The entire MHE was immersed in an external water bath, which was contained in a 2 liter glass beaker and placed on top of a hot plate. During testing, the hot plate heated the beaker and the water bath, which was continuously stirred to ensure temperature uniformity within the bath and enhance heat transfer between the MHE and the water bath. The water bath provided a stable temperature environment surrounding the entire Cu MHE, with bath temperature ranging from 55°C to 71°C during testing. This elevated temperature water bath served as the heat source for the MHE while room temperature water flowed through the microchannel arrays within the MHE.
Thermocouples (K-type, Omega Engineering Inc.) were used for all temperature measurements. Figures 5-6(a) and 5-6(b) show locations of thermocouple placement within the MHE device. Thermocouple was inserted into each plenum for measuring the liquid inlet temperature and outlet temperature, $T_{in}$ and $T_{out}$. To measure the solid wall temperatures, $T_{s1}$ and $T_{s2}$, a 0.8mm-diameter hole was drilled into each of the two 3.2mm-thick Cu sheets, parallel to the microchannel planes. Two additional thermocouples were inserted into the drilled holes on the 3.2mm-thick Cu sheets, and placed in the center of the microchannel array. In the ideal case, identical readings of $T_{s1}$ and $T_{s2}$ from these two thermocouples should be obtained. In reality, a slight difference in readings from these two thermocouples was observed, typically ~0.2-0.4°C. Thus, the solid wall temperature $T_s$ was taken as an arithmetic average of $T_{s1}$ and $T_{s2}$. For data acquisition, an HP34970 data acquisition unit interfaced to a computer was used to record the temperature readings from the thermocouples.

![Figure 5-6](image_url)  
Figure 5-6. A schematic of thermocouple positions for the single- and double- layered MHEs.
All data were recorded at steady state, including temperature readings and water flow rates. The volumetric water flow rates through MHEs were obtained by measuring the volume of water collected at the exit over a 1min period. During this time period, temperatures readings were recorded once every 5sec and used later in data analysis. The final temperature reading was obtained by averaging these 12 data points.

5.2.2.3 Heat Transfer Testing of Double-Layered, Liquid-Liquid Counter-Flow MHE

During testing, a steady flow of hot water was fed through one layer of the MHE device while cool water was fed through the other layer. Measurements were conducted at a series of water flow rates, which were measured in the same manner described in section 2.2.1. A K-type thermocouple was placed into each inlet and outlet for measuring the inlet and outlet temperatures for cool water and hot water, Tc,in, Tc,out, Th,in and Th,out. All temperature readings were recorded at steady state using the data acquisition instrument described in Section 2.2.2.

5.3. Data Analysis

The average microchannel hydraulic diameter of each MHE is defined as $D_h=\frac{4A_c}{p}$, where $A_c$ and $p$ are respectively the microchannel cross-sectional area and perimeter. Both parameters were measured from SEM images similar to that shown in Figs. 5-1 and 5-2, using the software ImageJ [38]. Imaging and dimensional measurements were performed on each and every microchannel, and average values of $A_c$ and $p$ were used in subsequent calculations.

The average velocity of water flowing through the MHE was obtained from the measured volumetric water flow rate, $\dot{V}$, together with the total number of channels in the array, $N_c$, and the microchannel cross-sectional area, $A_c$, according to
\[ V_{avg} = \frac{\psi}{N_c A_c}. \]  

(5-3)

The Reynolds number for water flow was then calculated,

\[ Re = \frac{\rho D_H V_{avg}}{\mu}, \]  

(5-4)

where \( \rho \) and \( \mu \) are respectively the water density and viscosity, evaluated at the mean water temperature, \( T_m = (T_{in} + T_{out})/2 \). These physical properties of water were obtained from tabulated data [12] through 3rd order polynomial or exponential curve-fitting. Compared to the original data [12], the maximum fitting errors for \( \rho \) and \( \mu \) are less than 0.01\% and 0.5\%, respectively.

The total heat exchange rate, \( \dot{q} \), was calculated from the temperature change of water going through the MHE,

\[ \dot{q} = \rho c_p \dot{V}(T_{out} - T_{in}). \]  

(5-5)

In Eq. (5-5), \( c_p \) is specific heat of water at \( T_m \), the value of which was again obtained through curve fitting to original data [12] with a maximum error of 0.004\%. The input admittance \( Y \), a parameter that quantifies the device-level heat transfer performance, is given by

\[ Y = \frac{\dot{q}}{T_{max} - T_{min}}, \]  

(5-6)

where \( T_{max} \) and \( T_{min} \) are respectively the highest and lowest temperature of the MHE device. For single- and double-layered MHEs, \( T_{max} = T_s \) and \( T_{min} = T_{in} \), where \( T_s = (T_{s1} + T_{s2})/2 \) is the average solid wall temperature. For the double-layered, liquid-liquid counter-flow MHE, \( T_{max} = T_{h,in} \) and \( T_{min} = T_{c,in} \).
The microchannel heat transfer coefficient, \( h \), characterizes convective heat transfer between the microchannel wet surface and the water flowing through it. \( h \) is defined as

\[
h = \frac{\dot{q}}{A_w(T_s-T_m)},
\]

where \( A_w \) is the microchannel wet area, calculated from the cross-sectional perimeter of the microchannel \( p \), the channel length \( L \), and the total number of channels \( N_c \), \( A_w = p \times L \times N_c \). The Nusselt number, \( Nu \), is a non-dimensional parameter that quantifies the ratio between heat convection and heat conduction that take place within the liquid. It is defined as

\[
Nu = \frac{hD_h}{k},
\]

where \( k \) is the thermal conductivity of the working liquid. Its value was again curve-fitted from original data [12] with a maximum error of 0.04%.

Table 5-1. Estimated measurement uncertainties for experimental parameters.

<table>
<thead>
<tr>
<th>( \delta P ) (psi)</th>
<th>( \delta T ) (°C)</th>
<th>( \delta \dot{V} ) (mL/min)</th>
<th>( \delta W ) (μm)</th>
<th>( \delta H ) (μm)</th>
<th>( \delta L ) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>±0.1</td>
<td>±0.1</td>
<td>±2.5</td>
<td>±5</td>
<td>±6</td>
<td>±0.2</td>
</tr>
</tbody>
</table>

Table 5-1 shows estimated uncertainty for each parameter measured in the experiments. The pressure uncertainty of ±0.1psi represents typical pressure fluctuations read during testing, while the measurement sensitivity of the manometer is much higher, of 0.01psi. The uncertainties for all temperature measurements were estimated to be ±0.1°C. Volumetric flow rates were measured by collecting the volume of water over a 1min period. Collected volumes were greater than 360mL in all cases, and the accuracy of the measuring cylinder was ±2.5mL. For the microchannel dimensions, the
measurement uncertainties for channel width $W$, channel height $H$ and channel length $L$ were ±5μm, ±6μm, and ±0.2mm, respectively. The error bars for all quantities, $y$, derived from measured quantities or curve-fitted properties, $x_i$, e.g., those shown in Eqs. (5-3)-(5-8), were deduced from the estimated uncertainties assuming all individual measurements are independent and uncorrelated [67],

$$
\delta y = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial y}{\partial x_i} \right)^2 (\delta x_i)^2}.
$$

(5-9)

5.4. Results

Figure 5-7 compares results of water pressure drop across the single- and double-layered MHEs as a function of the water flow rate. In Fig. 5-7(a), values of water pressure drop across the two MHEs are plotted against the rate of total water flow through the MHE devices. The liquid pressure drop increases with increasing flow rate. The pressure drop increases faster at higher flow rates, because higher flow rates introduce more prominent entrance effect into the liquid flow [68, 70].

Due to the configurations of the single- and double-layered MHEs, the flow rate per channel of the single-layered MHE is twice that of the double-layered MHE when the total flow rate is the same. According to Eq. (5-2), this would lead to the pressure drop for the single-layered MHE being twice that for the double-layered device when the comparison is made at the same total flow rate. From Fig. 5-7(a), it is noted that the pressure drop of single-layered MHE exceeds twice that for the double-layered MHE, especially at higher flow rates. The latter observation is attributable to the added resistance due to the more prominent entrance effect at higher flow rates. In Fig. 5-7(b), values of water pressure drop across the two MHEs are re-plotted against the flow rate per microchannel or Reynolds number. These Reynolds numbers were calculated based
on average dimensions of the two MHEs and water properties at 20 °C. The two data sets, obtained from the single- and double-layered MHE, overlap each other. Because both MHE devices possess identical nominal microchannel dimensions, the same pressure drop is observed at the same flow rate per channel. The overlap of the two data sets verifies the correctness of the pressure drop measurements.

Figure 5-7. Data on liquid flow characteristics for single- and double-layered MHEs: (a) pressure drop versus total flow rate through device, (b) pressure drop versus flow rate through single channel.
Figure 5-8 shows Nusselt number, \( Nu \), vs. Reynolds number, \( Re \), for the single- and double-layered MHE. This can be interpreted as normalized heat transfer performance per channel at different values of \( Re \). It is evident from Fig. 5-8 that \( Nu \) increases with increasing \( Re \) in an approximately linear fashion. Values of \( Nu \) are close for the two data sets at the same \( Re \), the difference between the two data sets being \(~10\%\). This implies similar heat transfer performance of each microchannel in both types of MHE devices.

![Figure 5-8. Data on heat transfer characteristics for the single- and double-layered MHEs: Nusselt number versus Reynolds number.](image)

Figure 5-8 shows results of device-level heat transfer performance, in terms of input admittance \( Y \) vs. the total water flow rate, through the single- and double-layered MHE. For both MHEs, values of \( Y \) increases approximately linearly with increasing total flow rate. The single-layered MHE device exhibits a slightly better performance, \(~15\%\), as compared to the double-layered MHE. Figure 5-10 shows \( Y \) values of the
double-layered, liquid-liquid counter-flow MHE vs. the cool water flow rate, measured at two different rates of constant hot water flow. From Fig. 5-10, it is noted that the device-level heat transfer performance increases with an increase in either the hot- or cool- water flow rate. For the data set measured at the constant hot-water flow rate of 822mL/min, $Y$ ceases to exhibit any significant increase as the cool-water flow rate exceeds 1600mL/min. That is because, at sufficiently high cool-water flow rates, the overall thermal resistance is dominated by the hot-side water-solid convection. At any given hot-water flow rate, boosting the cool-water flow beyond a certain level would therefore yield no further improvement to the device-level heat transfer performance.

Figure 5-9. Data on device-level heat transfer performance for the single- and double-layered MHEs: input admittance versus total flow rate through device.

5.5. Discussion

The difference exhibited by the two data sets displayed in Fig. 5-8 appears to exceed measurement errors. One possible reason for this observed difference lies in the
difficulty in obtaining an accurate measurement of the microchannel solid wall temperature. As shown in Fig. 5-6, two thermocouples were placed on either side of the microchannel array(s), and the solid wall temperature was taken as the arithmetic average of the two thermocouple readings. This measurement scheme yields a solid wall temperature that tends to be higher than the actual value because of the inwards direction of heat flow from the two external surfaces of the MHE, even with the high Cu bulk thermal conductivity. Another possible explanation lies in the different configurations of the two devices, e.g. the double-layered MHE contains two microchannel arrays in two different planes while only one microchannel array in one plane exists in the single-layered MHE. The different cross-sectional configurations may affect the liquid flow pattern in the supply/drain plenum connecting the microchannel array(s), resulting in different entrance effects in the microchannels.

The similar device-level performance for the single- and double-layered MHEs, shown in Fig.5-9, comes from two factors. On one hand, at the same total flow rate, the flow rate per channel of the single-layered MHE is twice that for the double-layered device. This higher flow rate per channel results in larger Reynolds number and consequently higher heat transfer performance at the individual microchannel level, as indicated in Fig. 5-8. On the other hand, the double-layered MHE possesses twice as many microchannels and therefore twice the microchannel wet area where liquid-solid convection takes place. Combining these two factors, the double-layered MHE exhibited similar heat transfer performance as compared to the single-layered device while suffering a much lower pressure penalty. Data in Fig. 5-9 show that, at same total flow rate, the single-layered MHE exhibited a ~15% heat transfer performance boost over the
double-layered MHE. However, it suffered a more-than-double pressure drop penalty, as shown in Fig. 5-7(b). On the other hand, if the same pressure were to be applied on the two MHEs, much higher heat transfer performance would be expected from the double-layered MHE. This means that the double-layered MHE delivers a much higher performance-to-penalty ratio, and is thus the preferred device configuration.

In the current heat transfer test configuration in which a uniform high-temperature environment surrounds the MHE device, it is conceivable that more layers of microchannel arrays can be stacked using the same fabrication/assembly protocol to build a multi-layered MHE. However, it is expected that the device-level heat transfer performance improvement would diminish as more microchannel array layers are stacked over each other. This is because heat is applied only through the external surfaces of the MHE and needs to travel through more solid walls and bonding interfaces to reach the inner microchannel array layers. The added thermal resistance lowers the solid wall temperature of the inner layer, and renders the additional convective heat transfer less effective. Therefore, there is a limit to the number of microchannel array layers that can be stacked, which depends on parameters such as the microchannel height, solid wall thickness, solid thermal conductivity, and the thermal resistance of the bonding interface.

In Fig. 5-10, the highest value of input admittance obtained during testing was 34.6 W/K. At this level of performance, the heat removal rate from the hot water would be 242W at hot water flow rate of 1476 mL/min and cool water flow rate of ~1800 mL/min, given an inlet hot water temperature of 90°C and an inlet cool water temperature of 20°C. It is worthy to note that such a liquid-liquid counter-flow device is multi-layer stackable. In contrast to single-liquid MHEs, the heat transfer performance improvement
for the liquid-liquid counter-flow MHE does not diminish as more microchannel array layers are stacked. This is because the high-temperature heat source in this device configuration is the flowing-through hot liquid. In the alternating hot/cool liquid microchannel array configuration, the distance for heat transport remains unchanged when multiple microchannel array layers are stacked over each other. The heat transfer capacity of such liquid-liquid counter-flow MHEs is therefore scalable. Because the thickness of a single microchannel array layer typically stays below 1mm, as shown in Fig.5-1, low-volume, high-performance, liquid-liquid counter-flow MHEs can be achievable through the fabrication/assembly protocols described in this chapter.

![Graph](image)

Figure 5-10. Data on device-level heat transfer performance for the double-layered, liquid-to-liquid, counter-flow MHE: input admittance versus cool water flow rate through device at constant hot water flow rates.

5.6. Summary

We have fabricated, assembled and tested the single- and double-layered MHEs. The results of liquid flow measurement showed that the double-layered MHE suffers less than half of the pressure penalty that the single-layered MHE does. The heat transfer testing was performed on both MHEs. The results demonstrated that both devices exhibit
similar heat transfer performance at individual microchannel level as well as at device level. These measurement results bring it to a conclusion that the extra microchannel array layer added to the MHE reduces pressure drop across the device while keeping performance largely unchanged. Thus, the two-layered configuration delivers a higher performance-to-penalty ratio, and therefore outmatches the single-layered one to be the preferred MHE design.

The double-layered, liquid-to-liquid, counter-flow MHE was fabricated and assembled. Its device-level heat transfer performance was measured. Analysis has shown that the heat transfer performance of such an MHE is scalable by stacking over more microchannel array layers in design. Thus, a compact, high-performance, multi-layered, liquid-to-liquid MHE is achievable following the present MHE manufacturing procedures.
CHAPTER 6.  MICROELECTRONIC CHIP COOLING: AN EXPERIMENTAL ASSESSMENT OF A LIQUID-PASSING HEAT SINK, A MICROCHANNEL HEAT REJECTION MODULE, AND A MICROCHANNEL-BASED RECIRCULATING-LIQUID COOLING SYSTEM

6.1.  Introduction

As already discussed in chapter 1, the processing power of microelectronic modules has increased in an exponential fashion over the past five to six decades [10]. As the processing power increases, module level heat generation has increased in an equally dramatic manner, as summarized graphically in Fig. 1-1 [9]. Data previously presented in Fig. 1-1 show that the trend of increasing module-level heat output continues for more recent devices, e.g., the maximum heat flux generated by the Intel® Core™2 Extreme QX9775 chip can be as high as 80W/cm² [11].

![Solid Cu fin array and Solid Cu base plate](image)

Figure 6-1. A commercial, all-solid heat sink.

Effective heat dissipation is a major challenge to continued improvement of microelectronic modules. At present, the prevailing technology for heat removal from
microelectronic modules combines all-metal, multiple-fin, heat sinks with forced-air convection cooling. An example of a commercial, all-Cu, multiple-fin heat sink is shown in Fig. 6-1. It is generally recognized that forced air cooling will be insufficient for higher performance microelectronic modules. Chip makers have already attempted to sidestep this exponentially-growing, module-level, heat dissipation problem by going to multiple chip cores, each of which generates less heat. Because of this strategy, current personal computers (PCs) include “dual core”, “quad-core”, and even “octa-core” products. This “multi-core” approach postpones the heat dissipation problem temporarily, without eliminating the shortcomings associated with the current cooling strategy. First, overall capacity for heat flux removal is too low. This in turn limits maximum chip-power and consequently chip-performance. Second, the cooling device occupies significantly more space than the chip to be cooled. The large area/volume footprint of cooling devices limits the installation of multiple processors in close proximity on the same printed circuit board, and consequently the total number of processors.

Figure 6-2. Schematic layout of a typical forced-liquid-flow cooling system.

Alternative cooling technologies with higher heat transfer performance and lower area/volume footprint at the chip location are of current interest as a critical enabler for
next-generation, better-performing PCs and other computing devices. Amongst different approaches, forced-liquid-flow cooling technologies are being intensely studied for dissipating heat from next-generation microelectronic modules. Figure 6-2 shows a schematic layout of typical forced-liquid-flow cooling systems. The heat absorption module (HAM) is placed in intimate thermal contact with the heat source to be cooled. Energy from the heat source transfers to the body of the HAM through conduction. Cold liquid is forced to enter the HAM by the liquid pump and gains energy through liquid-solid convective heat transfer. Hot liquid exits the HAM, carries the energy to the heat rejection module (HRM), and transfers the energy to the solid body of the HRM through liquid-solid convective heat transfer. Final energy removal from the system is achieved through air-solid convective heat transfer by air flow around the exterior of the HRM, forced by the fan module placed in proximity to it.

As stated in chapter 2, liquid flowing through the microchannel array embedded within such low-profile, metal-based MHEs can remove a heat flux of over 240W/cm² at reasonable pressure drops [81]. Such low-profile, metal-based MHEs combine high heat flux removal capacity with low area/volume footprint, and are promising candidates for HAMs within forced-liquid-flow cooling systems shown in Fig. 6-2.

In addition to the HAM, a well-engineered, recirculating-liquid, cooling system demands effective HRMs which can accommodate liquid flow. Furthermore, the operational characteristics of the entire cooling system need to be evaluated quantitatively. As a first step in this direction, this chapter reports results of an experimental evaluation of the heat transfer performance of two liquid-passing HRMs. The first such HRM was obtained by modifying a commercial, all-Cu heat sink to
accommodate liquid flow through the module. The second such HRM was obtained by assembling multiple low-profile Cu MHEs. In each case, the heat transfer performance of an all-solid benchmark device of the same overall dimensions was measured experimentally as a comparison. In addition, a closed-loop, recirculating-liquid, cooling system was built and instrumented. The system-level operational characteristics were evaluated through experimental testing.

In what follows, Section 2 gives a detailed description of the device configurations that are experimentally assessed. Section 3 documents the experimental setup and testing procedures. Section 4 provides details on data analysis procedures. Each of these sections is subdivided into three for the liquid-passing heat sink, the microchannel HRM, and the closed-loop cooling system. Section 5 presents results and discussion. Section 6 offers some concluding remarks.

6.2. Device Configurations Assessed

6.2.1 The Liquid-Passing Heat Sink and its All-Solid Benchmark

Figure 6-3. Channel modification to the base plate of the liquid-passing heat sink shown in Fig. 6-1. Numbers on the drawing are in mm.
Two commercial, 89mm×89mm×26mm, Cu heat sinks, one of which is shown in Fig. 6-1, were used for experimentation. One heat sink remained in the as-received state and served as the benchmark device. The other was modified into a liquid-passing HRM with a meandering channel embedded in its base plate. The dimensions of the meandering channel and its placement on the base plate are shown in Fig. 6-3. The original base plate had a thickness of 6mm. This base plate was milled down to 3mm, then a meandering channel with a depth of 2mm was milled into it. Another 3mm thick Cu plate with the same footprint was Sn bonded to the modified base plate with the open meandering channel to form an enclosed channel structure. In this way, the finished base plate had the same total thickness of 6mm as the benchmark device. Two 3/8” diameter holes with pipe threads were drilled and tapped on the back side of the base plate to connect with the ends of the meandering channel. Two Swagelock® fittings were attached to the holes to serve respectively as the liquid inlet and outlet.

6.2.2 The Microchannel Heat Rejection Module and its Solid-Fin Benchmark

Figure 6-4. A microchannel HRM prototype before assembly. Numbers on the ruler are in mm.
Figure 6-4 shows a microchannel HRM prototype with 20 fins, containing within each fin an array of parallel microchannels. The 20 microchannel fins, together with two fin adapters and two cover plates, made up the entire HRM prototype. Figure 6-5 shows a cross-sectional scanning electron microscopy image of a portion of one typical microchannel fin, with a total thickness of ~760μm. Each fin contained 12 parallel, nominally-identical microchannels in an array configuration.

![Figure 6-5. A cross-sectional SEM image of a portion of one typical microchannel fin used in the microchannel HRM.](image)

The microchannel fins were fabricated from Cu 110 (99.9%+ Cu) sheets. One microchannel fin was made up from two Cu sheets: a 50.5mm×40mm×660μm base sheet and a 50.5mm×40mm×100μm cover sheet. A microchannel array with an active area of 50.5×22mm was cut by sinking-mode micro electrical discharge machining (μEDM) onto the base sheet surface. Prior to μEDM, the Cu base sheet was polished with silicon carbide papers of decreasing grit sizes down to 800. A flat sheet of carbon steel with a thickness of ~500μm was used as the blade electrode for sinking-mode μEDM. This cutting generated microchannels of ~550μm in width and ~550μm in depth. As is typical with the μEDM process, channel bottom exhibited a rounded profile, as shown in Fig. 6-5.
For each microchannel fin, 12 parallel cuts were made sequentially to form an open array of 12 microchannels, with a channel-to-channel spacing of ~1450μm.

To form enclosed microchannels, the base sheet with the open and parallel microchannel array was transient liquid phase (TLP) bonded to a blank Cu cover sheet with a ~10μm thick Al foil as the intermediate bonding layer. The TLP bonding process was carried out in vacuum at a temperature of ~600°C with an applied pressure of ~3MPa. Further details of the Cu/Al/Cu TLP bonding process have been described elsewhere [63, 82].

As shown in Fig. 6-4, a pocket was milled out on the back side of the fin adapters. Twenty parallel through-slots were cut within the pocket. One microchannel fin was inserted into each slot with a fin-to-fin spacing of ~1.5mm. Sealing between the microchannel fins and the fin adapters, and between the fin adapters and the cover plates, was achieved through Sn soldering using the commercially available Kester EM907 soldering paste. The so-constructed fin adapter provided both a plenum for the inlet/outlet liquid as well as a means of connecting to the 20 microchannel fins in a leak-tight package. One 3/8” diameter through-hole was drilled on the back side of the cover plate. Pipe threads tapped into the hole provided a connection to a Swagelock® fitting, which served as the external liquid inlet/outlet port. After assembly, the length of the portion of microchannel fin exposed in air was 47mm. Therefore, the total area available for air-solid heat exchange on each microchannel fin was 2×47mm×40mm.

A companion HRM with solid-Cu fins was built and used as the benchmark for the microchannel HRM prototype. The solid fins shared the same overall dimensions with the microchannel fins, a total thickness of 760μm and a width of 40mm. To attach
20 solid fins to the solid base plates, 20 slots were milled on two solid Cu base plates. The solid fins were Sn soldered to the base plates with the Kester EM907 paste. After assembly, each solid fin within the benchmark device had the same total area available for air-solid heat exchange, 2×47mm×40mm. As shown in Fig. 6-6, the finished, all-Cu, benchmark device shared the same overall dimensions as the microchannel HRM prototype device.

![Benchmarking the heat rejection device: (left) the assembled microchannel HRM; (right) the assembled solid-fin benchmark heat sink.](image)

Figure 6-6. Benchmarking the heat rejection device: (left) the assembled microchannel HRM; (right) the assembled solid-fin benchmark heat sink. Numbers on the ruler are in mm.

### 6.2.3 The Closed-Loop, Recirculating-Liquid, Cooling System

A closed-loop, recirculating-liquid, cooling system was assembled, a schematic of which is shown in Fig. 6-7. Pure water was used as the liquid medium. This recirculating-liquid system was made up of a HAM, a HRM, a liquid pump, and a liquid flow meter. A solid-Cu heater block was placed in intimate thermal contact with the HAM. Four holes arranged in a linear array were drilled onto the heater block for temperature measurements with inserted thermocouples. A compressed air tank was connected through an air flow meter to a custom-made air flow duct. The dimensions of
the air exit were made to fit those of the HRM. During system operation, pure water was pumped through the HAM and absorbed heat from the heater block placed in thermal contact with it. Heated water flew through the HRM, releasing heat into air through forced-air-flow convective cooling. A number of additional thermocouples were placed at appropriate points for measuring needed temperatures throughout the system, as indicated by the thermocouple locations shown in Fig. 6-7.

Figure 6-7. A schematic for testing of the closed-loop, recirculating-liquid, cooling system.

Figure 6-8. The HAM within the recirculating-liquid cooling system, made up of a low-profile Cu MHE. The MHE was Sn-bonded to the Cu heater block. Numbers on the ruler are in mm.
As shown in Fig. 6-8, the HAM consisted of a low-profile, Cu-based MHE Sn-bonded to the solid-Cu heater block. The overall dimensions of the heater block were 37mm×25mm×80mm. The Sn bonding reduced the interfacial thermal resistance between the Cu MHE and the Cu heater block. During system operation, energy was supplied from six cylindrical cartridge heaters inserted into the body of the Cu heater block, transferred by conduction to the body of the Cu MHE, and removed by water flowing through the MHE via liquid-solid convective heat transfer. The MHE was fabricated from Cu 110 alloy (99.9%+ Cu) sheet metal following procedures similar to that used for fabrication of the microchannel fins, described in Section 2.2. This particular Cu MHE had overall dimensions of 42mm×40mm×0.8mm, and enclosed 82 parallel microchannels in an array with an area of 26mm×40mm. The dimensions of individual microchannels were ~172μm in width, ~400μm in depth, and ~40mm in length. A Flojet D3835E7011A diaphragm pump was used as the liquid pump. An Omega FLR1010 flow meter was used to measure water flow throughout the system. During operation, the compressed air tank supplied a continuous and steady air flow to cool down the water flowing through the HRM. Air flow was measured by an Omega® FLR7710D flow meter. The microchannel HRM, depicted in Figs. 6-4 and 6-6 and described in Section 2.2, was used in the cooling system.

6.3 Experimental Setup and Testing Procedures

6.3.1 The Liquid-Passing Heat Sink and its All-Solid Benchmark

Figure 6-9 shows the testing schematic for both the liquid-passing heat sink and its all-solid benchmark. During testing, hot water with temperatures ranging from 54.7°C to 59.6°C was fed through the liquid-passing heat sink. Cooling was accomplished
through air supplied from the compressed air tank, forced to flow through the air duct, and blown on the heat sink. The air duct cross section matched the air passage of the heat sink, so that no contraction or expansion occurred within the air flow. K-type thermocouples were used for temperature measurements in all tests.

Figure 6-9. A schematic for testing both the liquid-passing heat sink and the microchannel HRM.

For the liquid-passing heat sink, two thermocouples were placed respectively at its water inlet and outlet to measure the inlet and outlet temperatures, $T_{i,lp}$ and $T_{o,lp}$. The subscript “lp” in the symbols just defined refers to the “liquid-passing” heat sink. Two additional thermocouples were placed respectively in front and behind the heat sink to measure the air inlet and outlet temperatures, $T_{air-in,lp}$ and $T_{air-out,lp}$. During the test, all data was recorded at steady state at different water flow rates and air flow rates.

For the all-solid benchmark, five thermocouples were placed within the base plate underneath the fins at a depth of ~2mm, as shown in Fig. 6-10. Four thermocouples, $T_{a1}$ to $T_{a4}$, were placed in a 2D Gaussian quadrature pattern to better represent the base plate mean temperature $T_{b,asb}$, in this case taken as $T_{b,asb} = (T_{a1}+T_{a2}+T_{a3}+T_{a4})/4$. The subscript “asb” in $T_{b,asb}$ refers to the “all-solid benchmark”. The thermocouple $T_{a5}$ was placed in the center of the base plate. Two thermocouples were placed respectively in front and
behind the heat sink to measure air inlet temperature $T_{\text{air-in,asb}}$ and air outlet temperature $T_{\text{air-out,asb}}$. A heater with a footprint of 41.3mm×41.3mm was attached to the center of the base plate as the heat source. During testing, the heater was thermally insulated by ~60mm thick PVC foam plates to minimize heat loss. A thermal paste was placed in between the heater and the heat sink to ensure a uniform thermal contact across the entire contact area.

Figure 6-10. Thermocouple positions on the base plate of the all-solid benchmark for the liquid-passing heat sink.

6.3.2 The Microchannel Heat Rejection Module and its Solid-Fin Benchmark

The microchannel HRM and its solid-fin benchmark were also tested with the same setup depicted in Fig. 6-9, with the exception of a different air duct which matched the air passage of the microchannel HRM and its benchmark. During testing, hot water with temperatures ranging from 52.5°C to 59.0°C was fed through the microchannel HRM. Two thermocouples were placed at the water inlet and outlet, respectively, to measure the inlet and outlet temperatures, $T_{i,mHRM}$ and $T_{o,mHRM}$. The subscript “mHRM” in the symbols just defined refers to the “microchannel HRM”. Two additional
thermocouples were placed in the air passage, in front and behind the HRM, respectively, to measure the air inlet and outlet temperatures, $T_{\text{air-in,mHRM}}$ and $T_{\text{air-out,mHRM}}$. Air flow was measured by the same Omega® FLR7710D air flow meter.

Figure 6-11. Thermocouple positions on the base plate of the solid-fin benchmark for the microchannel HRM.

For the solid-fin benchmark, seven thermocouples were placed within the base plate underneath the fins at a depth of ~2mm, as shown in Fig. 6-11. Two thermocouples were placed in the air passage in front and behind the benchmark, respectively, to measure the air inlet and outlet temperatures, $T_{\text{air-in,sfb}}$ and $T_{\text{air-out,sfb}}$. The subscript “sfb” in the symbols just defined refers to the “solid-fin benchmark”. A heater with a footprint of 54.5mm×50.0mm, the same as that of the device base plate, was attached to the backside of the base plate. The heater was thermally insulated by ~60mm thick PVC foam plates to minimize heat loss. A thermal paste was again applied between the heater and the base plate to ensure a uniform thermal contact.
6.3.3 The Closed-Loop, Recirculating-Liquid, Cooling System

As shown in Fig. 6-7, pure water was forced by the liquid pump to flow through the low-profile Cu MHE. Hot water exited the MHE, passed through the microchannel HRM, and was cooled by air flow passing its exterior. The water flow rate was measured by an Omega FLR1010 water flow meter. As shown in Fig. 6-7, three thermocouples were placed within the water passage: before the MHE to measure $T_{in}$, after the MHE to measure $T_{out,1}$, and at the microchannel HRM outlet to measure $T_{out,2}$. During testing, both the MHE and the microchannel HRM were thermally insulated by ~60mm thick PVC foam plates. A heater was attached to the MHE and supplies heat to the system. In order to obtain the temperature of the heater/MHE interface, which is the maximum temperature in the system and the “chip temperature” in electronics cooling, four thermocouples were inserted into the heater block to measure $T_1$ to $T_4$, as shown in Fig. 6-7. The distance between two adjacent thermocouples was 6mm. The distance from the $T_1$ thermocouple to the top surface of the heater was 3mm. The chip temperature was obtained through linear extrapolation of temperature measurements $T_1$ through $T_4$. Two thermocouples were placed in the air passage in front and behind the microchannel HRM, respectively, to measure the air inlet and outlet temperatures. Air flow was measured by the Omega FLR7710D air flow meter.

6.4 Procedures for Data Analysis

6.4.1 The Liquid-Passing Heat Sink and its All-Solid Benchmark

The Reynolds number corresponding to air flowing through the gaps between fins is related to the average air velocity $\bar{V}$, or the volumetric flow rate of air $\dot{Q}_{air}$, as measured by the air flow meter:
where \( D_h \) is the hydraulic diameter of the cross sections of individual air passages between the fins, \( p \) is the cross-sectional perimeter, and \( \mu \) is the air viscosity evaluated at the mean air temperature \((T_{\text{air-in}} + T_{\text{air-out}})/2\). In steady state, the power removed from the hot water flowing through the liquid-passing heat sink is expressed as

\[
q_{lp} = \rho c_p \dot{Q}(T_{i,lp} - T_{o,lp}),
\]

where the water volumetric flow rate \( \dot{Q} \) was obtained by measuring the volume of water collected at the heat sink outlet over a fixed period of time. To quantify the air side heat transfer, an effective heat transfer coefficient is defined as

\[
h_{\text{eff,lp}} = \frac{q_{lp}}{A_b(T_{b,lp} - T_{\text{air-in,lp}})}
\]

where \( A_b \) is the area of the heat sink base plate and \( T_{b,lp} \) is its mean temperature, which in this case is taken as \( T_{b,lp} = (T_{i,lp} + T_{o,lp})/2 \). The overall performance of the heat sink is expressed through a device-level input admittance,

\[
Y_{lp} = \frac{q_{lp}}{(T_{\text{max,lp}} - T_{\text{air-in,lp}})},
\]

where the highest temperature \( T_{\text{max,lp}} \) in this case is the water inlet temperature \( T_{i,lp} \).

For the all-solid benchmark, the steady state power removal by air, \( q_{\text{asb}} \), was read from the electrical power input to the heater. In an analogous manner, an effective heat transfer coefficient for the all-solid benchmark is defined as

\[
h_{\text{eff,asb}} = \frac{q_{\text{asb}}}{A_b(T_{b,asb} - T_{\text{air-in,asb}})},
\]

where \( T_{b,asb} \) is the mean base plate temperature, taken as \( T_{b,asb} = (T_{a1} + T_{a2} + T_{a3} + T_{a4})/4 \) (see Fig. 6-10). The device level input admittance for the all-solid benchmark is likewise defined as
\[ Y_{\text{asb}} = \frac{q_{\text{asb}}}{(T_{\text{max,asb}} - T_{\text{air-in,asb}})}, \quad (6-6) \]

where \( T_{\text{max,asb}} \) in this case is the highest temperature reading from thermocouples \( T_{a1} \) to \( T_{a5} \), as shown in Fig. 6-10.

### 6.4.2 The Microchannel Heat Rejection Module and its Solid-Fin Benchmark

The Reynolds numbers for both the microchannel HRM and its solid-fin benchmark are defined in the same way as what is expressed in Eq. (6-1). In steady state, the power removed from the hot water flowing through the microchannel HRM is expressed as

\[ q_{\text{HRM}} = \rho c_p \dot{Q}(T_{i,mHRM} - T_{o,mHRM}), \quad (6-7) \]

where the water volumetric flow rate \( \dot{Q} \) was again obtained by measuring the volume of water collected at the microchannel HRM outlet over a fixed period of time. The overall performance of the microchannel HRM is expressed through a device-level input admittance,

\[ Y_{\text{mHRM}} = \frac{q_{\text{mHRM}}}{(T_{\text{max,mHRM}} - T_{\text{air-in,mHRM}})}, \quad (6-8) \]

where the highest temperature \( T_{\text{max,mHRM}} \) in this case is the water inlet temperature \( T_{i,mHRM} \).

For the solid-fin benchmark, the steady state power removal by air, \( q_{\text{sfb}} \), was read from the electrical power input to the heater. The device-level input admittance is similarly defined, as compared that for the microchannel HRM, except for a different maximum temperature,

\[ Y_{\text{sfb}} = \frac{q_{\text{sfb}}}{(T_{\text{max,sfb}} - T_{\text{air-in,sfb}})}, \quad (6-9) \]

where \( T_{\text{max,sfb}} \) is the maximum temperature reading from thermocouples \( T_{b1} \sim T_{b7} \), as shown in Fig. 6-11. To capture the maximum base temperature, thermocouples \( T_{b1} \) to \( T_{b7} \)
were placed asymmetrically with $T_{h5}$ to $T_{h7}$ close to the air flow exit where the temperature was expected to be the highest.

6.4.3 The Closed-Loop, Recirculating-Liquid, Cooling System

The rate of heat absorption by the Cu MHE during system-level testing is calculated from the water flow rate and temperature rise across the MHE,

$$q_{MHE,s} = \rho c_p \dot{Q}(T_{out,1} - T_{in}),$$

(6-10)

where $\dot{Q}$ is the volumetric water flow rate, $T_{in}$ and $T_{out,1}$ are respectively the water inlet and outlet temperatures shown in Fig. 6-7. The subscript “s” in the symbol $q_{MHE,s}$ refers to the “system”. The rate of heat rejection into the air by the microchannel HRM is similarly calculated from the water flow rate and temperature drop across it,

$$q_{mHRM,s} = \rho c_p \dot{Q}(T_{out,1} - T_{out,2}).$$

(6-11)

The MHE input admittance is defined as

$$Y_{MHE,s} = \frac{q_{MHE,s}}{(T_{max,MHE,s} - T_{in})},$$

(6-12)

where, as previously stated, $T_{max,MHE,s}$ is the “chip” temperature obtained by extrapolation,

$$T_{max,MHE,s} = (T_1 + T_2 + T_3 + T_4)/4 - (T_4 - T_1) \times 12/18.$$  

(6-13)

The microchannel HRM input admittance is defined as

$$Y_{mHRM,s} = \frac{q_{mHRM,s}}{(T_{max,mHRM,s} - T_{air,in})},$$

(6-14)

where $T_{max,mHRM,s}$ is the maximum temperature of the microchannel HRM, which in this case is $T_{out,1}$. The system-level input admittance, which characterizes the overall heat transfer performance of the system, is expressed as

$$Y_s = \frac{q_{MHE,s}}{(T_{max,s} - T_{air,in})},$$

(6-15)
where $q_{MHE,s}$ is the rate of total heat input into the system, $T_{\text{max},s}$ is the maximum temperature in the system or the “chip temperature” as expressed in Eq. (6-13).

### 6.5 Results and Discussion

![Image of graph showing effective heat transfer coefficient plotted against air flow Reynolds number.](image)

Figure 6-12. Comparison between $h_{\text{eff}}$ values of the liquid-passing heat sink and its all-solid benchmark. For data taken from the liquid-passing heat sink, the corresponding water flow rates are noted on the figure legend.

Figure 6-12 shows measured values of the effective heat transfer coefficient for the liquid-passing heat sink and its all-solid benchmark, $h_{\text{eff,lp}}$ and $h_{\text{eff,asb}}$, plotted as a function of the air flow Reynolds number, Re. Four sets of measurements for $h_{\text{eff,lp}}$ were obtained at four different rates of water flow through the heat sink. One set of $h_{\text{eff,asb}}$ data was obtained. For all five data sets, measured values of $h_{\text{eff}}$ increase with increasing Re. All five data sets overlap to $\pm 5\%$. Results shown in Fig. 6-12 demonstrate good agreement between $h_{\text{eff}}$ values measured from the all-solid benchmark and the liquid-passing heat sink under different water flow conditions.
By construction, these two devices have nominally identical solid fin geometries. As shown in Section 4.1, the calculation of $h_{\text{eff}}$ is based on the mean base temperature. Given the same air flow, the type of heat source—either direct heating from the attached heater or heating by hot water flowing through the device—should not make any difference on the $h_{\text{eff}}$ value. Agreements in measured $h_{\text{eff}}$ values are therefore expected because power removal is dominated by air-solid convective heat transfer from the fin array. Data shown in Fig. 6-12 therefore attest to the correctness of the measurement scheme and the quality of the measurements. With the same multiple-solid-fin-array geometry, approximately constant air-solid heat transfer performance was observed, even at very small liquid flow rates, as long as the average base temperature is the same. It is also noted that the magnitude of measured $h_{\text{eff}}$ values is typical for air-solid convective heat transfer, and much smaller when compared to liquid-solid convective heat transfer.

Figure. 6-13. Comparison between the overall performance of the liquid-passing heat sink and its all-solid benchmark - input admittance vs. air flow Re. For data taken from the liquid-passing heat sink, the corresponding water flow rates are noted on the figure legend.
Figure 6-13 shows measured values of the input admittance for the liquid-passing heat sink and the all-solid benchmark, $Y_{lp}$ and $Y_{asb}$, plotted as a function of the air flow Reynolds number $Re$. For both devices, $Y$ value increases with increasing $Re$. At the same $Re$, $Y_{lp}$ increases monotonically with increasing rate of water flow through the device base plate. At the water flow rate of ~130ml/min, measured $Y_{lp}$ values are lower than those of $Y_{asb}$. Measured $Y_{lp}$ values approximately match values of $Y_{asb}$ when the water flow rate was increased to ~350ml/min, and exceed $Y_{asb}$ values at higher flow rates. The heat transfer performance of the liquid-passing heat sink at ~700ml/min of water flow rate is ~25% better than that of the all-solid benchmark.

While data shown in Fig. 6-12 indicate that there is no difference in air-solid convective heat transfer for the liquid-passing heat sink and its all-solid benchmark, Fig. 6-13 suggest that sufficient water flow within the base plate of the liquid-passing heat sink does improve the device-level heat transfer performance. For the all-solid benchmark, the heater was attached to the center of the base plate, and heat conduction causes a temperature drop from the center to the peripheries of the base plate. For the liquid-passing heat sink at small water flow rates, e.g., ~130ml/min, the base plate temperature distribution was apparently even less uniform, and entering hot water was cooled down before it reached the outlet. At sufficiently high water flow rates, e.g., ~530ml/min or higher, measurements shown in Fig. 6-13 imply that the base plate temperature distribution in the liquid-passing heat sink became more uniform than that of the all-solid benchmark, as a high rate of hot water flow distributes heat faster and more uniformly across the base plate. Figure 6-13 shows that, at the same maximum temperature difference between the base plate and the inlet air, more power can be
removed from the liquid-passing heat sink than the all-solid benchmark, once a certain water flow rate is exceeded.

Figure 6-14 shows measured values of the input admittance for the microchannel HRM and its solid fin benchmark, $Y_{\text{mHRM}}$ and $Y_{\text{sfb}}$, as a function of the air flow Reynolds number $Re$. Two sets of measurement on the solid fin benchmark were performed at total heater input powers of 100W and 150W, respectively. As expected, values of $Y_{\text{sfb}}$ as defined in Eq. (6-9) do not depend on the heater input power and these two data sets coincide with each other. Data on $Y_{\text{sfb}}$, shown in Fig. 6-14, confirm that the measurements were conducted correctly and give an indication of the measurement quality. Three sets of measurements on the microchannel HRM were conducted, at increasing total water flow rates of 350ml/min, 640ml/min, and 980ml/min. In all cases,
measured $Y_{\text{nlHRM}}$ values are higher than those of $Y_{\text{sfb}}$. The performance of the microchannel HRM at the total water flow rate of $\sim980\text{ml/min}$ exceeds that of the solid fin benchmark by $\sim28\%$ and $\sim50\%$ at $\text{Re}$ of $\sim825$ and $\sim3100$, respectively.

In the case of the solid fin benchmark, because the heater had the same footprint as that of the benchmark’s base plate, lateral heat distribution over the base plate plays little role in device-level heat transfer performance. Difference in performance of the microchannel HRM and its solid fin benchmark results mainly from temperature distribution on the fins. Temperatures at the junctions between fins and the base plate are the highest, and they drop along the fins as heat is being transported from the base plate to the top. For the microchannel HRM, hot water flows within each and every one of the fins from the base plate to the top. Besides solid conduction, the hot water flow offers an additional mechanism for energy transport from the base plate to the top and leads to more uniform temperature distribution along the fins. This more uniform temperature distribution is responsible for the microchannel HRM outperforming its solid fin benchmark in terms of the device-level input admittance.

Figure 6-15. Dependence of the overall recirculating-liquid cooling system performance on air flow and water flow: (a) system performance vs. air flow to the HRM; (b) system performance vs. water flow through the system.
Testing of the closed-loop, recirculating-liquid, cooling system was carried out, as shown schematically in Fig. 6-7. Figure 6-15(a) shows the system-level input admittance \( Y_s \) at \( \sim 820 \text{ml/min} \) of constant water flow through the system, measured versus the Reynolds number \( \text{Re} \) of air flow through the microchannel HRM. Figure 6-15(b) shows \( Y_s \) under a constant air velocity of \( \sim 11 \text{m/s} \) and a corresponding \( \text{Re} \) of \( \sim 2100 \), measured versus the water flow rate. Data shown in Fig. 6-15 indicate that the system-level performance increases with increasing air flow through the HRM, and is largely independent of the rate of water flow through the system.

![Figure 6-16. Module-level heat transfer performance of the heat absorption module and the heat rejection module under: (a) constant air flow to the HRM; (b) constant water flow through the system.](image)

In the same system test, module-level heat transfer performance was also obtained separately for the HAM, i.e., the low-profile Cu MHE, and the microchannel HRM. Figure 6-16(a) plots the input admittance of the Cu MHE, \( Y_{\text{MHE,s}} \), and the input admittance of the microchannel HRM, \( Y_{\text{mHRM,s}} \), measured as a function of the water flow rate through the system. As described above, the air flow rate was kept constant at \( \sim 11 \text{m/s} \) (\( \text{Re} \sim 2100 \)). Figure 6-16(b) plots values of \( Y_{\text{MHE,s}} \) and \( Y_{\text{mHRM,s}} \), measured at a constant water flow rate of \( \sim 820 \text{ml/min} \) as a function of the air flow \( \text{Re} \). At a fixed air
flow rate, Fig. 6-16(a) shows that increasing water flow through the system leads to a ~33% increase in $Y_{MHE,s}$ while $Y_{mHRM,s}$ remains approximately a constant. At a fixed water flow rate of ~820 ml/min, Fig. 6-16(b) shows that increasing the air flow from Re ~750 to ~3200 brings an approximately two-fold increase in $Y_{mHRM,s}$ while $Y_{MHE,s}$ remains approximately a constant. In both scenarios, it is noted that $Y_{MHE,s}$ is 6-10 times larger than $Y_{mHRM,s}$. The HRM therefore limits the performance of the entire cooling system.

Data shown in Fig. 6-16(a), indicating that the performance of the Cu MHE increases with increasing water flow while the performance of the HRM is largely independent of water flow, are consistent with the overall system performance being independent of water flow, as indicated by Fig. 6-15. Similarly, Fig. 6-16(b) shows that increasing air flow enhances the HRM performance and consequently boosts the overall system performance, again consistent with data shown in Fig 6-15.

The fact that $Y_{MHE,s}$ is 6-10 times larger than $Y_{mHRM,s}$ helps to illustrate the operating characteristics of such closed-loop, recirculating-liquid, cooling systems. Liquids forced to flow through the Cu MHE is highly effective in removing energy from the heat source and transporting the energy to a remote location where more effective heat rejection can be achieved. The low-profile nature of the present Cu MHEs provides a much reduced area/volume footprint at the heat source or “chip” location. As shown in Figs. 6-15 and 6-16, better engineered HRM is critical to increasing the overall performance of recirculating-liquid cooling systems. Such engineering efforts are currently ongoing.
6.6 Summary

A detailed experimental assessment of a liquid-passing heat sink, a microchannel heat rejection module, and a microchannel-based recirculating-liquid cooling system was carried out. Verification of the testing protocol was accomplished by comparing measured $h_{\text{eff}}$ values for the liquid-passing heat sink and its all-solid benchmark. By adopting a liquid-passing base plate, performance of a commercial heat sink was enhanced by up to $\sim$25%. By incorporating an array of microchannels within each fin, the device-level performance of the microchannel HRM exceeded that of its solid fin benchmark by up to $\sim$50%.

Detailed testing was performed on a recirculating-liquid cooling system, comprised of a low-profile Cu MHE as the HAM and a microchannel fin assembly as the HRM. System-level performance increased with increasing air flow to the HRM and was largely independent of the rate of water flow through the system. The heat transfer performance of the HAM exceeded that of the HRM by 6-10 times. The present testing results indicate that low-profile Cu MHEs are highly effective in heat flux removal while having a small area/volume footprint, and that enhancing the HRM performance is critical to boosting the overall performance of such recirculating-liquid cooling systems.
CHAPTER 7.  ROLL MOLDING OF MICROCHANNEL ARRAYS ON AL AND CU: AN ALTERNATIVE METHOD FOR HIGH-THROUGHPUT MANUFACTURING

7.1 Introduction

As discussed in Chapter 1, various fabrication technologies for metal-based microscale structures have been studied, including micro-electrical-discharge-machining (EDM) [28], micro-milling [24], and direct laser writing [83]. These methods are serial: the entire structure is fabricated one cut at a time and often involves multiple passes, and are therefore time-consuming. Micro EDM and milling also suffer from tool wear. These methods are therefore less suitable for mass production.

Direct compression molding of metals at the microscale was studied as a potential pattern generation technique, and has been demonstrated successfully in Pb and Zn [50], Al [51], Cu [53], Ni and NiTi [56]. Microscale compression molding involves a mold insert containing a pattern inverse to the final desired pattern on the work piece or substrate. Patterns on the mold insert are transferred onto the substrate surface through deformation of the substrate when it is placed in contact with the insert under a compression load [43]. In the case of metals, the contact stresses induce large-strain plastic deformation of the substrate, which plastically flows around the microscale patterns on the insert to form the desired pattern, to a certain molding depth. As compared to micro EDM, micro milling, and direct laser writing, pattern replication by compression molding enjoy a number of advantages, chief among them is fast and parallel pattern formation. There are also several shortcomings associated with this method. First, the required compression force scales linearly with the area of the pattern,
making scaling up to larger pattern footprints more difficult. Second, automation of this method requires complex, multi-stage equipment with precision positioning capabilities.

This chapter introduces roll molding as an alternative method for fabricating microchannel arrays with large depths on sheet metals. Roll molding can be viewed as a variant of the compression molding method, but with fabrication geometry more analogous to flat rolling of sheet metals [84]. In roll molding, cylindrical rollers are replaced with rollers containing microscale protrusions on the external surface. In this geometry, line contact occurs between the patterned roller(s) and the substrate, instead of simultaneous contact over the entire pattern area as in compression molding. The required compressing force for achieving a certain pattern depth is thus much lower as compared to compression molding. In addition, scaling up for roll molding to larger patterns requires a force increase that is proportional to the contact width instead of the entire pattern area. These two factors combined to make pattern replication onto larger substrates easier to implement. The roll molding geometry is also conducive to high-throughput manufacturing, as will be demonstrated in this chapter.

7.2 Experimental Setup and Procedures

Roll molding of microchannel arrays on Al sheet metals was carried out on a custom-designed and built instrument. Figure 7-1 shows a schematic of the roll molding process. Two steel rollers, with nominal outer diameters of 108 mm (4.25 in), rotated with an angular speed range of 0-6 RPM. The upper roller was attached to a hydraulically-driven mechanical head assembly, and was actuated to move in the vertical direction. The rolling machine was built such that continuous measurements can be made of the normal loading force applied to the upper roller, the total input torque, and the
displacement of the upper roller through contact with four linear-variable-displacement-
transducers (LVDTs), two placed in front of the roller entrance and two placed behind the
roller exit.

![Figure 7-1. A schematic for making microchannel arrays on sheet metals by rolling
molding.]

Both steel rollers were made to accommodate roller sleeves containing microscale
protrusions on their external surfaces. One roller sleeve, as shown schematically in Fig.
7-2(a), was assembled onto the upper steel roller and locked with a key. As shown in Fig.
7-2(b), this roller sleeve was made of the Ni-based superalloy Inconel X750 and contains
an array of circumferential micro-protrusions. After roll molding, the inverse of the
protrusion pattern was imprinted onto the surface of Al sheet metals, forming an array of
microchannels. Figure 7-2(c) shows the nominal dimensions of the protrusions on the
roller sleeve. The protrusion sidewalls were designed to possess a slight taper, ~7.6°, for
increasing its bending resistance. The outer diameter of the roll sleeve was 108.7 mm
(4.28 in), slightly larger than the outer diameter of the steel roller.
Inconel X750 is known for its combination of high strength and toughness, with good retention of mechanical properties at elevated temperatures [85]. The micro-protrusion pattern on the roller sleeve was fabricated by micro wire EDM. After EDM, the roller sleeve was subjected to an electrochemical polishing in 1:1 HClO₄:CH₃COOH solution to remove surface residue formed during the EDM process. An unbalanced magnetron sputtering process was then used to deposit a W-containing amorphous hydrogenated carbon (W-C:H) coating, ~1 μm in thickness, onto the roller sleeve surface. An adhesion-promoting, elemental Cr layer, ~200nm in thickness, was deposited onto the
roller sleeve surface prior to W-C:H deposition. The presence of the adherent W-C:H coating on roller sleeve surface acts to reduce roller-to-metal surface interaction during roll molding and the consequent damages [51]. Figures 7-3(a) and 7-3(b) show respectively magnified scanning electron microscopy (SEM) images of typical micro-protrusions on the Inconel roller sleeve, before and after W-C:H coating.

![Figure 7-3. SEM images of typical micro-protrusions on the Inconel X750 roller sleeve: (a) before W-C:H coating, (b) after W-C:H coating.](image)

Commercially available Al 1100 (99%+ Al) and Cu 110 (99.9%+ Cu) sheet metal strips were chosen as the substrates for microchannel roll molding experiments. The initial thicknesses of Al and Cu strips were 2.35 mm and 1.59 mm, respectively. The initial widths of Al and Cu strips were both 19.1 mm. As-received Al and Cu strips were annealed in air at 300°C for ~12 hours and ~18 hours, respectively. Because the strip widths were less than the pattern width on the roller sleeve (25.4 mm), microchannels patterns were imprinted onto strip surfaces across their entire widths after roll molding.

During roll molding, the two rollers, one plain steel roller without pattern and the Inconel roller sleeve with pattern, were rotating such that a constant rolling surface speed of 1.4 mm/s was achieved. Annealed metal strips were placed in between the bottom and top rollers, and subjected to an increasing normal load until a pre-set load level is reached. The metal strip was then rolled in steady state between the rollers, while data from the
load cell monitoring the normal load force and the four LVDTs monitoring the upper roller displacement were continuously recorded. Multiple independent experiments were carried out by varying the normal loading force applied to annealed Al and Cu strips, while keeping all other parameters unchanged. Different normal loading forces resulted in different microchannel depths. No surface damage or structure deformation was observed on the Inconel roller sleeve after roll molding, as is evident in Fig. 7-2(b). The depth of a microchannel resulting from roll molding was measured on a VanGuard® optical microscope with a calibrated focal depth dial by focusing on the microchannel top and bottom. At least 3 independent depth measurements were carried out in the middle portion of a microchannel array, from which the average depth value and its standard deviation were calculated.

Characterization of as-received and roll molded materials were conducted. Morphological examinations were conducted on a Hitachi S3600N scanning electron microscope (SEM). Additional examinations were conducted on a FEI Quanta3D FEG Dual-Beam focused ion beam (FIB) instrument, from which electron-induced secondary electron (SE) images and ion-induced secondary electron (ISE) images can be obtained. Microhardness measurements on as-annealed and rolled metal strips were conducted on a Future-Tech® FM-1E tester, using a diamond Vickers indenter. For all measurements on Al and Cu strips, constant loads of 25 g and 100 g were used for Al, respectively. Diagonal length of Vickers impressions ranged from ~25 μm to ~50 μm, corresponding to a range of indentation depth ranging from ~3 μm to ~7 μm.
7.3 Validation of Rolling Experiments: Rolling of Flat Al Strips

In order to validate the measurements from roll molding experiments, two sets of experiments were conducted in the flat rolling geometry with annealed Al strips. Two sets of annealed Al 1100 stripes, with respective initial thicknesses of 2.2 mm and 1.0mm, were rolled between two featureless cylindrical rollers to different thickness reductions. The normal force and torque applied were measured and recorded in steady state. The final strip thicknesses were obtained through averaging 6 independent thickness measurements at random locations on the rolled strip, from which an average thickness and a standard deviation were calculated. The widths of rolled strips were also measured, and were found in all cases to not exceed that of the initial strip by more than 5.2%. The collection of dimensional measurements for rolling of flat Al strips is shown in Table 7-1.

Table 7-1. Results of dimensional measurements for flat rolling of annealed Al strips.

<table>
<thead>
<tr>
<th>No.</th>
<th>Average initial thickness</th>
<th>Average final thickness</th>
<th>Average initial width</th>
<th>Average final width</th>
<th>Width increase</th>
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<tr>
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<td>25.36</td>
<td>0.9</td>
</tr>
<tr>
<td>2</td>
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</tr>
<tr>
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</tr>
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<td>25.3</td>
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</tr>
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<td>25.88</td>
<td>1.3</td>
</tr>
<tr>
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<td>1.056</td>
<td>25.3</td>
<td>26.25</td>
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</tr>
<tr>
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<td>25.23</td>
<td>26.54</td>
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</tr>
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</tr>
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</tr>
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<td>1.042</td>
<td>25.36</td>
<td>26.36</td>
<td>3.9</td>
</tr>
</tbody>
</table>
Fig. 7-4 shows the geometry of flat rolling, assuming no change in strip width during rolling. In all flat rolling experiments conducted, the observed width expansion of Al strips was $< 5.2\%$, indicating that the plane strain deformation assumption holds well. The von Karman equation captures the essential mechanics of flat rolling [86]:

$$d (\sigma_x h) = -2P_r (\tan \alpha - \mu) dx, \quad \text{entry to neutral}, \quad (7 - 1a)$$

$$d (\sigma_x h) = -2P_r (\tan \alpha + \mu) dx, \quad \text{neutral to exit}. \quad (7 - 1b)$$

In Eq. (7-1), $\sigma_x$ is the normal stress in the $x$ direction, $h$ is the strip thickness, $P_r$ is the local pressure, and $\mu$ is an effective coefficient of friction between the Al strip and the rollers. The deformation region is divided into two segments, separated by the neutral point where the relative speed between the Al strip and the rollers equals zero. Assuming the material being rolled is elastic - perfectly plastic, a solution for the local pressure can be obtained from the von Karman equation [86]:
\[ P_r = \left[ \frac{-1 + \mu A \alpha}{\mu^2 A} + e^{\mu A (\alpha_{\text{max}} - \alpha)} \left( 1 + \frac{1 - \mu A \alpha_{\text{max}}}{\mu^2 A} \right) \right] \frac{2}{\sqrt{3}} \sigma_0, \text{ entry to neutral}, \quad (7-2a) \]

\[ P_r = \left[ \frac{-1 - \mu A \alpha}{\mu^2 A} + e^{\mu A \alpha} \left( 1 + \frac{1}{\mu^2 A} \right) \right] \frac{2}{\sqrt{3}} \sigma_0, \quad \text{neutral to exit.} \quad (7-2b) \]

A number of parameters enter Eq. (7-2). \( \sigma_0 \) is the flow stress of the strip. The parameter \( A \) is defined by the roller radius and the final strip thickness, \( A=2R/h_f \). The reduction ratio, \( r \), is defined by the initial and final strip thicknesses, \( r=(h_i-h_f)/h_i \). The maximum contact angle can be calculated from the initial strip thickness, the roller radius, and the reduction ratio, \( \alpha_{\text{max}} = \sqrt{rr_i/R} \). Location of the neutral point is determined by \( P_r(\text{entry to neutral}) = P_r(\text{neutral to exit}) \). The normal loading force required to achieve a particular reduction ratio can then be obtained by integrating \( P_r \) from \( \alpha = 0 \) to \( \alpha = \alpha_{\text{max}} \).

Figure 7-5. Flow stress vs. strain obtained from microhardness measurements on as-annealed and roller Al strips: (a) data for 2.2mm thick Al strips; (b) data for 1.0mm thick Al strips.

While the von Karman equation was derived assuming that the rolled material is elastic—perfectly plastic, the actual Al 1100 material strain hardens. Figure 7-5 shows microhardness values on as-annealed and rolled Al strips, as a function of the reduction ratio. Following Tabor, measured Vickers hardness number was then converted into a flow stress through \( \sigma = 3.333HV \) [87]. Thickness reductions were converted into strains,
through $\epsilon = \ln \left( \frac{h_i}{h_f} \right)$. Figures 7-5(a) and 7-5(b) show data obtained from 2.2 mm and 1.0 mm thick Al strips, respectively. Curve-fitting was performed on data sets shown in Fig. 7-5 assuming power-law hardening, $\sigma = a\epsilon^n + b$. It is evident that reasonable fits were obtained on each data set and that reasonable agreement exists between the two data sets. From fits to the stress-strain data shown in Fig. 7-5, an effective mean flow stress associated with a particular thickness reduction in flat rolling was obtained through dividing the integrated stress over strain by the final strain. This effective flow stress was then used as the flow stress parameter, $\sigma_0$, in the von Karman solution described above.

![Figure 7-6](image-url)

**Figure 7-6.** Thickness reduction ratio vs. normal loading force per width: a comparison between experimental data and solution to the von Karman equation for the 2.2 mm thick Al strips.

The effective friction coefficient between the Al strips and the rollers remains undetermined. The value of $\mu$ was assumed to be a constant and chosen such that the von Karman calculation of the normal loading force per width vs. the thickness reduction ratio matches the experimental data on 2.2 mm wide Al strips. The comparison is shown in Fig. 7-6. The normal loading force per width was calculated from measured normal loading force value divided by the final strip width. Error bars in Fig. 7-6 were derived
from multiple measurements on the initial and final thicknesses of Al strips. As shown in Fig. 7-6, a good comparison between the experimental data and the von Karman calculations was obtained by taking the value of $\mu$ to be 0.20. The von Karman calculations were then repeated for 1.0 mm thick Al strips, taking the same $\mu$ value of 0.20 and obtaining effective mean flow stresses from data shown in Fig. 7-5(b). Figure 7-7 shows that a reasonable agreement between the experimental data and the calculated values was again obtained in the case of 1.0 mm thick Al strips, with no adjustable parameters existing in the von Karman calculations. The chosen value of $\mu$, 0.20, is also reasonable as a friction coefficient. Data and calculations shown in Figs. 7-6 and 7-7 offer validation to the measurements made on the present rolling machine.

![Figure 7-7. Thickness reduction ratio vs. normal loading force per width: a comparison between experimental data and solution to the von Karman equation for the 1.0mm Al strips.](image)

**7.4 Results and Discussion**

Figure 7-8 shows overviews of microchannel arrays on top of Al and Cu strips, formed by roll molding with the same W-C:H coated Inconel roller sleeve at room temperature. It is evident that parallel microchannel arrays were imprinted uniformly
across Al and Cu strips. Figure 7-9 shows close-up views of typical roll molded microchannels on surfaces of Al and Cu strips. Figure 7-9 shows sharp transitions from sheet metal top surfaces to microchannel sidewalls, and from microchannel sidewalls to the bottoms. The morphology of the microchannel bottom surfaces reflects typical surface morphology at the top of micro-protrusions on the roller sleeve.

![Figure 7-8. Overview of typical roll molded microchannel arrays on surfaces of Al and Cu strips: (a) Al; (b) Cu.](image)

![Figure 7-9. Close-up view of typical roll molded microchannels on surfaces of Al and Cu strips: (a) Al; (b) Cu.](image)

Figures 7-10(a) and 7-10(b) show cross sectional ISE images of a typical roll molded Cu microchannel. A sidewall taper is evident in Fig. 7-10(a). The measured sidewall taper angle, ~10°, is slightly larger than the nominal value for the micro-protrusions on the roller sleeve. While the exact cause for this discrepancy cannot be
ascertained until more detailed comparisons between experiments and elasto-plastic simulations are made, it is believed that elastic spring back is a contributing factor to this discrepancy. The Cu grain structure is delineated clearly through Ga\(^+\) ion channeling contrast in Fig. 7-10(a). It is evident from Fig. 7-10(a) that the majority of material flow is around the bottom corners of the microchannel. Figure 7-10(b) shows a close-up ISE image near the channel bottom corners, in which streamlines of materials flow are clearly visible. Figures 7-8, 7-9, and 7-10 demonstrate that microchannel arrays with large depths can be generated in Al and Cu sheet metals by room temperature roll molding. The so-generated microchannel arrays have good structural integrity and reasonable surface roughness.

Figure 7-10. Cross-sectional view of a typical roll molded Cu microchannel: (a) channel geometry and grain structure; (b) close-up view near channel bottom corners showing pattern of material flow during roll molding.

Figure 7-11 shows the relationship between the average final microchannel depth and the normal loading force per width during roll molding, for both annealed Al and Cu. Here, force per width was calculated from the total applied normal force divided by the final strip width after roll molding. As evident from Fig. 7-11, an approximately linear relationship between the microchannel depth and the normal loading force per width
exists for both Al and Cu. During the experiment, the total input torque was also measured and recorded. Figure 7-12 shows that an approximately linear relationship also exists between the final microchannel depth and the total input torque applied. The finite intercept at zero microchannel depth signifies the system resistance in idle rotation.

Figure 7-11. Depths of Al and Cu microchannels versus the normal loading force per width.

Figure 7-12. Depths of Al and Cu microchannels versus the total applied torque.

It is of interest to examine strip elongation resulting from roll molding, due to material flow in the longitudinal direction perpendicular to the roller axes. The elongation is defined by $e = (l_f - l_i)/l_i$, where $l_i$ and $l_f$ are respectively the initial and final length of the Al strip, before and after roll molding. This elongation was calculated from the area reduction ratio, $r_A = (A_i - A_f)/A_i$, where $A_i$ and $A_f$ are respectively the initial and
final cross-sectional area of the metal strip. Based on volume conservation, $e$ can be calculated through $e = r_{A_i}/(1-r_{A_f})$. Figure 7-13 shows cross sectional profile of typical roll molded Al and Cu microchannel arrays. The initial cross sectional area $A_i$ was calculated from measurements of the initial thickness and width of the rectangular metal strips. Quantitative measurements of the final cross sectional area $A_f$ was obtained from cross-sectional images similar to those shown in Fig. 7-13, using the software ImageJ [38] to perform area integration. Figure 7-14 shows values of $e$ as a function of the microchannel depth. At small channel depths, ~200μm, a minimal amount of elongation is observed, ~7% or less, and plastic deformation is mainly restricted in the plane of contact. The elongation increases approximately linearly with increasing channel depth, to ~20% as the channel depth increases to ~600μm. In such cases, material flow in the longitudinal direction cannot be ignored, and plastic deformation becomes truly three dimensional (3D). Results of detailed dimension measurements related to roll molding of Al- and Cu- based microchannel arrays are summarized in Tables 7-2 and 7-3. In the case of Al strips, expansion in strip width was less than 2.5% while the maximum elongation was 19.4%. In the case of Cu strips, expansion in strip width was less than 4.1% while the maximum elongation was 22.0%.

Figure 7-13. Cross section profile of roll-molded microchannel arrays: (a) in Al; (b) in Cu.
Figure 7-14. Longitudinal elongation vs. microchannel depth.

Table 7-2. Results of dimensional measurements for roll molding of microchannels on Al strips.

<table>
<thead>
<tr>
<th>No.</th>
<th>Microchannel depth (um)</th>
<th>Initial width (mm)</th>
<th>Final width (um)</th>
<th>Width increase (%)</th>
<th>Elongation (%)</th>
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<td>19.35</td>
<td>1.3</td>
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<tr>
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<td>1.3</td>
<td>18.6</td>
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<td>19.57</td>
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<td>18.1</td>
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</tbody>
</table>

Table 7-3. Results of dimensional measurements for roll molding of microchannels on Cu strips.

<table>
<thead>
<tr>
<th>No.</th>
<th>Microchannel depth (um)</th>
<th>Initial width (mm)</th>
<th>Final width (um)</th>
<th>Width increase (%)</th>
<th>Elongation (%)</th>
</tr>
</thead>
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<tr>
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<td>1.5</td>
<td>0.7</td>
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<td>19.66</td>
<td>3.3</td>
<td>14.1</td>
</tr>
</tbody>
</table>
Figure 7-15. Microchannel depth versus normal loading force per width, normalized by flow stresses of as-annealed Al and Cu strips, 101±6 MPa and 278±18 Mpa, respectively.

Detailed measurements of normal loading force and insert displacement have been carried out in the case of plane strain compression molding. In particular, measurements made in the case of microscale compression molding of Cu at temperatures ranging from 450°C to 550°C have shown that an appropriate flow stress at the molding temperature can be used as a scaling parameter to collapse widely varying molding force – insert displacement curve onto a universal molding response curve [60]. This statement holds true in the presence of significant strain hardening and dynamical recrystallization [60]. In the present experiments of room temperature roll molding, annealed Al and Cu strips were roll molded by the same Inconel roller sleeve, with identical micro-protrusions on the sleeve surface. While the plastic deformation associated with roll molding is complex and three dimensional, the same deformation geometry applies to both Al and Cu. It is thus expected that an appropriate flow stress might again be a good scaling parameter for normalizing the channel depth – force per width data shown in Fig. 7-11. Figure 7-15 shows roll molded microchannel depth as a
function of measured normal loading force per width, normalized by the flow stress of as-annealed Al and Cu strips. The flow stresses were again converted from microhardness measurements on as-annealed Al and Cu strips. Figure 7-15 shows an essential overlap of the microchannel depth – normalized normal force per width curves for Al and Cu, and that an appropriate flow stress is indeed a good scaling parameter for normalizing roll molding responses from different materials.

![Graph showing normalized stress vs. strain for Al and Cu strips. Stress at different strain was normalized by the stress at zero strain.](image)

Figure 7-16. Normalized stress vs. strain for Al and Cu strips. Stress at different strain was normalized by the stress at zero strain.

A true representative strain and the corresponding representative flow stress for rolling molding is difficult to obtain, as the plastic deformation is 3D and highly localized, as evident from Tables 7-2 and 7-3 as well as Fig. 7-10. Figure 7-16 shows flow stresses versus strain for annealed Al and Cu strips, again converted from microhardness measurements. All error bars were derived from multiple measurements. The stresses were normalized by the value for as-annealed strips or the initial flow stress at zero strain, and shown in percentages. The general agreement between the two data sets shown in Fig. 7-16 indicates that annealed Al and Cu strips share similar strain hardening. The
success of using the as-annealed flow stress as the scaling parameter in Fig. 7-15 is therefore understandable in view of data shown in Fig. 7-16, since the ratios between the true representative flow stresses and the as-annealed flow stresses should be close for Al and Cu for the same deformation geometry, however complex.

Results shown in Figs. 7-8 ~ 7-15 demonstrate an alternative method for low-cost, high-throughput manufacturing of Al- and Cu- based microchannel arrays. The roll molding method allows for manufacturing of microchannel arrays in sheet metals in a continuous manner. A maximum microchannel depth of ~600μm was achieved in both Al and Cu at room temperature, which in the present microchannel geometry corresponds to features with aspect ratios of ~2:1. After completing all roll molding experiments on Al and Cu strips, no sign of wear or structural damage was evident from the micro-protrusions on the Inconel roller sleeve.

Double-sided roll molding with a pair of facing roller sleeves both containing microscale features is also possible. This configuration incurs minimal additional cost, and provides the key element for fabrication of low-profile, double-layered, microchannel heat exchangers [88]. In cases where microchannel arrays with even higher aspect ratios are desired, pre-heating of metal strips before they are fed into the rollers will serve to reduce their flow stress, and consequently reduces the stresses required to reach a certain channel depth.

7.5 Summary

This chapter has shown an alternative manufacturing method for low-cost, high-throughput production of high-aspect-ratio, metal-based, microchannel arrays. The roll molding method allows for a streamlined manufacturing of microchannel arrays in sheet
metals in a continuous manner, and has been demonstrated successfully in the cases of Al and Cu. Normal loading forces and torques were measured during roll molding of annealed Al and Cu, with measurements validated by additional flat rolling experiments conducted on Al strips. Results shown in this chapter serve as a basis for future design optimization of patterned roller sleeves, and provide guidance for fabrication of metal-based microchannel arrays by roll molding.
CHAPTER 8. SUMMARY

This dissertation focused on improving the functionality of metal-based microchannel heat exchangers (MHEs), as well as pushing this technology toward real-world applications. Efforts were made on increasing MHEs’ heat transfer performance and reducing the fluid flow pressure drop penalty. Other than water, the use of another commonly available coolant in metal-based MHEs was studied and its flow and heat transfer characteristics were quantified.

It was experimentally demonstrated that building low-profile metal-based MHEs through simple fabrication protocols is feasible. Heat transfer testing was conducted on Cu-based MHEs to characterize device performance. A ~95000W/m²-K surface heat removal capacity was achieved by one of the Cu-based MHEs. The results show the promise of low-profile, Cu-based MHEs for high heat flux cooling applications. A simple two-dimensional FEA simulation was performed and it adequately captured major factors influencing the heat transfer performance of the device. The simple 2D model offers guidance to design optimization.

Adoption of metal-based MHEs in many applications demands quantification of liquid flow and heat transfer performance with application-relevant coolants, e.g. ethylene glycol (EG)/water mixtures rather than pure water. Therefore testing was conducted on Cu-based MHEs with pure EG as the working fluid. Good agreement between data on flow characteristics was obtained from pure EG. Heat transfer testing results were in general agreement with known macroscale correlations, while the observed differences were rationalized.
Transient liquid phase (TLP) bonding of Cu structures with a thin elemental Al intermediate bonding layer was used to assemble Cu-based MHEs. The heterogeneous Cu/Al/Cu TLP bonding interface region, formed during the TLP bonding process, serves as a barrier for heat flow and thus impacts heat transfer performance of assembled MHE devices. To quantify this thermal barrier, transient light flash measurements were performed on separately prepared samples. Results showed that thermal conductivity of the interlayer material could be as low as 1/10 that of pure Cu, and therefore imposes a significant thermal resistance. Heat treatment was conducted on these samples and turned out to be not very effective for reducing the overall interlayer thermal resistance. This study suggests future efforts should be directed toward reducing the total thickness of the bonding interface region while keeping the same bonding structural integrity.

A double-layered MHE was fabricated, assembled, tested, and compared to a corresponding single-layered MHE. Results showed that the double-layered MHE suffers less than half of the fluid flow pressure drop penalty as compared to the single-layered MHE, while exhibiting largely similar heat transfer performance. This suggests a MHE design preference toward multi-layered MHEs, in order to deliver a higher heat transfer performance to fluid flow pressure drop penalty ratio. A double-layered, liquid-to-liquid, counter-flow MHE was manufactured and tested. Results demonstrated that a compact, high-performance, multi-layered, liquid-to-liquid MHE is achievable following the present MHE manufacturing procedures.

A microchannel heat rejection device was built by assembling multiple low-profile microchannel MHEs. Connecting to a heat absorption Cu MHE and a pump, the heat rejection module was integrated into a recirculating-liquid cooling system. Testing
results show that the system performance increases with the performance increase in the heat rejection module, while largely independent of the heat absorption MHE. The reason is because the performance of the absorption MHE is one order of magnitude larger than that of the rejection module. In other words, the absorption MHE delivers better performance than what the system requires and the overall system-level performance is limited by air-solid convective heat transfer occurring at the heat rejection device. These testing results offer guidance to future system design and system-level performance improvements.

An alternative manufacturing method--rolling molding, was developed for low-cost, high-throughput production of high-aspect-ratio, metal-based, microchannel arrays. The roll molding method allows for a streamlined manufacturing of microchannel arrays in sheet metals in a continuous manner, and was demonstrated successfully in the cases of Al and Cu. Normal loading forces and torques were measured during roll molding of annealed Al and Cu, with measurements validated by additional flat rolling experiments conducted on Al strips. The obtained results serve as a basis for future design optimization of patterned roller sleeves, and provide guidance for fabrication of metal-based microchannel arrays by roll molding.

High reliability is much desired in light of wide use of MHEs. There are two main failure modes involved: mechanical failure of bonding interface, and slow corrosion of working fluid into MHE solids. Major future efforts should be directed towards solving these two issues.
REFERENCES


[38] ImageJ is an open access software supplied by the Research Services Branch of National Institutes of Health. Website: http://rsbweb.nih.gov/.


[41] Release 11.0 Documentation for ANSYS.


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