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Experimental investigation and numerical simulation of a copper micro-channel heat exchanger with HFE-7200 working fluid

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EXPERIMENTAL INVESTIGATION AND NUMERICAL SIMULATION OF A COPPER MICRO-CHANNEL HEAT EXCHANGER WITH HFE-7200 WORKING FLUID

by

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A Dissertation Presented in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy

COLLEGE OF ENGINEERING AND SCIENCE LOUISIANA TECH UNIVERSITY

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We hereby recommend that the dissertation prepared under our supervision
by Eric George Borquist
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Experimental Investigation and Numerical Simulation of a Copper
Micro-Channel Heat Exchanger with HFE-7200 Working Fluid
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Ph.D in Engineering

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ABSTRACT

Ever increasing cost and consumption of global energy resources has inspired the development of energy harvesting techniques which increase system efficiency, sustainability, and environmental impact by using waste energy otherwise lost to the surroundings. As part of a larger effort to produce a multi-energy source prototype, this study focused on the fabrication and testing of a waste heat recovery micro-channel heat exchanger. Reducing cost and facility requirements were a priority for potential industry and commercial adoption of such energy harvesting devices. During development of the micro-channel heat exchanger, a new fabrication process using mature technologies was created that reduced cost, time, and required equipment. Testing involved filling the micro-channel heat exchanger with 3M™ Novec™ HFE-7200 working fluid. The working fluid was chosen for appropriate physical and environmental properties for the prototypes intended application. Using a dry heat exchanger as the baseline, the addition of the working fluid proved advantageous by increasing energy output by 8% while decreasing overall device temperatures. Upon successful experimental testing of the physical device, internal operation was determined based on implementation of the lattice Boltzmann method, a physics-based statistical method that actively tracked the phase change occurring in a simulated micro-channel. The simulation demonstrated three primary areas of phase change occurring, surfaces adjacent to where the heat source and heat sink were located and the bulk vapor-liquid
interface, which agreed with initial device design intentions. Condensation film thickness grew to 5\(\mu m\) over the time interval, while the bulk interface tracked from initial 12\(\mu m\) from the lid to 20\(\mu m\) from the lid. Surface tension effects dominating vapor pressure kept the liquid near the heat source; however, the temperature and pressure VLE data suggested vapor interface growth from the heated surface to 5\(\mu m\) above the heated copper plate. Reinforcing the simulation results, including location and movement of phase interfaces, was accomplished through a thorough ten dimensionless number analyses. These specialized ratios indicated dominant fluid and heat transfer behavior including phase change conditions. Thus, fabrication and empirical results for the heat energy harvesting prototype were successful and computational modeling provided understanding of applicable internal system behavior.
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Author  Eric Borquist

Date  07/17/2016
DEDICATION

First and foremost, I would like to dedicate this document and all other academic endeavors to my ever-loving and supportive parents, Nils and Susan Borquist. The dedication to education and the constant quest to explore new challenges was a seed planted early on in my life. I cannot thank them enough for their sacrifice, effort, and inspiration in getting me to the point I am at in life today. Secondly, I dedicate this document to my three siblings, Karl, Kristina, and Nils, who have assisted and supported my lengthy academic career. While not always constructive, you three have consistently steered me back on the path I should not have wavered from. Lastly, I want to dedicate my academic career and this document to my grandfather, Nils Borgkvist, who sadly was never able to capitalize academically on his immense creativity and intelligence in science and engineering due to strict old country traditions. Your natural abilities, coupled with your kind spirit, continually inspire me to push beyond my own expectations.
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The first degree path I chose was Mathematics. Though my family thought I should select a less rigorous program, I was steadfast in pursuing a degree in a difficult yet globally relevant subject. The professors who had the most impact and influence on me were Bernd Schroeder, Ph.D, Weizhong Dai, Ph.D, and Richard Greechie, Ph.D. All three professors were very demanding which I did not agree with initially but now realize it was in my best interest professionally. The inherent complexity of advanced mathematics has benefited my academic and professional career immensely.
by requiring understanding of any problem and designing the most advantageous solution. The progression I have made in time as both an engineer and a scientist stems from my initial development in mathematics.

After graduating in 2006 with my bachelor's degree, I was fortunate to find employment with Ventech Engineers International. My drafting and design capacity with the company required working under some of the most experienced and insightful designers in the United States. Their invaluable advice while working on both civil and piping projects in the process industry opened my eyes to the engineering profession and how to effectively manage time constraints and deadlines. For their help in my development, I would like to thank Stacy Williams, Jerry Young, Cliff Tarver, Andy Brown, and Matt Johnston. The three years that I worked with them greatly assisted me in my engineering endeavors.

Soon after starting employment with Ventech, my superiors asked whether I would be amenable to re-enrolling at Tech for an engineering degree. I began my second undergraduate degree program in 2007 for Chemical Engineering. The previous degree and Ventech prepared me for many of the struggles which ensued in the engineering curriculum. However, the excellent faculty at Louisiana Tech definitely found ways to increase course difficulties and increase student interaction. In particular, Brad Cicciarelli, Ph.D, Daniela Mainardi, Ph.D, Scott Gold, Ph.D, and James Palmer, Ph.D found ways to engage all of the students and ultimately develop better engineers. Over the course of three years in the program, I would also like to acknowledge the help and interaction with fellow students including Jonathan Wheelis, Brian Criswell,
and Ryan Redding. Upon graduating in 2010, I began looking for career options and eventually settled on continuing my education in graduate school.

Wavering in graduate school for various reasons, I found myself enrolled in the Physics Master's program at Tech in 2011 while being hired by Schlumberger Industries in Lafayette, LA to perform offshore Coil Tubing Services. Though I gained experience in personnel management, equipment operations and maintenance, and overall knowledge of the oil industry, the largest benefit from my time at Schlumberger was the friendships developed with all operators, maintenance, and supervisors that I worked with over my year in Lafayette. Thanks Kris Kottingham, Cory Babineaux, Mitch Barras, Guy Edmonds, Toby Falcon, Kirk Girouard, Rodney Green, Joe Harrison, Brent James, Steve Meyer, Carl Patin, Shawn Rabel, Brandon Thibeaux, and Josh Walsh. All of you taught me valuable lessons and kept me laughing while struggling through the twelve hour shifts.

During my experience at Schlumberger, the enrollment in the Louisiana Tech Physics Master’s program inspired the basis for my present academic journey. While mathematics taught me to open my mind and engineering taught me to always approach problems with the entire team, physics taught me to embrace the ambiguity of science and to happily stumble on the path to solution however winding. While I have had classes with many of the faculty, I would like to especially thank Lee Sawyer, Ph.D, Zeno Greenwood, Ph.D, Pedro Derosa, Ph.D, Dentcho Genov, Ph.D, and Klaus Grimm, Ph.D. Physics is a demanding subject, yet the initial struggle belies the extremely good feeling from solving or even better understanding the problem.
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Currently, I am working at Radiance Technologies, Inc. on Louisiana Tech's campus. My supervisor, Dr. Heath Berry, has continually supported me in progressing my academic career. Since we had sophomore English class together in high school, we have crossed paths many times over the past 17 years. Heath and the rest of the Radiance team, most definitely including myself, provide a work environment that can be technically, emotionally, and socially challenging. Cheers!
NOMENCLATURE

Acronyms

\textit{FDM} \hspace{1cm} \text{finite difference method}

\textit{FEM} \hspace{1cm} \text{finite element method}

\textit{ILG} \hspace{1cm} \text{immiscible lattice gas}

\textit{LBM} \hspace{1cm} \text{lattice Boltzmann method}

\textit{LGA} \hspace{1cm} \text{lattice gas automata}

\textit{MHE} \hspace{1cm} \text{micro-channel heat exchanger}

\textit{PDE} \hspace{1cm} \text{partial differential equation}

\textit{TEG} \hspace{1cm} \text{thermoelectric generator}

Constants

\textit{R} \hspace{1cm} \text{gas constant}

\textit{g} \hspace{1cm} \text{gravitational acceleration}

Dimensionless Numbers

\textit{Bo} \hspace{1cm} \text{Bond number}
$Ca$  Capillary number

$Co$  Condensation number

$Fo$  Fourier number

$Ja$  Jakob number

$Nu$  Nusselt number

$Pe$  Peclet number

$Pr$  Prandtl number

$Re$  Reynolds number

$We$  Weber number

**Geometry Properties**

- $A$  cross-section area

- $D_h$  hydraulic diameter

- $d_c$  critical diameter

**Lattice Boltzmann Method**

- $a$  Peng-Robinson equation of state parameter

- $b$  Peng-Robinson equation of state parameter

- $c_s$  lattice sound speed

- $e_i$  particle $i$ velocity
\( f_i \)  
particle distribution function

\( f_i^{(eq)} \)  
equilibrium distribution function

\( g_i \)  
temperature distribution function

\( g_i^{(eq)} \)  
temperature equilibrium distribution function

\( r \)  
position

\( p \)  
momentum

\( t \)  
time

\( \mathbf{V} \)  
maximum velocity

\( \mathbf{x} \)  
position

\( \delta_t \)  
change in time

\( \phi \)  
source term

\( \tau \)  
relaxation time

\( \tau_T \)  
temperature relaxation time

\( \omega_i \)  
weighting coefficients

**Material Properties**

\( k \)  
thermal conductivity

\( c_p \)  
specific heat capacity

\( h \)  
heat transfer coefficient

\( h_{fg} \)  
Heat of vaporization
\( p_i \)  partial vapor pressure

\( T \)  temperature

\( y_i \)  mole fraction of component \( i \)

\( \alpha \)  thermal diffusivity

\( \gamma \)  vapor-liquid interfacial surface tension

\( \mu \)  dynamic viscosity

\( \nu \)  kinematic viscosity

\( \rho \)  density

\( \sigma \)  surface tension

\( \omega \)  acentric factor

**Subscripts**

\( L \)  liquid

\( V \)  vapor

**Units**

\( cP \)  centipoise

\( cSt \)  centistokes

\( g \)  gram

\( J \)  Joule
$K$ Kelvin

$L$ characteristic length

$m$ meter

$mL$ milliliter

$mN$ millinewton

$mol$ molar mass

$P$ pressure

$Pa$ Pascal

$W$ Watt

$\mu L$ microliter

$\mu m$ micrometer

$^\circ C$ Celsius
CHAPTER 1

INTRODUCTION

The ever-growing consumption of finite global energy sources and their environmental cost is constantly encouraging the development of processes and systems that promote increased sustainability, efficiency, and environmental responsibility. Energy harvesting is a process that captures residual fractions of energy that would otherwise be lost to the environment as heat, light, sound, vibration, or movement in order to improve efficiency and encourage new technologies. The miniaturization of modern devices is increasing avenues for waste harvesting possibilities, such that energy harvesting is proposed to potentially replace batteries for smaller, low-power electronics in the future. The advantages and benefits from employing energy harvesting include increased efficiency and sustainability, minimal maintenance time and cost, improved environmental responsibility, and the ability to deploy devices in hostile environments since direct interaction with personnel is not necessary.

While there are several energy harvesting sources, one of the most promising is waste thermal energy recovery due to many standard thermodynamic processes having low efficiencies, whereby much of the thermal energy generated is rejected to the surrounding environment [11]. Many scales of waste thermal energy harvesting have been proposed, from large industry down to the individual user personal computer and
electronics [12, 13, 14]. However, there has been a large research thrust over the past decade into using waste energy for low-power electronics for remote sensing networks and health monitoring of process systems [15, 16, 17, 18]. Harnessing the rejected thermal energy via transduction mechanisms allows existing physical effects to power additional systems, such as remote and hazardous location sensors [19].

The ability to use "affordable and abundant power" while increasing "available power" are two of NASA’s Top Technical and Grand Challenges which require research into waste energy harvesting [20]. As part of a larger research objective, this study reports on the development of waste thermal energy harvesting enhancements pertaining to a copper micro-channel heat exchanger (MHE) with HFE-7200 working fluid. The overall prototype includes waste energy harvesting of vibration, solar, and thermal energies all feeding into a single circuit package to provide multiple avenues of sensor power. However, the necessary thermoelectric generator (TEG) to convert the waste thermal energy to usable electrical power is not the priority topic for the present study. In addition to promoting generator efficiency, there is a need to develop and optimize the heat exchanger incorporated in the harvesting system [21]. The consistent deployment of TEG-based waste heat harvesting devices hinges on the advancement of the associated heat exchanger [14, 21, 22].

MHEs use multiple fluid-filled channels which, at such reduced scales, produce effective heat and mass transfer due to the development of dynamic boundary layers [22]. Maximizing harvesting of waste heat for sensor power with MHEs necessitates optimizing channel fluid dynamics while minimizing thermal resistance [22, 23]. Thermal resistance is reduced using two-phase flows and materials with
high thermal conductivities [22, 23]. Priority for this investigation is the development and understanding of the copper-based MHE fabrication and overall efficiency and the multi-phase flows achieved during operation. The investigation includes design, fabrication, and experimental testing of the MHE as well as numerical simulation of phase change within the MHE and subsequent dimensionless analysis of the vapor-liquid phase interfaces.

The introduction of micro-electronics ushered in a new age of prototyping and manufacturing technologies. As scales decreased down to the micro- and nano-levels, new fabrication methodologies and techniques were established, including development and advancement of processing equipment. Unfortunately, increasing facility capabilities has come at a price. Companies and universities with less funding and purchasing abilities quickly found themselves at a disadvantage while trying to maintain technological pace [24, 25]. However, recent attempts are establishing general laboratory techniques for producing high-quality micro- and nano-scale devices without the requirements of clean-room conditions and equipment constraints [26, 27, 28]. These attempts at general laboratory fabrication methods serve two benefits for furthering research. First, advancements in general laboratory techniques allow less fortunate facilities the ability to continue important research in micro- and nanotechnology. Secondly, the ability to fabricate small-scale devices in a general laboratory environment enables lower cost of manufacturing, thereby allowing production and adoption of such devices on a global scale. There has been no research regarding general laboratory methods for fabricating metal-based MHEs up to this point. Thus, a general laboratory method producing MHEs would further promote the adoption
of waste thermal energy harvesting, increasing the efficiency and sustainability of primary and auxiliary processes.

The potential of MHEs was first suggested by Tuckerman and Pease in 1981 for cooling electronics for very-large-scale integrated (VLSI) circuits [29]. In their seminal article, Tuckerman and Pease announced that by using heat exchangers fabricated at micrometer dimensions, power densities of kilowatts per square centimeter were feasible [29]. While initially conceived for electronics, including MEMS, the widespread embrace of micro-scale devices drove researchers to study the possibilities of micro-scale transport phenomena and how it differed from traditional macro-scale transport processes [30, 31, 32, 33]. The first studies of MHEs examined channel geometry, fluid flow characteristics, and heat transfer characteristics for single-phase flows [34, 35, 36]. However, a shift in research towards multi-phase transport was quickly adopted due to improved transport capabilities of multi-phase devices [35, 36].

Energy transfer using two-phase fluid flows relies on working fluid conversion from liquid to vapor. Devices which take advantage of the latent heat transfer of fluids are more effective than devices which only use sensible heat transfer [37]. Though many of these studies include devices where flow is induced via an external work mechanism, the nature of micro-channels offers a capillary flow mechanism, whereby fluid naturally flows through the channels [36, 38]. Removing the external work mechanism allows the capillary flow-driven MHEs to achieve greater device efficiency and application in remote and hostile environments due to minimal necessary interaction. However, experimental testing is necessary to quantify the capillary flow-driven MHE for practical
application and experimental boundary conditions are required for further numerical analysis of the system's interior operation.

While efficient performance of the MHE is vital to the overall goal of the research, there are internal operations of the closed system MHE which can provide fundamental device performance characteristics. Conducting precise experimental measurements of the internal flow and temperature fields on a scaled MHE is a difficult task. One reason numerical modeling and simulation complements empirical work is by allowing insight into otherwise difficult or impossible measurements. While understood in practice and application, multi-phase flow is not fully understood theoretically. Therefore, numerical modeling and simulation also gives researchers the ability to observe behavior in systems which incur phase change. The traditional method to model the physical fluctuations at the phase change interface is to perform macroscopic discretizations of continuum equations, most notably solving for the Navier-Stokes equations. However, the nonlinearity and constant variations of phases at the interfaces, where different phases are continually combining, separating, and evolving, present difficulties for macroscopic numerical schemes. In the physics discipline of statistical mechanics, it is known that behavior of fluids can be determined through statistical analysis of ensemble systems on the microscopic scale. Though the evolution of complex systems on the microscopic scale can involve multitudes of variables, these systems can be expanded artificially to the "mesoscopic" scale where the physics of the system is maintained.

In 1986, Frisch, Hasslacher, and Pomeau presented a Boolean case for using lattice-based gas cellular automata (LGA) to simulate nonlinear fields such as the
Navier-Stokes equations [39]. This led to the development of a computational method named the lattice Boltzmann method (LBM), based on the continuous Boltzmann equation which has been discretized in time and phase space [40, 41]. When researchers expanded the LBM to macroscopic systems, the Navier-Stokes equations were recovered [39, 41, 42]. Soon after, studies began examining the use of LBM to simulate multi-phase and multi-component systems. Initially, these LBM models incorporated a density distribution function which was then substituted into a given equation of state to provide the pressure function which indicated the location of phase change interfaces. However, when temperature differences play a vital role in the system and boundary conditions, an additional distribution function based on temperature was required. Fusing the density and temperature distribution functions with the chosen equation of state presented numerical difficulties which caused many researchers to test models using normalized parameters, such as the gas constant, and arbitrary equation of state parameters [43, 44, 45, 46]. Unfortunately, for most fluids, this generalization and normalization does not determine the phase change interface within a physical system with a physical working fluid. After experimental testing of the copper MHE was completed, boundary conditions for the actual system, along with HFE-7200 working fluid properties, were numerically modeled for interior operation of the MHE. Once the model was simulated, reinforcement of interior MHE operation was attempted by both the equation of state and dimensionless group analysis.

The extensive literature on MHE fabrication and performance does not include general laboratory fabrication techniques and an analysis of experimental and numerical performance of efficiency and phase change with a non-normalized, real working fluid.
While normalizing data and simulating analytical solutions to mature problems provides mathematical stability properties of a model, the gap in the knowledge arises from a lack of effective test data from a physical, capillary-flow driven MHE operating as a closed system to simulation behavior within the channel from real working fluid phase change. The culmination of testing and simulating this particular MHE and its application in the overall energy harvesting prototype narrows the knowledge gap, allowing for greater understanding of the physical system and potential improvements for MHE application in waste heat energy harvesting.

Incorporation of general laboratory fabrication procedures promotes a much larger number of researchers who can prepare and explore MHE behavior with different working fluids. Development of research methodologies for studying working fluid phase change within the MHE leads to effectively characterizing MHEs for particular applications and determining which working fluid provides the best performance for specific environmental conditions. Therefore, the significance of this study is to initiate investigations into physical working fluids and their behaviors under experimental conditions to produce highly efficient and sustainable heat transfer devices, thereby increasing the adoption of waste heat energy harvesting for varied situations and environments.

Primary research questions for this study are whether a general laboratory procedure can be developed which effectively produces reliable and repeatable MHEs for testing, whether the fabricated MHE can be experimentally shown to be efficient for possible energy harvesting applications, and whether a lattice Boltzmann model of a single channel can provide insight into the working fluid phase change which is
occurring during device operation. In addition to answering these questions directly, further analysis is performed using dimensionless group analysis of the working fluid behavior within a micro-channel to understand the overall dominant forces governing operation and the movement of the vapor-liquid interfaces during device operation.

1.1 Objective and Thesis Structure

One independent study examining the fabrication, experimental testing, and lattice Boltzmann simulation of a copper micro-channel heat exchanger is presented in this dissertation. The first part of the investigation, Chapter 2, focuses on the fabrication of prototype devices of various heights, more importantly in a general laboratory environment free of clean-room constraints and requirements. The fabrication procedure uses low-cost materials and minimal equipment to produce highly repeatable devices. Chapter 3 covers the second part of the study determining experimental performance of the fabricated devices in conditions applicable to the first field testing of the overall prototype. A practical low-energy heat source is used to characterize the MHE efficiency of energy transfer using the HFE-7200 working fluid. Once the device is produced and tested, the experimental boundary conditions and initial conditions are input into the lattice Boltzmann model.

While many publications using lattice Boltzmann are normalized and modeled using standard parameters and fluids including water, this work analyzes the simulated performance of the device with copper boundaries and HFE-7200 working fluid properties. Chapter 4 of the study gives an overview of the lattice Boltzmann procedure, including the fundamental equations, necessary to produce the device
simulation. After demonstrating the thermodynamic properties of the simulation as they evolve over time and the phase change occurring within the channel, Chapter 5 examines pertinent dimensionless numbers of the system and how those numbers increase understanding of system behavior. These dimensionless numbers cover fluid flow, heat transfer, and phase change during device operation. Finally, Chapter 6 presents a general conclusion of the study and offers recommendations for future work.
CHAPTER 2

GENERAL LABORATORY COPPER MICRO-CHANNEL FABRICATION

2.1 Review of General Laboratory Fabrication and Lithography Techniques

The past fifty years have seen the exponential advancement of device and component miniaturization in manufacturing. Spurred mainly by the development of electronics and semiconductor research, many techniques were invented to allow fabrication of increasingly smaller components. Unfortunately, along with the ability to fabricate such technology, the equipment and facilities necessary have progressively increased in complexity and expense. Such expenditures are not possible for all research institutions. However, not all fabrication methods and applications require such stringent laboratory constraints. Micro-scale metal-based waste energy harvesting devices are one such application where potential harsh environments demand robust fabrication methods which do not require large investments in equipment and facilities.

Micro-channel heat exchangers (MHEs) have been researched extensively since Tuckerman and Pease's study on using micro-channels for electronics cooling [29]. These specific heat transfer devices use increased surface area to volume ratios to develop dynamic boundary layers, producing enhanced heat and mass transfer effects [22]. Traditional micro-fabrication is performed with silicon or glass
wafers; however, for heat transfer applications, metal-based MHEs achieve greater performance due to increased thermal conductivities [47, 48]. While there are five broad fabrication classifications for processing micro-channels, most techniques require expensive equipment or cleanroom facilities [49, 50]. Even though these methods are not all applicable for general laboratory adoption, there are alternatives which require minimal equipment and facility infrastructure.

Fabrication of micro-metallic components has been demonstrated for over 25 years, primarily using the LIGA technique [51, 52]. LIGA (German for lithography, galvanofomung (electroplating), and abformung (molding)) is a method where thick photoresists are applied to substrates as molds and filled with electroplated metals [48]. However, traditional LIGA uses X-ray illumination and requires expensive synchrotron radiation facilities for exposure [53]. The UV LIGA technique is an economical alternative which, in addition to costing less in equipment and facilities, is a more expedient process [53, 54]. The molding component of UV LIGA is exposed at near-UV (350-400nm) wavelength to polymerize the photoresist, typically using a negative photoresist such as SU-8 [26, 53, 55]. Traditional photoresists are spin-coated onto substrates; however, in order to fully minimize overhead equipment and facility cost, there are photoresist substitutes to act as the LIGA molding component.

The development and subsequent rapid global expansion of the electronics industry initiated a need for a cost-efficient method to produce printed circuit boards (PCBs). In the 1970s, Riston® negative film was introduced by DuPont® which was applied to a substrate via lamination and exposed using a UV source [52]. Outside of the electronics industry, these dry films have been used for several different applications.
including electroplating molds and microfluidic channels [52, 54, 56, 57, 58]. The advantages of using dry-film photoresist included good conformability and adhesion to any substrate, improved processing speeds and time, excellent thickness uniformity, low cost, and safety conveniences due to simple handling and no solvent/liquid use, all of which are favorable especially for general laboratory environment execution [52, 54, 59].

Exposing dry-film photoresist using an adapted UV LIGA technique requires modifications to standard lithography mask processing which are developed for low cost use in general laboratories. Traditionally, chrome masks are generated for photolithography via e-beam or laser writing [48, 60, 61]. Though resolutions down to 10nm are possible, fabrication of the masks is time-consuming and requires extensive and expensive equipment and facilities [60, 61]. Saving time and money without losing one order of magnitude resolution, microscope projection photolithography, developed in the 1970s, manages resolution down to 500nm using modified laboratory microscopes [60, 62, 63]. However, in order to promote a globally acceptable fabrication method, it must be low-cost for the mask and necessary equipment and allow general laboratory capability without including cleanroom facilities.

Inkjet printers have evolved greatly from the first demonstrated printers developed by Siemens in the 1950s [64]. Advancing from continuous deflection print systems to the widely adopted drop-on-demand method, modern commercial off-the-shelf (COTS) inkjet printers have achieved resolutions of 5760x1440 DPI, corresponding to approximately 4µm and 17µm between dots [64]. Resolutions with such precision have allowed some researchers to develop methods to print the lithography mask with
an inkjet printer [27, 28]. However, further reduction in processing steps and economic cost by using standard office equipment and minimal laboratory equipment is possible which produces metallic micro-structures down to 300\(\mu\)m scales.

A lithographic micro-fabrication method which incurs little cost to a laboratory, allowing any facility the ability to manufacture metallic micro-structures on any flat substrate including metals, is presented here. The process shown demonstrates copper-based MHEs with channel widths of 100\(\mu\)m and 300\(\mu\)m using negative dry-film photoresist. The minimal equipment necessary for this fabrication method makes the micro-fabrication process more accessible for facilities without cleanroom capabilities. After the fabrication process is explained in detail, sample results are given followed by an analysis of process capabilities.

### 2.2 Fabrication Methodology

The fabrication of MHEs from the modified UV LIGA technique concentrates on electroplating copper onto copper sheeting substrates in order to take advantage of copper's high thermal conductivity. Previous studies established an initial device topology, shown in Figure 2.1, which requires electroplating copper from solution onto a copper substrate to produce the reservoir boundaries and micro-channel walls. Using a rigid substrate provides a platform which reduces plating delamination and cracking due to operating conditions and handling. Copper sheeting of 600\(\mu\)m thickness is used for the MHE substrates and lids. Minimizing necessary equipment and fabrication time, Ordyl\textsuperscript{®} dry-film photoresist is used which is produced by Elga Europe\textsuperscript{®}. Initial fabrication testing progressed from Ordyl\textsuperscript{®} P50100 dry-film, 100\(\mu\)m thick film to the
current Ordyl® Alpha 375 dry-film. Alpha 375 dry-film is 75 \mu m thick photoresist film. Processing conditions required are equivalent for both dry films, given in Table 2.1.

![Fluid Reservoirs](image)

Figure 2.1: Initial MHE topology with two reservoirs and 27 micro-channels

Table 2.1: Ordyl® parameters [1]

<table>
<thead>
<tr>
<th>Lamination</th>
<th>Development</th>
<th>Stripping</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$K_2CO_3$</td>
<td>$NaOH$</td>
</tr>
<tr>
<td>1-3m/min (Speed)</td>
<td>0.8% Solution</td>
<td>1-3% Solution</td>
</tr>
<tr>
<td>105 – 125°C</td>
<td>28°C</td>
<td>40 – 60°C</td>
</tr>
<tr>
<td>2.5-6.0 bar</td>
<td>1.5 bar</td>
<td>1.5-4.0 bar</td>
</tr>
</tbody>
</table>

2.2.1 Mold Preparation

Fabricating micro-scale resolution structures without advanced equipment requires pre-processing the copper substrate in order to decontaminate and smooth the substrate for further processing. The copper substrate sheet is initially smoothed with a wire brush and polishing wheel attachments with a Dremel® multi-tool. An acetic
acid solution of 50% concentration at 60°C is then used to clean and decontaminate the substrate. After rinsing with de-ionized (DI) water and dried, the substrate is ready for processing. The first step requires adhesion of the Ordyl® photoresist to the substrate.

According to Elga Europes Ordyl® datasheet, the optimal conditions to apply the dry film photoresist is with either a manual or automatic laminator [1]. The Akiles® Pro-Lam Ultra X-6 automatic laminator allows for roller temperature control and speed control. The laminator is shown in Figure 2.2. In order to facilitate laminating the sample and to ensure that the dry film does not adhere to the hot rollers, the substrate is placed onto a sheet of Teflon© with an additional sheet of printing paper on top. The paper is placed on top of the substrate to help smooth out the photoresist, aiding in removing any wrinkles which occur when the film is placed on the substrate. After laminating the sample, there is a minimum 20 minutes hold time prior to exposing the sample to allow for adequate sample cool-down [1].

![Figure 2.2: Akiles® Pro-Lam Ultra X-6 automatic laminator [6]](image)

There is an important processing objective which bears mention at this point in the process prior to continuing the fabrication profile. Most dry-film photoresists,
including Ordyl®, has a 1:1 ratio of film thickness to maximum resolution. Therefore, for Alpha 375 the film thickness is 75μm and the maximum resolution possible is 75μm. Unfortunately, for 300μm channel widths, 75μm height does not provide ample height to the channel walls, thereby reducing volume of working fluid and decreasing effectiveness of capillaries to facilitate good fluid flow, thus reducing efficiency of the MHE. Therefore, a further study into the possibilities of raising film mold height without losing resolution is justified for appropriate device height.

Though not explicitly discussed or even mentioned in the literature or product datasheets, multiple layers of Ordyl® dry-film photoresist can be applied when required thickness of micro-structures exceeds available dry-film thickness. The dry-film has wrappings on both sides prior to use; however, once the first layer is applied and the substrate cools, the top wrapping of cellophane can be removed and another layer of Ordyl® applied with the laminator. The difficulty in applying multiple layers arises from the natural adhesion of the Ordyl® dry-film. When the top wrapping is removed and another layer applied prior to laminating, the layers adhere to one another causing potential bubbles and inconsistencies in the layers. To reduce this effect, a metal sheet was placed into a freezer to lower its temperature. Before placing the second Ordyl® layer, the substrate's temperature was lowered from contact with the metal sheet. Lowering the temperature briefly reduces the cling of the two Ordyl® sheets and allows several attempts to get the second layer in proper position without adhering to the first layer. While this technique is not necessary for the lithography process, it provides researchers with flexibility regarding available materials and film thicknesses. This
development also extends the capabilities of using dry-film photoresists to previously unheard of heights.

Once the Ordyl® dry-film is laminated onto the substrate, the next step in the process is the UV LIGA exposure which requires the lithographic mask. Searching for a mask technique that requires minimal equipment, standard office transparency sheets are an optimal platform for the mask print. An Epson® Artisan 1430 Inkjet Printer with 5760x1440 DPI resolution is used to print the mask. An axial symmetric rectangular mask pattern is drawn in DipTrace®, a PCB design and schematic freeware. Printing several designs and channel widths to test the printer’s resolution resulted in minimal possible channel widths of 100μm. Reducing scales further increasing the possibility of the mask having alignment issues due to print drops. Therefore, the 300μm channel width MHE design is well within the capabilities of the printer. Figure 2.3 shows the MHE pattern printed on a transparency sheet.

![Figure 2.3: Printed MHE pattern on a transparency sheet](image)

Since the process and dry-film are adapted from PCB use, the light source used is a Kinsten® KVB30D UV Exposure Box, shown in Figure 2.4. Ordyl® specification
for exposing Alpha 375 is 25-30 $\text{mJ/cm}^2$ [1]. For the given MHE pattern geometry, the exposure time was calculated to be 15 seconds per layer of dry film. The key to using a transparency mask directly on the sample is the built-in vacuum unit of the UV exposure box. This minimizes the distance between mask and sample and potential light diffraction which causes over exposure of the film in locations not desirable. One important note is when the sample is removed from the exposure box, there may or may not be indication that the exposure is successful. The key to the process is ensuring that the sample is not over-exposed. Over-exposure causes the subsequent development step to fail due to semi-polymerized resist which is to be removed.

Figure 2.4: Kinsten® KVB30D UV Exposure Box [7]
The recommended developing properties are given in the Ordyl\textsuperscript{©} datasheet [1]. Potassium carbonate solution of 0.8\% concentration is mixed and added to a store purchased garden sprayer. The sprayer technique allows the developer to be applied at proper pressure to remove the unexposed photoresist from narrow channels and structures. Samples are sprayed evenly for 20 second intervals and rinsed in water to prevent over-developing the mold. If over-developed, the polymerized channel walls begin to delaminate and the bulk polymerized film can strip off of the substrate. This process is continued until all of the unexposed photoresist is removed. The samples are then dried with a laboratory fan and examination under a microscope confirms when all of the unexposed photoresist is removed. Figure 2.5 shows a sample of height 75\(\mu\)m thickness after development and prior to electroplating.

![Figure 2.5: Example 75\(\mu\)m mask prior to electroplating](image-url)
2.2.2 Electroplating

Prior work indicates that electropolishing the copper substrate provides an improved and clean surface for electroplating [65]. Polishing the substrate creates a surface with enhanced uniformity, prevents delamination of plating, and allows for repeatability of MHE samples. A phosphoric acid based solution is used for the polishing step due to its widespread use and relatively safe handling when comparing with other possible polishing solutions. Table 2.2 shows the final solution composition and conditions [2]. After testing, samples indicate that the polishing step could be performed before or after the Ordyl® mask processing. However, for this work, the polishing step is performed after the dry-film photoresist has been developed.

Table 2.2: Electropolishing parameters [2]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Working Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H_3PO_4$</td>
<td>157mL</td>
</tr>
<tr>
<td>$H_2O$</td>
<td>75mL</td>
</tr>
<tr>
<td>Al powder</td>
<td>3g</td>
</tr>
<tr>
<td>Current</td>
<td>0.05A</td>
</tr>
<tr>
<td>Temperature</td>
<td>55°C</td>
</tr>
</tbody>
</table>

The mature solution of copper sulfate and sulfuric acid is used for electroplating due to ease of preparation and accounting for chemical hazards. Table 2.3 shows the final solution composition and conditions for plating. Using the literature current density of $0.61 \frac{A}{dm^2}$, the appropriate calculated current for the geometry is 0.67A [3]. However, this current was too large. An example of using large current, causing visually uneven plating, is shown in Figure 2.6. The optimum plating current for
the MHE geometry was experimentally determined to be 0.4A [66]. Figures 2.7 and 2.8 show samples plated at 0.4A with a BK Precision 9130 DC Power Supply. After establishing the initial polishing and plating process, numerous samples prepared prove that the process is reliable and repeatable in a general laboratory. The process flow chart to create the MHE is shown in Figure 2.9.

Table 2.3: Electroplating parameters [3]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Working Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$CuSO_4 \cdot 5H_2O$</td>
<td>188°F</td>
</tr>
<tr>
<td>$H_2SO_4$</td>
<td>74°F</td>
</tr>
<tr>
<td>Current</td>
<td>0.4A</td>
</tr>
<tr>
<td>Temperature</td>
<td>40°C</td>
</tr>
</tbody>
</table>

Figure 2.6: Sample plated with non-optimal 0.5A current
Figure 2.7: Example 75\(\mu\)m mask height channels plated at 0.4A

Figure 2.8: Example 225\(\mu\)m mask height channels plated at 0.4A
Figure 2.9: Preparation steps for the MHE: (a) clean substrate and electropolish if desired; (b) apply dry-film photoresist, single or multi-layers; (c) UV expose the photoresist with appropriate mask geometry; (d) remove unexposed photoresist with developer; (e) electroplate the sample; (f) remove exposed photoresist with stripper leaving the finished device
Additional study on MHE geometry focused on 100μm radial channels. While initially thought to be scalable using the dry-film photoresist, it was quickly determined that fabrication techniques were reduced in repeatability and consistency as the minimum dry film resolution was approached. However, though the 100μm geometry was not used moving forward with the waste heat recovery prototype, some samples were produced. Figure 2.10 shows a radial 100μm geometry MHE after electroplating was completed. The few 100μm samples produced proved that the general laboratory procedure was adaptive to varying MHE dimensions.

![Sample radial MHE with 100μm channel width and 50μm height](image)

Figure 2.10: Sample radial MHE with 100μm channel width and 50μm height

### 2.3 Results and Discussion

Results for the general laboratory MHE fabrication with dry-film photoresist were determined for 300μm channel width, both quantitatively and qualitatively. Tests were conducted to characterize channel wall plating, both for verticality and overall height, plating deposition rate, and to provide SEM image analysis for channel wall
structure. The analyses for all three tests were performed on both the 75μm height film mask samples and the 225μm height film mask samples.

If the dry film over-develops prior to electroplating, the subsequent processing steps lead to channel walls with rounded bases. The change in channel wall geometry from the desired square/rectangular geometry affects both the fluid flow characteristics and the heat transfer characteristics [67, 68]. Therefore, a Dektak 150 Surface Profiler is used to determine channel wall verticality and channel wall height for both plated MHE heights. Figure 2.11 shows a 3mm sample distance for the 75μm mask height samples while Figure 2.12 shows a 3mm sample distance for the 225μm mask height samples. The profile figures both verify the verticality which can be achieved with proper lamination and development processing steps. There is little to no vertical variation until the channel walls exceed the height of the dry-film mask. Note that while the verticality of the channels is maintained during electroplating, the top surfaces of the walls are quite rough, according to the profile. This is primarily due to plating enhancements, such as brighteners and levelers, not being added to the electroplating bath in order for evaluation of the most cost-effective MHE fabrication.
Figure 2.11: Plating profile for 75\(\mu\)m sample

Figure 2.12: Plating profile for 225\(\mu\)m sample
Determination of the plating deposition was crucial for developing processing times which produced repeatable MHE samples of approximately equal height. The calculation for deposition rate was experimentally verified by plating ten samples, five of 75\(\mu\)m mask height and five of 225\(\mu\)m mask height, for different periods of time, namely 20 through 100 minutes, in increments of 20 minutes. Surface profiles of each sample were plotted and measurements of each channel wall height averaged, per sample. Results indicated an approximate deposition rate of \(2.0 - 2.4 \frac{\mu m}{min}\) at the electroplating conditions given in Table 2.3. Sample calculations of 75\(\mu\)m mask height and 225\(\mu\)m mask height are given in Table 2.4. The anisotropic nature of electroplating caused samples to exhibit detrimental properties if deposition exceeded dry film height. These properties stemmed from the channel walls losing verticality and experiencing growth in all directions. This “mushrooming” quality reduced effectiveness of channels and made stripping the photoresist difficult.

Table 2.4: Average deposition rates for 75\(\mu\)m and 225\(\mu\)m samples

<table>
<thead>
<tr>
<th>Mask Height</th>
<th>Plate Time</th>
<th>Ave. Wall Height</th>
<th>Ave. Dep. Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>75(\mu)m</td>
<td>40 min</td>
<td>81.75(\mu)m</td>
<td>2.04 (\frac{\mu m}{min})</td>
</tr>
<tr>
<td>225(\mu)m</td>
<td>100 min</td>
<td>233.6(\mu)m</td>
<td>2.33 (\frac{\mu m}{min})</td>
</tr>
</tbody>
</table>

Once the electroplating parameters were measured and characterized for the MHEs, scanning electron microscope (SEM) image analysis was used to determine the microscopic structure of the plated walls. Images of 120\(\mu\)m mask height samples, given in Figures 2.13 and 2.14, depicted the channel walls having porosity which was estimated at 16% using the SEM. Physical estimation of the porosity was performed using a linear bulk volume measurement. Given the size of the designed and plated
MHE, the fabricated sample was massed and converted to volume using the density of copper. It was calculated that the samples' volumes were 85% of the theoretical volume of the MHE, thus giving a porosity of 15%. The porous structure increased usable device surface area and enhanced contact area for the device working fluid. Though further explained in Chapter 3, the porous result of channel wall fabrication caused wicking action throughout the entire MHE structure.

Figure 2.13: SEM image of channel walls at 500μm scale
Figure 2.14: SEM image of channel walls at 500nm scale
Similar to capillary flow, wicking is dependent on meniscus curvature and is a primary flow mechanism in two-phase flow through porous media [69]. Interconnecting pores vie for fluid flow and a branching competition occurs at every interface throughout the media's complex geometry [69]. During two-phase flow of vapor and liquid, any mixture segregates based on pore size [70]. The separation of phases allows the liquid to give better wettability over the channel surfaces without drying out [70]. Liquid motion throughout the MHE channels and walls provides a continuous surface flow throughout the lattice which is advantageous for energy transfer through the MHE.

2.4 Conclusion

An efficient method of fabricating copper-based MHEs with minimal equipment is established using dry-film photoresist. While dry films have thickness limitations for structures needing sufficient height, such as MHEs, this previously unpublished novel method proves that layering multiple films produces MHE molds over 200μm in height without reducing photoresist resolution. This general laboratory preparation process allows for global adoption of micro-fabrication of structures and devices, even if electroplating is not the desired method for metal deposition. In the case of MHEs, using the mature electroplating process further reduces required overhead expenditures and processing equipment. Though the literature for electroplating is well established, the 100μm and 300μm channel width geometries require altering the suggested literature current densities to produce reproducible samples of approximately equal height and smoothness. While the 100μm samples are possible, the consistency of the 300μm samples far exceeds them; thus the 300μm channel width geometry is
used for the energy harvesting prototype. After fabricating several samples of various channel wall heights, analysis of the reproducibility, deposition rate, and channel wall structure shows that the 300μm process is repeatable with excellent channel wall geometry where the deposited copper provides added wicking potential due to an approximate 15% porosity.

Once the fabrication process is determined, numerous samples of equal 120μm channel wall height are prepared for experimental performance testing of the MHE. The nature of the first prototype for delivery demands an immediately reproducible solution and maintains an adequate aspect ratio of channel width to height. Characterizing the fabricated MHEs experimentally for overall heat transfer efficiency is the objective of the following chapter. Enhancing the heat transfer qualities of MHEs through general laboratory fabrication methods is a key component to the global adoption of such technologies for greater system efficiency and sustainability.
CHAPTER 3

MICRO-CHANNEL HEAT EXCHANGER TESTING FOR
WASTE HEAT RECOVERY EFFICIENCY

Processes which convert energy into usable forms are ubiquitous in everyday life. Unfortunately, many processes in use today are inherently inefficient. Industry is developing techniques, such as cogeneration, which attempt to recover fractions of the expended waste heat for reducing costs and pollution emissions [71, 72]. For smaller scale waste heat recovery, the benefits are reduction of equipment costs, potential independence from batteries, and simple maintenance [16]. Adoption of waste heat recovery techniques for small-scale application therefore requires greater efficiencies than large industry processes like cogeneration since the usable input energy is much lower than the large-scale applications.

Most thermal energy conversion processes require heat exchangers which transfer thermal energy from one fluid to another fluid. However, the process and equipment can be simplified if the aim of the device is not conversion of thermal energy but only transfer of thermal energy. For small-scale waste heat recovery using a thermoelectric generator (TEG), the objective is supplying as much energy to one side of the TEG while keeping the other side from increasing in potential energy. Therefore for the current program, an effective means of transferring energy to either a TEG or
thermal storage unit is desired. Reducing thermal resistances and taking advantage of thermal conductivities for optimizing efficiency, cost benefit, and device robustness encourages the use of metals such as copper. Scaling opportunities for potential device environments and magnitudes of available energy on such small scales for device use drives the approach toward employing micro-channel heat exchangers (MHEs).

3.1 Review of Previous and Current Micro-Channel Heat Exchangers

Compared with conventional heat exchangers, MHEs produce better heat transfer performance primarily due to higher surface area to volume ratios [36, 37]. Scaling effects from the enhanced surface area to volume ratio increases the impact of surface forces and viscous forces while enhancing axial heat conduction at the micro-channel wall [73]. Large heat transfer coefficients, driven by the MHE channel's hydraulic diameter, are achievable with MHEs [33, 74, 75]. Therefore, using an MHE for the small-scale waste heat recovery prototype application provides the best performance with a minimal footprint.

Initially, much of the research on MHEs was conducted with single-phase flow devices at only a few institutions [36]. This was done since fluid flow characteristics through the micro-channels did not always follow classical theory [74]. As researchers learned more about MHE behavior and potential, the larger focus changed from single-phase flows to two-phase flows. The addition of phase change enhanced MHE performance due to larger heat transfer coefficients, reduced coolant flow requirements, reduced thermal resistance, and greater channel temperature uniformity [76, 77, 78]. Two-phase heat transfer occurred when the lower temperature liquid experienced
partial conversion to vapor. While sensible heat transfer was occurring in these systems, the primary heat transfer performance relied on the latent heat exchange [78].

Using two-phase flow MHEs with small hydraulic diameters results in substantial pressure drops across channel lengths [79, 80]. Countering the pressure drop requires many researchers to rely on external work mechanisms such as pumps to provide the necessary fluid flow pressure [22, 81]. Unfortunately for the current program, including additional devices remove essential efficiency for waste heat recovery. Therefore, the current MHE employs capillary flow for fluid motion. Capillary flow is a product of driving forces due to hydrostatic pressure, inertial force of liquid in the channel, inertial force at channel inlet, capillary force, and viscous drag [82]. The study MHE has a reduced hydrostatic pressure due to approximate two-dimensional operation. For axially symmetric channels with laminar flow, Brody calculated an exact analytical solution to flow velocity distribution with a specific boundary condition of zero velocity halfway down the channel [82]. Due to the two-phase flow in the channels, there is also an evaporative factor which causes meniscus deformation and recession as the fluid travels farther from the inlet [83]. The current study MHE shows that the bi-directional fluid flow and symmetry of both reservoir inlets causes minimal meniscus recession along the channel axis.

In addition to capillary flow along the micro-channels, the porous result of channel wall fabrication causes wicking action throughout the entire MHE structure. Similar to capillary flow, wicking is dependent on meniscus curvature and is a primary flow mechanism in two-phase flow through porous media [69]. Interconnecting pores
vie for fluid flow and a branching competition occurs at every interface throughout the media’s complex geometry [69]. During two-phase flow of vapor and liquid, any mixture segregates based on pore size [70]. The separation of phases allows the liquid to give better wettability over the channel surfaces without drying out [70]. The combination of capillary flow and wicking action provides a liquid motion throughout the MHE channels and walls which gives continuous surface flow throughout the lattice, further increasing surface area to volume ratios contributing to larger energy transfer capability in the MHE.

Applications for MHEs have ranged from electronics cooling to harnessing waste energy [29, 84]. Recent research has explored system level waste heat energy harvesting solutions by characterizing heat transfer devices coupled with thermoelectric generators [85, 86, 87, 88, 89, 90]. However, researchers have noted that in addition to improving the conversion efficiency of thermoelectric material itself, there is a parallel need to research and develop advanced associated heat exchangers [91, 92, 93, 94]. In fact, researchers promote the optimization of TEG based systems rather than individual components in order to maximize the output power. The use of advanced heat transfer devices is critical to efficient TEG operation and future deployment [93]. Therefore, while researchers continue to investigate system level solutions for energy harvesting, the optimization of energy transfer via the MHE is the primary priority presented here.

Though current publications have shown developed waste heat recovery processes and techniques from multiple energy sources and industries, including automotive, engines, oil and gas, steel, and steam, focus here is on maximizing energy transfer
via the MHE for any low-temperature waste heat source [95, 96, 97, 98, 99, 100, 101].

Low-temperature heat sources allow the production of power from renewable energy sources, such as solar radiation. Much of the most recent research has been facilitated by developing mid- to large-scale Organic Rankine Cycle (ORC) systems; however, there is still continued interest in the possibilities of micro-scale energy harvesting [102, 103, 104, 105, 106].

The MHE in the current study creates a robust piece of equipment with minimal laboratory equipment while maintaining a high heat transfer capacity. Once made, such a device can be coupled with other devices, such as thermal storage systems. Heat exchanger development, along with materials research, are the cornerstones in developing effective and renewable thermal energy storage [107]. One challenge, when studying the phase change materials (PCMs) which are vital to the thermal energy storage, is that most PCMs have low thermal conductivities leading to low charging and discharging rates [108]. The MHE studied attempts to maximize structural heat transfer, while providing a platform capable for use in varied environments.

The intentional ambiguity of design parameters, in fabrication and in operation, makes the MHE multi-faceted, both in environmental condition and cost. Economic and situational demands may dictate the use of an alternative metal and working fluid. Following fabrication of the MHE systems are evaluated in thermal testing and thermal absorption of simulated heat sources. The MHE is designed for closed system operation whereby no external work mechanism is needed for fluid movement. Thermal and momentum transport are driven by using phase change of a working fluid. The micro-channels drive capillary action of the fluid such that the working
fluid is dispersed over the heat source area, improving heat transfer to the working fluid. The boiling working fluid travels to the MHE copper lid where heat transfer to the heat sink allows condensation to occur and the working fluid returns to either the micro-channels or the reservoirs on the MHE ends. The boiling and condensation process within the MHE is shown in Figure 3.1. This internal action is further explored and detailed in Chapter 4 through unique LBM modeling and validation.

![Figure 3.1: Fluid cycle interaction in the MHE, where each channel is simultaneously boiling and condensing due to the heat source and heat sink](image)

### 3.2 Working Fluid Description

3M™ Novec™ Engineered Fluids have been produced for extreme environment thermal management applications [5]. Based on initial temperature estimates for the intended environment of the MHE, Novec™ HFE-7200 was chosen as an appropriate working fluid candidate. The bulk component of Novec™ HFE-7200, ethoxy-nonafluorobutane ($C_4F_9OC_2H_5$), was designed to replace ozone-depleting substances and to offer favorable environmental properties such as low toxicity and nonflammability [4, 5]. A summary of HFE-7200 environmental and safety properties is given in Table 3.1.
Table 3.1: HFE-7200 environmental and safety properties [4]

<table>
<thead>
<tr>
<th>Properties</th>
<th>HFE-7200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ozone Depletion Potential - ODP</td>
<td>0.00</td>
</tr>
<tr>
<td>Global Warming Potential - GWP</td>
<td>55</td>
</tr>
<tr>
<td>Atmospheric Lifetime (yrs)</td>
<td>0.77</td>
</tr>
<tr>
<td>Flashpoint</td>
<td>None</td>
</tr>
<tr>
<td>Flammability Range in Air</td>
<td>2.4 - 12.4%</td>
</tr>
<tr>
<td>Exposure Guidelines, 8hr TWA (ppm)</td>
<td>200</td>
</tr>
<tr>
<td>Acute Toxicity, 4hr LC₅₀ (ppm)</td>
<td>&gt;92,000</td>
</tr>
</tbody>
</table>

There are three particularly noteworthy properties regarding HFE-7200 fluid properties integrating with MHE operation. First, the low surface tension, even at ambient temperatures, provides a superior wetting agent for the porous micro-channels. Since low surface tension corresponds to larger surface area dispersion, HFE-7200 is an excellent choice for infiltrating the porous channel walls as well as wetting along the channel length. Increasing fluid distribution increases effective surface area for convective heat transfer. The low heat of vaporization enhances vapor saturation and requires less energy to begin the internal MHE cyclical process. Reducing energy input necessary to begin effective heat transfer means less residence time of liquid once the heat source power is input into the MHE, increasing time efficiency of the device. Lastly, the large materials compatibility of HFE-7200 allows optimization for MHE materials of construction [109]. Table 3.2 provides physical properties of HFE-7200, mostly at ambient temperature and pressure. Therefore, depending on the application and intended environment, HFE-7200 can be used regardless of device composition. Initial analysis of working fluid possibilities confirms the appropriate choice of HFE-7200 based on favorable environmental characteristics and favorable physical properties for the MHE application.
Table 3.2: 3M™ thermal management fluid properties [5]

<table>
<thead>
<tr>
<th>Property</th>
<th>Units</th>
<th>Novoc HFE-7200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boiling point</td>
<td>°C</td>
<td>76</td>
</tr>
<tr>
<td>Molecular Weight</td>
<td>g/mol</td>
<td>264</td>
</tr>
<tr>
<td>Critical Temperature</td>
<td>°C</td>
<td>210</td>
</tr>
<tr>
<td>Critical Pressure</td>
<td>MPa</td>
<td>2.01</td>
</tr>
<tr>
<td>Vapor Pressure</td>
<td>kPa</td>
<td>16</td>
</tr>
<tr>
<td>Heat of Vaporization</td>
<td>kJ/kg</td>
<td>119</td>
</tr>
<tr>
<td>Liquid Density</td>
<td>kg/m³</td>
<td>1420</td>
</tr>
<tr>
<td>Coefficient of Expansion</td>
<td>K⁻¹</td>
<td>0.0016</td>
</tr>
<tr>
<td>Kinematic Viscosity</td>
<td>cSt</td>
<td>0.41</td>
</tr>
<tr>
<td>Absolute Viscosity</td>
<td>cP</td>
<td>0.58</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>J/kgK</td>
<td>1220</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>W/mK</td>
<td>0.068</td>
</tr>
<tr>
<td>Surface Tension</td>
<td>mN/m</td>
<td>13.6</td>
</tr>
</tbody>
</table>

3.3 Test Setup

Prior to testing with working fluid, the fabricated MHEs were baselined in a dry condition. Devices were tested with no fluid first and then working fluid was added to the MHEs. The baseline measurement was performed to determine the energy lost to surroundings. The amount of fluid necessary to fill the MHE reservoirs and channels was 150μL based on the device geometry. This value was used to determine an appropriate volume of working fluid for testing. A vapor-liquid interface was desired so the MHE was not filled entirely with working fluid and forced into a pressurized pool boiling operating mode. In this operating mode, the capillary channels would be ineffective. Based on the necessity to have the vapor-liquid interface and condensation at the MHE lid, the volume of working fluid added to the MHE was 120μL, or 80% of total MHE volume. The MHE was designed to have multiple applications in multiple environments. In this work, a low-temperature waste heat environment was targeted.
with temperatures of the MHE below 100°C. Therefore, 3M™ HFE-7200 engineered fluid was selected as the working fluid. The low liquid viscosity increased capillary flow through the channels and the boiling point of 76°C was low enough to provide a testing platform for low-power input applications. Loading of the MHE fluid was performed by manually filling the MHE using a Gilson® Pipetman™ pipette. Following filling of the MHE, the devices were ready for testing. Figure 3.2 shows an example of a finished MHE.

Figure 3.2: Completed MHE sample with lid and working fluid access port

Tests were conducted to characterize energy absorption and transfer of the MHE and temperature profiles of MHE bottom and lid. The fabricated MHEs were tested in a setup depicted in Figure 3.3. Minimizing energy losses to surroundings due to test setup, the testing was performed on a balsa wood platform. Balsa wood has a thermal conductivity only twice that of air but the same order of magnitude, 0.048 $\frac{W}{mK}$ for balsa versus 0.024 $\frac{W}{mK}$ for air. Low thermal conductivities helped reduce unnecessary energy losses into the test setup platform. Omega® k-type bare wire
thermocouples (TC) were used to measure temperature at discrete points on the heater, heat sink, heat flux sensors, and the MHE. Omega® listed the error of each TC as ±0.5°C. The heat source of the test device was provided by an Omega® KHLV-101/10-P resistance heater. The heater was connected to a power supply with a measured 3.6W input power maintained for every test. This power was selected to ensure the bottom of the MHE reached temperatures above the working fluid boiling point in order to cause the required and desired phase change.

![Figure 3.3: Test setup with thermocouple locations shown as yellow circles](image)

To quantify the thermal energy transfer through the setup, an Omega® HFS-4 thin film heat flux sensor (HFS) was placed between each layer as noted in Figure 3.3. Worth noting was a method used to minimize energy loss through the test setup under Heater #2. Heater #1 and HFS #1, from Figure 3.3, were included in order to offset heat output from Heater #2 towards the setup base. The additional HFS underneath Heater #2 was used to minimize energy loss through the balsa base by tracking HFS #1 from Figure 3.3 and maintaining a heat flux measurement of zero, such that Heater #1 acted as a guard heater. This ensured that the majority of output from Heater
#2 was directed towards the MHE. The nominal sensitivity of each HFS-4 has a sensitivity of ±10%, according to Omega®. Each HFS was calibrated and verified by Omega®.

The heat sink used in the test setup was a Xigmatek A10 S80DP, typically used for cooling PC processors. The heat sink had enough thermal mass to hold constant temperature during device testing. A lab jack was used to ensure good thermal contact between the MHE, all sensors and heaters, and the heat sink. As shown in Figure 3.3, this setup was maintained for all tests. Care was taken to maximize thermal contact between setup layers by using Omega® Omegatherm®201 thermal paste. Data was recorded and analyzed via a Lab-View program on a connected PC. The two heat flux sensors (HFS #2 and HFS #3) provided heat flux data which was used to calculate energy transference and losses. The heat flux transfer through the MHE and temperature data allowed characterization of the MHE as described in the Experimental Results section.

Experimental work carries with it an inherent uncertainty when quantifying measurements. For fabrication of the MHE, there are two process steps which have measurable error: printing the MHE mask and the channel wall height due to electroplating. Once fabricated, the MHE experimental testing has three occurring measurement uncertainties: the amount of working fluid added to the MHE with the micro-pipette, instrumentation error due to the heat flux sensors, and instrumentation error due to the thermocouples. Results of the fabrication and testing error are summarized in Table 3.3.
Table 3.3: Error for fabrication and experimental measurements

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Absolute Error</th>
<th>Relative Error</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>MHE Mask Transparency</td>
<td>4μm ; 17μm</td>
<td>1.3%; 5.7%</td>
<td>Based on 5760x1440dpi. Depends on print orientation.</td>
</tr>
<tr>
<td>Dektak Plating Height</td>
<td>3μm</td>
<td>4%; 1.3%</td>
<td>Profiler absolute error. Relative error given for 75um and 225um samples.</td>
</tr>
<tr>
<td>Omega® Thermocouple</td>
<td>0.5°C</td>
<td></td>
<td>Heat source</td>
</tr>
<tr>
<td>Thermocouple</td>
<td></td>
<td>0.5%</td>
<td>Dry MHE bottom plate</td>
</tr>
<tr>
<td>Thermocouple</td>
<td></td>
<td>0.53%</td>
<td>Wet MHE bottom plate</td>
</tr>
<tr>
<td>Thermocouple</td>
<td></td>
<td>0.58%</td>
<td>Heat sink</td>
</tr>
<tr>
<td>Thermocouple</td>
<td></td>
<td>0.78%</td>
<td>Dry MHE top plate</td>
</tr>
<tr>
<td>Thermocouple</td>
<td></td>
<td>0.9%</td>
<td>Wet MHE top plate</td>
</tr>
<tr>
<td>Thermocouple</td>
<td></td>
<td>1.4%</td>
<td>Heat sink</td>
</tr>
<tr>
<td>Omega® Heat Flux Sensor</td>
<td>10% sensitivity</td>
<td></td>
<td>Based on Omega® product sheet</td>
</tr>
<tr>
<td>Heat flux sensor</td>
<td>0.261μV/W/m²</td>
<td>0.002%</td>
<td>HFS-4 (#15120163)</td>
</tr>
<tr>
<td>Heat flux sensor</td>
<td>0.061μV/W/m²</td>
<td>0.0004%</td>
<td>HFS-4 (#14120030)</td>
</tr>
<tr>
<td>Heat flux sensor</td>
<td>0.161μV/W/m²</td>
<td>0.001%</td>
<td>HFS-4 (#14030129)</td>
</tr>
<tr>
<td>Micro-pipette</td>
<td>1μL</td>
<td>0.83%</td>
<td>Working fluid added to MHE</td>
</tr>
</tbody>
</table>

3.4 Results and Discussion

Tests were conducted to characterize the operation of the dry copper-based MHE versus a working fluid wetted MHE. The thermal testing reported how added thermal energy to the device altered heat flux and temperature measurements in order to characterize heat transfer and energy transference of the device. This was utilized in subsequent model evaluation and validation work.

Though the given energy input to the heater was 3.6W, resistive losses reduced energy input supplied to the MHE to 3.44W. This corresponded to a heat flux entering the MHE of $8.7\text{ kW/m}^2$. For the dry MHE, the output heat flux, measured at the interface between top MHE surface and heat sink, was $7.6\text{ kW/m}^2$ while the wetted MHE output
was $8.3\text{ kW/m}^2$, depicted in Figure 3.4. The percentage of input flux which exited through the MHE was 87.5% for the dry MHE and 95.4% for the wetted MHE. Through the use of the working fluid, energy transfer through the device was increased by approximately 8%. Energy rejected from the sides of the MHE was assumed negligible versus the overall thermal energy passing from source to sink. This was based on the operating temperatures, heat flux, and side cooling that was restricted to radiation or free convection. Direct calculation of the side losses for both the dry and wetted MHE tests yielded less than 1% of the total input power to the device. Remaining input which was lost to the surrounding environment was the result of energy absorbed by the copper MHE due to the increase in experimental temperature, energy lost through the MHE bottom plate into the balsa wood base, energy absorbed by the working fluid and vapor due to increases in temperature, and the energy absorbed by the working fluid during phase change. The heat flux summary of input, output, and side losses is given in Table 3.4.
Figure 3.4: Heat flux data from dry and wetted tests

Table 3.4: Energy input, output, and side losses for dry and wetted tests

<table>
<thead>
<tr>
<th></th>
<th>Dry Test</th>
<th>Wetted Test</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Heat Flux</td>
<td>$8.7 \frac{kW}{m^2}$ (100%)</td>
<td>$8.7 \frac{kW}{m^2}$ (100%)</td>
</tr>
<tr>
<td>Output Heat Flux</td>
<td>$7.6 \frac{kW}{m^2}$ (87.5%)</td>
<td>$8.3 \frac{kW}{m^2}$ (95.4%)</td>
</tr>
<tr>
<td>(Measured)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat Flux Lost</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Through Sides</td>
<td>$0.08 \frac{kW}{m^2}$ (0.92%)</td>
<td>$0.05 \frac{kW}{m^2}$ (0.57%)</td>
</tr>
</tbody>
</table>
The temperature profile of the dry and wetted MHE also yielded interesting results. The working fluid acted as a heat sink, which lowered the measured thermocouple temperatures an average of $9^\circ C$ except for the actual heat sink and heat source. Both the dry and wetted MHE testing yielded a heat sink temperature of $37^\circ C$ while the heat source maintained a temperature of $109^\circ C$. While the temperature difference between MHE bottom plate and lid for dry and wetted tests remains at approximately $35^\circ C$, the reduction in temperature due to added working fluid increases possibilities for adjusting working fluids dependent on MHE application. Use of TEGs in power generation requires the largest temperature possible on the TEG “hot side”. However, the more efficient transfer of energy due to the addition of working fluid provides steady energy throughput to the TEG with less continued waste heat loss to the environment. The temperature data for the dry and wetted MHE tests is shown in Figure 3.5.

![Temperature data from dry and wetted tests](image.png)
The underlying key to these tests was the contribution to efficiency through the use of the working fluid within the MHE. In the dry tests, there was only air in addition to the copper channels between the top and bottom which reduced the thermal conductivity and the energy transference of the MHE. However, after adding working fluid, the pocket was filled 80% with working fluid. The vapor-liquid interface allowed greater conductivity but also introduced phase change. The capillary action of the micro-channels pulled liquid over the heat source. Vaporization of fluid caused micro-channels to pull additional liquid and the temperature of the MHE lid indicated condensation of working fluid back to the channels and reservoirs to begin the cycle anew as shown in Figure 3.1. Subsequent modeling work examined this behavior in greater detail in Chapter 4.

Capillary and wicking action allowed working fluid to inundate the entire channel system, not just the capillary micro-channels. Liquid distribution in the MHE is primarily driven through the bi-directionality capillary flow in the main channels; however, because the channel walls are porous, there is also liquid distribution throughout the walls that act as a sponge. This behavior was previously noted using nickel-based porous channels [23]. Though the system is closed, this structure promoted a highly interactive vapor-liquid interface which increased overall heat transfer of the device.

3.5 Conclusion

The testing of the fabricated copper-based MHE yielded several important results. Most important, addition of working fluid increased energy transfer through
the MHE by 8%. The low profile height of the MHE minimized thermal losses through MHE sides. Though the addition of working fluid lowered overall temperatures of MHE surfaces, the overall benefit of more energy throughput was greater than additional increases in temperature supplied for TEG power output.

Using the low thermal conductivity of air and balsa wood, the test setup reduced potential energy losses through the test apparatus. Additionally, the introduction of the guard heater allowed for the reduction of input energy not releasing into the MHE. This is not vital for field testing but it gives a better indication of device efficiency due to decreasing potential alternate routes for energy input.
CHAPTER 4

NUMERICAL SIMULATION OF A MICRO-CHANNEL WITH THE LATTICE BOLTZMANN METHOD

Traditionally, teams of engineers and scientists produce experimental testing prototypes in order to progress toward a finalized equipment design. The development of computer systems over the past sixty years, exponentially increasing in speed and performance, has driven the process of design to shift towards numerical modeling. Computational simulation increases the speed of product or process development while reducing overhead costs [110]. However, unlike strict experimenting and prototyping, numerical simulation of physical phenomena requires mathematical descriptions [111]. Governing equations must be selected and then the modeler must adopt or devise methods to solve such equations [111]. The development of these solving techniques has become ubiquitous in science and engineering, especially for developing computer simulations of complex physical phenomena such as two-phase flow.

The mature numerical methods of mesh discretization such as the Finite Difference Method (FDM) and the Finite Element Method (FEM) attempt to solve partial differential equations (PDEs) for successively smaller meshes in order to converge on a solution. Unfortunately, for systems with multi-phase flow and potential moving phase interfaces, these traditional numerical schemes have difficulty converging...
towards a solution. The FDM uses structured meshes to solve differential equations, where derivatives are approximated with difference equations [112]. Two major issues with the FDM are difficulty of solution convergence due to discontinuities at the phase interface and complex geometries producing regions of low resolution [112, 113]. While the FEM uses unstructured meshes, which can help the resolution issue, there is still a challenge where PDEs fail to approximate solutions at interface discontinuities [113]. For a system such as the micro-channel heat exchanger (MHE), the inherent behavior and operation introduces a potentially active vapor-liquid interface which must be tracked to fully understand device operation. In order to successfully implement a numerical procedure for such a transient system, a physics-based computational approach called the lattice Boltzmann method (LBM) is used for more efficient system simulation.

4.1 Review of Lattice Boltzmann Method

The development and acceleration of modeling and simulation over the past thirty years has led to considerable advancement in understanding nonlinear dynamic systems. Though many methods exist for numerically solving these systems of equations, researchers have developed specific methods which reduce computational cost and time. One such idea is the lattice gas automata (LGA), originally meant to solve incompressible Navier-Stokes equations for fluid flows [114]. These LGA models are described by discretizing phase space and time where particles only operate on the lattice and control of collisions is by specified rules [114, 115]. Lattice sites then obtain values of 0 and 1, giving the term "Boolean molecules" [114]. While this
allows for simulations to exhibit macroscopic thermodynamic equilibrium, there are serious challenges with LGA models [114]. Most prominent are that the models can be difficult to extend to multiple dimensions for complex flows and the Boolean nature gives rise to an excessive statistical noise [115, 116].

The next progression moving from the LGA model was the conversion of LGA from Boolean operators, which can only follow evolution of each individual molecule, to continuous real numbers between 0 and 1 which then allows for one-particle distribution functions [115, 117]. While the continuous approach removed fluctuations due to ensemble averages, there was one primary problem with the Boltzmann equation (BE) approach and the LGA [115, 117]. The lack of Galilean invariance due to the lattice discretization caused scaling issues with multi-phase flows due to fluid interface velocities constrained by conservation of particles equaling average particle velocity which did not have the same scaling [114, 115, 118]. A subsequent lattice Boltzmann model, presented by Gunstensen et al., was designed for immiscible two-phase flows and included a Galilean invariant collision operator [116]. This suggested lattice Boltzmann model simulates surface tension at the interface using a perturbation of the linear collision operator which forces the pressure tensor to be locally anisotropic at the interface [116]. This model used an existing immiscible lattice-gas (ILG) model from Rothman and Keller with a modification to the ILG collision operator with a two-step approach: first, they added the perturbation creating the correct surface tension, and second, they kept the immiscibility by not allowing mass diffusivity from one fluid to another [116]. While the perturbed model allowed for easier extension to multiple dimensions, the difficulty in interface tracking of multi-phase flows led to
multiple lattice Boltzmann methods which attempted to rectify these non-ideal fluid difficulties [119].

Transport phenomena seen in nature is a complex and dynamic behavior. As a collection of microsystems, these phenomena can be considered statistical probabilities of the microsystem ensemble evolution. Unfortunately, it is impractical to attempt calculating properties of such systems due to involvement of extreme degrees of freedom. After development of initial LBM models, two models, one based on particle interaction potential and the other on free energy, accomplish conservation constraints and effectively contain non-ideal fluid behavior leading to proper fluid flow simulation [42, 119, 120]. The interaction potential model, by Shan and Chen, contributes a potential to the collision term which adds either attractive or repulsive energy to every collision [42]. Swift et al. suggest a model which specifies collision term rules such that the system develops toward the minimum of a non-local free-energy function [120]. While both models show promise, there are potential inconsistencies in the interaction potential model leading to inaccurate system property calculations and the free energy model initially does not satisfy Galilean invariance when the phase space is discretized [42, 120]. While all the given models provided initial fluid flow phenomena simulation, none of the models included temperature variations in the system. Newer research is incorporating such phase-change transition behavior due to both fluid flow characteristics and energy transfer within the system.

Thermal lattice Boltzmann models can be given in three broad categories: multi-speed approach, passive scalar approach, and the He et al. model which includes the internal energy density distribution function [121]. Macroscopic properties of
any system use the same distribution function as the multi-speed approach [121]. The variation in speed is due to the necessity to maintain mass, momentum, and kinetic energy for every collision [121, 122]. Since there are multiple speeds, the higher order velocity terms tend to introduce numerical instability [121, 123]. The passive scalar approach uses two distribution functions: one for velocity and density and a second scalar function for temperature [121, 124]. While numerical stability is better for passive scalar than for multi-speed, the scalar function for temperature does not affect flow but does progress through the flow field due to advection [121, 123, 124].

The model given by He et al. introduces an internal energy density distribution function, derived from the Boltzmann equation [121, 125]. While the He et al. model is an effective model for real thermal problems, the introduction of the complex gradient operator appearing in the evolution equation eliminates the simplicity of the LBM [121, 126]. The coupling of distribution functions for density and temperature provides challenges to practical models, the introduction of an equation of state allows simulations of phase transition, both single component and multi-component.

The use of LBM to simulate phase transitions started with the Shan and Chen pseudo-potential model [42, 127]. Soon after, Yuan applied a sign function which greatly increased application of the Shan and Chen model [127, 128]. Another model from Zhang introduced a body force term directly into the evolution equation; however, the Zhang model proved more numerically unstable for a smaller temperature range than the Shan and Chen model [127, 129]. As phase transition models progressed, there was a trend towards improving numerical stability and accuracy by incorporating multiple relaxation times and removing the decoupled double distribution function of
flow and temperature fields [46, 130]. In 2012, Gong and Cheng published papers using a new model technique which used a density distribution function and a temperature distribution function [43, 44]. The Peng-Robinson equation of state then used both distribution functions to simulate pressure changes resulting in phase change [43, 44]. Using normalized parameters, Gong and Cheng successfully simulated the coexistence curve of the Peng-Robinson equation of state and bubble growth and release during pool boiling [44].

While previous literature establishes an excellent precedent for moving forward with LBM implementation, there are many additional factors which must be included to produce a working model of the MHE system. The inclusion of all real properties of the HFE-7200 working fluid being the primary focus. Constructing a computationally efficient and realistic model requires using as many experimental and empirically determined parameters as possible. Subsequent sections explore the properties of the working fluid, as needed by the model, a full description of the simulated model, and the results and discussion of the simulation.

The major achievement for the current study simulation is the preservation of all SI units throughout the model. While the majority of LBM literature uses normalized and non-dimensionalized model properties, this is not advantageous for the system under study. Behavior of the physical fluid under MHE operating conditions is desired. Therefore, maintaining all units provides simulation data which is immediately accessible for qualitative feasibility analysis and potential errors in the program. In addition to the LBM phase simulation, Chapter 5 reinforces system behavior using the working fluid physical data equations with dimensionless number analysis. This
additional method of analysis is not possible if system properties are normalized prior to simulation.

4.2 Model Description

Since the MHE system includes multiple phases of working fluid, there is a need to establish both a density distribution function and a temperature distribution function. These two functions fully capture the particle evolution through space and time, allowing for further analysis including equation of state pressure determination and subsequent dimensionless analysis to understand additional system behavior. One key component to ensuring an accurate simulation is the correct model of the working fluid. Therefore, an additional review of HFE-7200 properties is also explored prior to simulation performance setup.

4.2.1 Fundamentals of the LBM

The MHE study involves a bounded fluid system with an external heat source. From initial conditions and time to the steady-state time where working fluid phase interfaces have reached equilibrium, the internal fluid system is continually undergoing evolution of density, temperature, and pressure, allowing for the vapor-liquid interface tracking. While non-equilibrium statistical mechanics attempts to characterize thermodynamic systems which are out of equilibrium, it is not necessary to integrate equations of motion for all constituent particles [131]. For situations such as the MHE with external forces and constraints, local equilibrium throughout the system can be determined from the mean free path and collision time. Assume a function $f$ of point particles, $f(r, p, t)d^3r d^3p$, defined as the number of particles with
position within $d^3r$ of $r$ and momenta within $d^3p$ of $p$ of time $t$ [132]. Differing from semi-classical mechanics, when collisions are included and particles with momenta collide and convert to conserved momenta, the collision integral is given by:

$$
\frac{\partial f}{\partial t} + \frac{dr}{dt} \cdot \frac{\partial f}{\partial r} + \frac{dp}{dt} \cdot \frac{\partial f}{\partial p} = \left( \frac{\partial f}{\partial t} \right)_{\text{collision}}.
$$

(4.1)

Rearranging the terms, the Boltzmann equation can be written as

$$
\frac{\partial f}{\partial t} = \left( \frac{\partial f}{\partial t} \right)_{\text{stream}} + \left( \frac{\partial f}{\partial t} \right)_{\text{collision}}
$$

(4.2)

such that

$$
\left( \frac{\partial f}{\partial t} \right)_{\text{stream}} = -\frac{dr}{dt} \frac{\partial f}{\partial r} - \frac{dp}{dt} \frac{\partial f}{\partial p}
$$

(4.3)

is the streaming term. Therefore, there are two contributions to particle motion over time: streaming and collision [131].

The LBM describes a system by these evolutions of the particle distribution functions streaming and colliding on a lattice network. Initializing the velocity of particles requires the implementation of the Chapman-Enskog distribution for non-equilibrium systems as well as a discretized collision operator for the lattice network, such as the traditional Bathnagar-Gross-Krook (BGK) model [133, 134]. The density distribution function, $f$, evolves using the traditional BGK collision operator, as given in Equation 4.4. A corresponding equilibrium distribution function relates the relaxation of the system, which is given in Equation 4.5. The last term in Equation 4.4
introduces a body force term as presented by Gong and Cheng and given in Equation 4.6 [44]. This force term includes three forces which change associated velocity terms during collisions: the interparticle interaction force, the interaction force between surfaces and liquids, and the gravitational force [44].

\[
f_i(x + e_i \delta t, t + \delta t) - f_i(x, t) = -\frac{1}{\tau}(f_i(x, t) - f_i^{eq}(x, t)) + \Delta f_i(x, t) \tag{4.4}
\]

\[
f_i^{eq} = \omega_i \rho \left[1 + \frac{e_i \cdot u}{c_s^2} + \frac{(e_i \cdot u)^2}{c_s^2} - \frac{u^2}{2c_s^2}\right] \tag{4.5}
\]

\[
\Delta f_i(x, t) = f_i^{eq}(\rho(x, t), u + \Delta u) - f_i^{eq}(\rho(x, t), u) \tag{4.6}
\]

where

\[
\Delta u = F \cdot \delta t / \rho \tag{4.7}
\]

is the velocity change due to body force during the time step. The force term, \(F\), is given by

\[
F = F_{int}(x) + F_s(x) + F_g(x). \tag{4.8}
\]

Prior to describing the temperature distribution function, there are some notation and scheme clarifications which must be made. While there are several different LBM schemes based on system dimensionality and potential locations in the
lattice available for particle movement, the proposed model uses the D2Q9 lattice scheme due to system symmetry. The D2Q9 scheme is shorthand for two dimensional, nine discrete velocities within the momentum discretization [45]. Figure 4.1 depicts the velocities for the D2Q9 scheme. The nine potential velocities, $e_i$, each has an associated probability, $\omega_i$, for a particle’s likelihood to move to that lattice node. Equation 4.9 gives the probabilities for the D2Q9 scheme.

\[
\omega_i = \begin{cases} 
\frac{4}{9}, & i = 0 \\
\frac{1}{9}, & i = 1,2,3,4 \\
\frac{1}{36}, & i = 5,6,7,8 
\end{cases}
\tag{4.9}
\]

The nine velocity vectors are given by Equation 4.10. Note that $m = \frac{\delta_x}{\delta_t}$ where $\delta_x \equiv$ lattice spacing and $\delta_t \equiv$ time spacing, which is based on the particular lattice scheme chosen. For the MHE study, both the lattice and time spacing are equal to
1\mu m and 1sec, respectively. Specific mention must be given to the two relaxation times, \( \tau \) and \( \tau_T \). For LBM simulations, the fluid kinematic viscosity, \( \nu \), is defined as in Equation 4.11 while the thermal diffusivity, \( \alpha \), is defined as in Equation 4.12. Since the working fluid has physical properties which can be experimentally tested, the relaxation times can be empirically determined from the fluid data in the following subsection.

\[
e_i = \begin{cases} 
(0,0), & i=0 \\
(\pm 1,0)m,(0, \pm 1)m, & i=1,2,3,4 \\
(\pm 1, \pm 1)m, & i=5,6,7,8
\end{cases} 
\quad (4.10)
\]

\[
\nu = c_s^2 \left( \tau - \frac{1}{2} \right) \delta_t 
\quad (4.11)
\]

\[
\alpha = c_s^2 \left( \tau_T - \frac{1}{2} \right) \delta_t 
\quad (4.12)
\]

Analogous to the density distribution function evolution equation, the temperature distribution function, \( g \), has an evolution equation given by Equation 4.13. The equilibrium function \( g^{eq}_i \), given by Equation 4.14, is very similar to the density equilibrium function \( f^{eq}_i \). The main difference between the two distribution functions is the last term of each equation. While the density evolution incorporates a body force term \( \Delta f_i(x,t) \), the temperature evolution uses an energy source term given in Equation 4.15. The weighting coefficients for both set of distribution functions are based on the statistical probabilities in Equation 4.9.
\[ g_i(x + e_i \delta_i, t + \delta_i) - g_i(x, t) = -\frac{1}{\tau_T} (g_i(x, t) - g_i^{eq}(x, t)) + \delta_i \omega_i \phi \] (4.13)

\[ g_i^{eq} = \omega_i T \left[ 1 + \frac{e_i \cdot U}{c_s^2} + \left( \frac{e_i \cdot U}{c_s^2} \right)^2 - \frac{U^2}{2c_s^2} \right] \] (4.14)

\[ \phi = T \left[ 1 - \frac{1}{\rho \gamma} \left( \frac{\partial P}{\partial T} \right) \right] \nabla \cdot U \] (4.15)

The density and temperature distribution functions allow calculation of the pressure field which provides the vapor-liquid interface motion via an equation of state (EOS). The Peng-Robinson EOS is chosen for its stability and accuracy [43, 44]. The Peng-Robinson EOS is

\[ P = \rho RT \left( 1 - b \rho \right) - \frac{a \rho^2 \epsilon(T)}{1 + 2b \rho - b^2 \rho^2} \] (4.16)

such that

\[ \epsilon(T) = \left[ 1 + \left( 0.37464 + 1.54226 \omega - 0.26992 \omega^2 \right) \left( 1 - \frac{T}{T_c} \right) \right]^2, \] (4.17)

where \( \omega \) is the acentric factor. From the density, temperature, and pressure fields, the internal operation of the MHE channel can be analyzed for behavior. However, attempting to simultaneously solve for the density and temperature fields can lead to convergence difficulties and model instabilities. This presents challenges using normalized and ideal fluid data. One option to remove the model abstraction employs
using all available physical data for the working fluid. Prior to implementing the LBM, the following subsection explores potential opportunities for optimizing and streamlining computational cost and simulation time by using HFE-7200 physical properties.

4.2.2 Working Fluid Model Properties

The environmental benefits and operational advantages of using 3M® HFE-7200 as a working fluid is explained in Chapter 3. For the numerical model, there is available information of specific working fluid properties. Unfortunately, much of the manufacturer published literature is at room temperature [4]. Since the MHE system is experiencing transient density, temperature, and pressure fields prior to reaching steady-state conditions, much of the available OEM data is not applicable for the simulation. However, there is some manufacturer published data coupled with several researchers studying HFE-7200 that is of great benefit for the simulation. Two equations given in the 3M® HFE-7200 datasheet correlate density and temperature, allowing transformation of the temperature field data into density field data [4]. Since there is a direct linear equation associating temperature $T$ and density $\rho$, the difficulty in fusing two distribution functions for solving simultaneously is simplified to include just density distribution data. As the external heat source is applied, changes in density $\rho$ correspond to changes in temperature $T$. The 3M® equations are given as Equations 4.18 and 4.19.

$$\ln P = 22.289 - 3752.1 \left( \frac{1}{T} \right)$$  \hspace{1cm} (4.18)
\[ \rho = 1.4811 - 0.0023026(T - 273.15) \]  \hspace{1cm} (4.19)

Though the density-temperature relationship is vital to the simulation of phase change, there are other working fluid properties which must be included with the simulation, including dynamic viscosity \( \mu \), surface tension \( \sigma \), thermal diffusivity \( \alpha \), and the acentric factor \( \omega \). These are necessary for determining the two relaxation times \( \tau \) and \( \tau_T \), part of the body force term, and the Peng-Robinson EOS calculation. Upon personally corresponding with 3M® engineers, the acentric factor was identified as \( \omega = 0.459 \) for HFE-7200. Experimental data points for dynamic viscosity \( \mu \) and surface tension \( \sigma \), recently published, are shown in Figures 4.2 and 4.3 [8]. In order to simulate continually changing dynamic viscosity \( \mu \) and surface tension \( \sigma \), trendlines are added to the published data for programmable equations, based on the working fluid temperature.
Similarly to dynamic viscosity and surface tension, thermal conductivity $k$ and specific heat capacity $c_p$ experimental data is used to establish a trendline equation for continual temperature ranges, which is then programmed into the simulation.
Figures 4.4 and 4.5 show the data with trendline equations. Thermal diffusivity $\alpha$ is determined using Equation 4.20 with the density array determined by the LBM. Using the thermal conductivity and specific heat equations, the thermal diffusivity equations programmed into the simulation, for both liquid and vapor, are shown in Figure 4.6.

\[
\alpha = \frac{k}{\rho c_p} \quad (4.20)
\]
Figure 4.5: Empirical and modeled equation for specific heat $c_p$ [10]

Figure 4.6: Calculated points and modeled equations for thermal diffusivity $\alpha$
4.3 Performance Setup

Prior to performing the simulation, the program which solves the LBM equations needs to be selected. Given the objectives of observing the density, temperature, and pressure fields and how the arrays develop over time, Matlab® is used for the LBM simulation. The first step in implementing the LBM is the determination of model lattice size, boundary conditions of the channel, and initial conditions for the working fluid in the channel.

The physical micro-channel has dimensions of 20mm length, 300μm width, and 120μm height. Therefore, the ratio of length to height is 166:1. While this ratio simulates conditions along the entire channel, symmetry and bi-directionality of fluid motion due to capillary action allows the physical ratio of the program to run at half of the channel size. Thus, the initial ratio of the program is 83:1. The height of the simulation was then halved such that LY (the length of the y-axis) is set to 60μm for scaling and reducing the computational power needed to perform the simulation, effectively producing a model having 2μm of physical dimension per μm of the simulation dimension. Therefore, the horizontal lattice length is set to 4800μm to give proper dimensionality of the simulated channel. However, once again, due to channel axial symmetry and bi-directionality of fluid flow in the channel, the horizontal lattice is reduced to efficiently conduct the simulation. After determining the model lattice size, the boundary conditions of the simulation are established.

The micro-channel is bounded on top and bottom by copper plates. These two boundaries influence both the momentum gradient and the thermal gradient along the channel length. Since the fluid particles are constrained within the channel,
a traditional bounce-back condition is implemented which causes particle motion to return from the lattice fringe back into the lattice network. The bounce-back condition is commonly referred to as the no slip boundary condition in macro-scale calculations [135]. The copper plates are modeled in the lattice network as boundaries occurring between lattice nodes, effectively ceasing particle motion in the boundary direction, causing the particle's momentum to return the particle to the node from which it arrived [135]. Thermal boundary conditions for the copper plates is set to the experimental conditions of the heat source and the heat sink. Temperature data from the experimental testing fixed the bottom plate temperature at 85°C and the top plate temperature at 50°C for simulation. Given the previous discussion of channel geometry and symmetries, the vertical boundaries of the lattice network are programmed for periodic boundary conditions whereby the fluid flow, assuming one direction, particles reappear on the adjacent vertical boundary. Establishing lattice geometry and boundary conditions for the channel prior to saturation as shown in Figure 4.7, initializing fluid and air conditions must then be calculated to determine initial fluid motion and composition of air/fluid mixture due to evaporative properties of the working fluid.
Figure 4.7: Boundary and initial conditions prior to saturation

Product information for 3M® Novec® HFE-7200 fluid included Equations 4.18 and 4.19 which provided correlation between vapor pressure, density, and temperature [4]. Maintaining the experimental setup parameters, 80% of the channel is filled with working fluid and 20% is air at room temperature. Thus, the inner array has initial densities of 1.43 $\frac{g}{mL}$ for the working fluid and 0.0012 $\frac{g}{mL}$ for dry air. Immediately after inserting working fluid into the MHE, the working fluid began evaporating until the dry air was saturated. In order to fully understand the system equilibrium initial conditions with saturated vapor and bulk working fluid, Raoult’s Law for a single condensable species was used which is given by Equation 4.21,
where \( p^*_i \) is the vapor pressure of the pure component and \( y_i \) is the mole fraction of component \( i \) in the mixture. Therefore, for the HFE-7200/air system at ambient conditions, the 3M® partial pressure equation, Equation 4.18, was used with Equation 4.21 to calculate molar composition as shown in Equations 4.22 and 4.23.

\[
y_{7200} = \frac{0.163}{1.01325} = 0.1607\frac{mol_{7200}}{mol_{mixture}} \quad (4.22)
\]

\[
y_{\text{air}} = 1 - y_{7200} = 0.8393\frac{mol_{\text{air}}}{mol_{mixture}} \quad (4.23)
\]

Using density properties and channel geometry measurements, the calculation of the initial density condition gave the saturated vapor density as less than the bulk working fluid density, at ambient temperature, by approximately 0.002%. Thus, upon saturating the dry air of the channel, the system was assumed to have relatively uniform initial density. The dry air/working fluid interface tracked downward to provide the initial interior composition conditions of the simulation, shown in Figure 4.8. Regarding initial fluid motion, the horizontal velocity component can be found by measurement of the physical device. Testing provided a flow velocity along the channel length of \( 17 \text{mm} \text{s}^{-1} \). At ambient temperature, 3M® published viscosity allows calculation of the Reynolds number to initially be approximately 700, giving laminar micro-channel fluid flow. Gravity and buoyancy effects were also accounted for in
the flow profile. However, Cheng and Wu published a correlation shown in Equation 4.24 [136]. For the system in question, channel diameter was much less than the equation requires, revealing that gravity effects on the symmetry flow was negligible. Establishing the physical system, boundary and initial conditions, and fluid momentum within the model allows solutions of LBM collisions. This is coupled with the two OEM equations and the equation of state to determine phase transitions within the channel.

\[ d_c = 0.224l_c \] (4.24)
In order to develop the pressure field, the parameters of the Peng-Robinson equation of state needed to be determined. Table 4.1 was generated from calculation and working fluid information from 3M\textsuperscript{®}. Note that all units were maintained due to the requirement that all model results be comparable to the experiment and experimental units. Taking all conditions and the physical system parameters into account, simulations were performed to solve for the phase change interface transitions. The results suggested promising simulations of the phase change regimes which occurred within the micro-channel, reinforcing the fundamental MHE design and initial considerations of internal fluid flows.

Table 4.1: Peng-Robinson EOS parameters

<p>| | |</p>
<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
<td>$a$</td>
<td>3.6705 $\frac{kgm^2}{z mol}$</td>
</tr>
<tr>
<td>$b$</td>
<td>0.00016 $\frac{m^6}{mol}$</td>
</tr>
<tr>
<td>$R$</td>
<td>8.314 $\frac{m^3}{mol Pa} K$</td>
</tr>
<tr>
<td>$T_c$</td>
<td>483.15 $K$</td>
</tr>
<tr>
<td>$P_c$</td>
<td>2.01 $MPa$</td>
</tr>
</tbody>
</table>

4.4 Simulation Results

The initial simulations are plotted in intervals of 40 time steps for the density $\rho$, temperature $T$, and pressure $P$ arrays. The density array at the initial time step show the dry air and bulk working fluid density difference. Time progression of the density as a function of position from the lid is given in Figure 4.9 while the percent change from each time step to the next is given in Figure 4.10. As time steps progress, three interfaces form, one at the lid, one at the bulk vapor-liquid interface, and one at the bottom plate. The time-based progression of the central vapor-liquid interface moves 12$\mu$m from the lid down to approximately 20$\mu$m from the lid in the density
array. After 360 time steps, there is less than 1% change between the densities at time step 320 and step 360, suggesting that the density array had reached steady-state conditions.

Figure 4.9: Time progression for density array

Figure 4.10: Percent change in density array time progression
The temperature array is simulated using Equation 4.19 and the density array. The temperature as a function of position from the lid is plotted in Figure 4.11. Initially, the bulk fluid is at room temperature with the vapor showing a warming trend due to the heat sink remaining at constant temperature and the bulk liquid absorbing initial bottom plate heat source energy. The time progression shows bulk vapor increasing in temperature while the bulk liquid acts as an internal heat sink. While temperature increases near the lid, the density and temperature arrays indicate liquid conditions, indicating that condensation is occurring at the lid. The large increase in temperature at the bottom plate indicates boiling occurring, given density and temperature data from 3M®. Similar to the density array, the largest changes were found in the span between time step 1 and time step 40, as shown in Figure 4.12. After 360 time steps, the temperature array changes less than 1%, indicating steady-state conditions.

![Figure 4.11: Time progression for temperature array](image-url)
Both the density and temperature arrays are used to produce the pressure array from the Peng-Robinson EOS, indicating where the phase change is occurring in the channel profile. Shown in Figure 4.13, the pressure array appears similar in pattern to the density array. The pressure array is the key to determining phase change within the micro-channel. Figure 4.14 reinforces Figures 4.10 and 4.12 suggesting the steadying of conditions after 360 time steps. Curiously, the pressure array does not suggest a developing phase interface at the heat source surface. Taking the temperature array data and the pressure array data, coupled with the working fluid vapor-liquid equilibrium data, Figure 4.15 shows the steady-state plot as a function of position within the channel. The VLE data suggests confirmation of the conclusions that there is liquid condensate from the MHE lid to between 4 – 5\(\mu m\), vapor from 5 – 20\(\mu m\), bulk vapor-liquid interface between 20 – 21\(\mu m\), liquid from 21 – 54\(\mu m\), boiling interface between 54 – 55\(\mu m\), and nucleate vapor from 55\(\mu m\) to the heated copper surface.
The large natural surface tension dominant behavior of fluid flow in micro-channel deters bubble growth as the temperature rises above the boiling point. At the increased pressure shown in the array, the large surface tension increases boiling point
The plot indicates that boiling interface at steady-state is located between 54-55μm from the lid.

The plot indicates that bulk interface at steady-state is located between 20-21μm from the lid.

The plot indicates that condensation at steady-state is located between 4-5μm from the lid.

Figure 4.15: Steady-state VLE plot of LBM data

of the working fluid. Thus, the working fluid absorbs more energy before yielding to boiling. Though the lack of bubble growth decreases introduction of nucleation sites, vapor bubbles that do initiate growth do not coalesce. This phenomena causes an increase in bubble population over time on the heated surface [137]. The process ends with the small bubbles deforming under surface tension and releasing, therefore contributing to the cyclical operation of the MHE [138].

Beyond the array plots, Figures 4.16 and 4.17 depict the time step progression of the pressure array with color maps in MATLAB™. The phase interfaces at the MHE lid and bottom are small in comparison with the bulk vapor-liquid interface, shown in yellow. The white shade indicates conditions where the working fluid is increasing in temperature past the boiling point. The orange and light red shades indicate conditions where liquid condensate is interacting with the bulk vapor.
Figure 4.16: Color map of pressure at time $t = 0$ in the micro-channel

Figure 4.17: Color map of pressure at steady-state in the micro-channel
The simulated cyclical operation of the MHE provides reinforcement that the experimental device is operating effectively as designed. Given the conditions of phase change at the heat source and heat sink and the large tracking vapor-liquid interface, the model reassures the operation of the experimental MHE as having a real internal fluid-vapor-fluid loop. Phase change verification by experimental working fluid data confirms the LBM simulation. For final color map comparison, Figure 4.18 shows both color maps side-by-side. The three arrows indicate the movement of the interfaces over time.
Figure 4.18: Side-by-side comparison of the initial and steady-state pressure color maps. The arrows indicate the three phase interfaces and their movement over time.
4.5 Conclusion

The development of a simulated copper micro-channel filled with working fluid began with looking for an approach which could successfully track phase change in the channel while being computationally efficient for implementation on a desktop or laptop personal computer. Recent statistical approaches for computational fluid dynamics led to performing the numerical solution to simulation with the lattice Boltzmann method. The implementation of the LBM provided an excellent avenue for determining internal operation approximations for the prototyped MHE.

Separate from much of the literature which normalized and non-dimensionalized system properties and conditions, the approach in this study prioritized the preservation of all required parameters. While this process introduced potential unit conversion difficulties initially, the overall result of the simulation showed immediately availability for feasibility analysis. While this process differed from much of the literature, it proved a viable alternative if modeling a physical system.

Information from 3M® allowed for the simulated results to be analyzed. The density, temperature, and pressure arrays showed that there is movement of the bulk vapor-liquid interface from approximately 12µm to 20µm and there is development of boiling and condensing interfaces at the heat source and heat sink surfaces. Prior to concluding successful operation of the MHE via the simulated model, further analysis was conducted with dimensionless numbers, only possible due to the inclusion of physical data and units, to provide conclusions for simulation-based channel fluid flow and heat transfer behavior.
Chapter 4 established the development and execution of the lattice Boltzmann simulated copper micro-channel with HFE-7200 working fluid. The pressure field was calculated from the Peng-Robinson equation of state using the dual distribution functions of the density and temperature fields. While the pressure field indicated the phase change occurring in the channel, secondary equations exist which can complement the pressure field data by reinforcing understanding of system behavior and performance, namely dimensionless numbers.

The history of dimensional analysis begins with such luminaries as Newton, Euler, Fourier, Maxwell, and Stokes [139, 140]. However, it was not until Osborne Reynolds popularized one of George Stokes' concepts, later introduced as the Reynolds number by Sommerfeld in 1908, on fluid flow patterns that led to an abundant development of dimensionless numbers in the late nineteenth and early twentieth centuries [140, 141, 142, 143, 144]. Dimensionless numbers, specific experimental instances of dimensional analysis, are mathematical expressions typically composed of either products or ratios with dimensions of unity [139, 145]. Though largely used in engineering, especially chemical engineering, dimensionless numbers exist in most engineering and science disciplines [139, 146, 147]. Most occur as coefficients when
non-dimensionalizing conservation laws and constitutive equations [139]. However, dimensionless numbers are predominant in transport phenomena, a field of study in engineering where momentum, energy, and mass transfer are analyzed.

All necessary physical properties and equations of the LBM simulation are intentionally incorporated in the model without normalization or dimensional analysis to the copper wall or HFE-7200 working fluid, thereby removing ambiguity from model implementation and subsequent simulations. Therefore, all resulting system characteristics, from viscosity to equation of state parameters to pressure, are kept in SI units which benefits the immediate determination of model feasibility in the physical world. However, there are several reasons for examining dimensionless numbers and their influence on understanding system behavior and performance. Dimensionless numbers can reduce the number and complexity of variables affecting a system [139, 145]. Scaling different sized models is simplified with dimensionless numbers, producing effective methods for researchers to advance from a model to a prototype with less difficulty [139, 145]. Most importantly for the current study, dimensionless numbers indicate dominant transport processes which govern the behavior of the system, providing a clearer physical understanding of phenomena within the system and predicting vapor-liquid interface movement [145]. Due to the enormity of dimensionless numbers in the literature, the current micro-channel system with free surface working fluid is examined with specific numbers pertinent to the physical and simulated system.

Investigating two-phase flows in micro-channels with free surfaces, there are ten dimensionless numbers which must be studied for greater system transport behavioral understanding: Reynolds number (Re), Bond number (Bo), Weber number (We),
Capillary number (Ca), Prandtl number (Pr), Nusselt number (Nu), Peclet number (Pe), Condensation number (Co), Fourier number (Fo), and Jakob number (Ja) [74, 139, 148, 149, 150, 151, 152]. These ten numbers cover specific energy and momentum transport characterization of the system while including vapor-liquid interface analysis. In addition to providing additional system behavior, movement and position of the phase interfaces, first shown with the LBM simulation, can be reinforced through the dimensionless group investigation.

5.1 Fluid Flow Analysis

5.1.1 Reynolds Number

Reynolds number may be the most ubiquitous dimensionless number, especially in fluid mechanics. It specifically relates the inertial forces to viscous forces pertaining to fluid flow within a channel; it also gives an indication of whether fluid flow is laminar or turbulent based strictly on its numeric magnitude. The typical equation for Reynolds number is

\[ Re = \frac{\rho V L}{\mu}. \]  \hspace{1cm} (5.1)

However, for flow in a pipe or tube, Reynolds number is a ratio of density \( \rho \), fluid velocity \( V \), hydraulic diameter \( D_H \), and dynamic viscosity \( \mu \),

\[ Re = \frac{\rho V D_H}{\mu}, \]  \hspace{1cm} (5.2)

where the hydraulic diameter is given by
\[ D_H = \frac{4A}{P}. \] (5.3)

The parameter \( A \) is the cross-sectional area of the pipe or tube and \( P \) is known as the wetted perimeter. For the simulated copper channel, Figure 5.1 gives the geometry of these parameters.

\[ A = H \times W \]
\[ P = h + h + W + W = 2h + 2W \]

Figure 5.1: Parameters for determining \( D_H \) in Reynolds number

Laminar flow occurs when a fluid flows in parallel layers. At low velocities, the fluid tends to flow without lateral mixing. In this flow regime, particle motion of the fluid is very orderly with all particles moving in straight lines parallel to the pipe walls. Laminar flow occurs when \( Re \leq 2040 \). Figure 5.2 shows the progression of the Reynolds number over time with respect to the channel height.

For the simulated micro-channel, since \( Re \leq 450 \), flow is laminar. Such low \( Re \) implies large viscous forces compared to the inertial forces. While inertial forces
are weak overall for the given \( Re \), there is one location where inertial forces are the strongest, namely the bulk liquid. This is a reasonable result given liquid velocity in the channel due to capillary action. There are three locations of interest based on Figure 5.2: system boundary phase interfaces with the top and bottom copper plate and the development of the phase interfaces between vapor and liquid. At \( t = 1 \), there is definite separation of the air and working fluid. The small deviations at the boundaries are the result of immediate temperature differences, leading to immediate density differences, at the source and sink. As time progresses, note the saturation of air by the increasing \( Re \). Immediately after \( t = 1 \) and progressing until steady-state is reached, two disjointed locations begin to develop. These are the locations of liquid-vapor interfaces, the development of a film condensation layer near the copper boundary lid and the bulk interface occurring in the channel. Similarly to the color
map plots in Chapter 4, these phase interfaces move over the time steps agreeing with the pressure array plot and color map of the Peng-Robinson EOS.

Analyzing the three phase change interfaces enlarged at their respective scales reveals potential behavior of fluid phase interaction. Figure 5.3 explores the fluid behavior near the MHE lid by expanding the view of Figure 5.2 for all time step simulations at the location of the disjointed region, namely from $1\mu m$ to $5\mu m$ from the MHE lid. Initially at $t = 1$, the slope of $Re$ is negative with an extremely low $Re$, decreasing from 66 to near 0 for the bulk air in the channel. However, as time progresses, $Re$ increases meaning inducement of flow due to increasing density. Initially, the MHE lid surface temperature is causing condensation to occur which is apparent by the negative slopes from $t = 1$ to $t = 80$. However, between $t = 120$ and $t = 200$, there is a change from negative to positive slope. At this point the condensing film is thickening which encourages the development of boundary flow where the relative velocity of the fluid layers decreases as the surface is approached. Interestingly, at certain points, the $Re$ decreases but corresponds to an increase in the distance from the MHE lid, suggesting an influx of saturated vapor which increases the thickness of the condensing film on the MHE lid.
Increasing film thickness from the MHE lid

Increasing condensation shown by increasing Re

![Graph showing Reynolds number progression for condensing vapor interface near the MHE lid.]

Figure 5.3: Reynolds number progression for condensing vapor interface near the MHE lid

The color map plot of Chapter 4 indicates that the bulk vapor-liquid interface increases in distance from the MHE lid over the time progression. At $t = 1$, there is a clear delineation of the air and working fluid in the channel. However, for analyzing the interface over time, the slope at the interface assists in indicating fluid behavior between phases. Figure 5.4 shows the changing slope of the three median points of the interface location. From $t = 1$ to $t = 40$, the phase interface transitions from a positive slope to a local saddle point which continues until steady-state is reached. The continual form of each successive time step suggests the development of a continuously evolving interface with mixing of vapor and liquid contributing to the local saddle point while moving away from the MHE lid until steady-state conditions are reached. The vapor-liquid interface begins to steady in behavior starting at $t = 200$ with each successive time step remaining in the same location with slightly reducing Re.
values. From the simulation, it appears that the bulk vapor-liquid interface for the experimental device settles at approximately 40\(\mu m\) from the lid.

![Graph showing Reynolds number progression for the bulk vapor-liquid interface in the MHE channel.](image)

**Figure 5.4:** Reynolds number progression for the bulk vapor-liquid interface in the MHE channel

The final Re location for further analysis is located at the bottom where the heat source is applied. At \(t = 1\), the heat source has only initiated heat transfer to the working fluid, as depicted in Figure 5.5 by the large slope directly at the MHE bottom surface. However, as time steps progress, more thermal energy is absorbed into the working fluid, contributing to the reduction of Re at the MHE bottom surface due to potential locations of nucleate boiling. Similarly to Figure 5.3, the boiling behavior acts in the following fashion. As more heat is absorbed by the working fluid, bubbles begin to form at local boiling locations causing Re to decrease due to vapor introduction into the bulk fluid velocity profile. Continual heat flux at the surface prompts further localized boiling to the point where some locations merge to form
pool boiling pockets, creating larger areas of vaporization which expands toward the MHE lid.

Based on the simulation results, including the density array and dynamic viscosity, the \( Re \) appears to reinforce the interior operation of the MHE agreeing with the arrays and color maps in Chapter 4. However, before stating that MHE interior operation simulation converges toward a singular ideal, other dimensionless numbers must be compared for system behavior concordance.

### 5.1.2 Bond Number

The Bond number, \( Bo \), gives the ratio of body forces to surface tension forces using both vapor and liquid densities \( \rho_V \) and \( \rho_L \), gravitational acceleration \( g \), characteristic length \( L \), and surface tension \( \sigma \). The equation for \( Bo \) is
The $Bo$ for the channel simulation is given in Figure 5.6. First, note the continuously decreasing $Bo$. Due to the numerator containing the difference of liquid and vapor densities, the $Bo$ at $t = 1$ has the largest difference. The gravity term and characteristic length are constants so they do not contribute to the changing $Bo$. However, the surface tension in the denominator contributes greatly to the $Bo$ of the system. As the densities converge due to saturation, the increasing temperature of the system causes surface tension to begin decreasing. From Figure 5.6, the surface tension is decreasing slower than the density difference causing the $Bo$ to decrease overall. Regarding the magnitude of the $Bo < 1$, this indicates that surface tension dominates body forces in the channel.

\[
Bo = \frac{(\rho_L - \rho_V)gL^2}{\sigma}.
\]

Figure 5.6: Bond number as a function of time for the simulated MHE channel
5.1.3 Weber Number

The determination so far suggests viscous forces dominating the \( Re \) and surface tension forces dominating the \( Bo \). The Weber number, \( We \), comparing inertial forces to surface tension forces with density \( \rho \), fluid velocity \( V \), characteristic length \( L \), and surface tension \( \sigma \), is given by

\[
We = \frac{\rho V^2 L}{\sigma}.
\] (5.5)

Based on the \( Re \) and \( Bo \), surface tension forces should dominate leading to a \( We \) of small magnitude. Figure 5.7 shows the \( We \) for the MHE channel as a function of time and position. The magnitude of the \( We \) axis for Figure 5.7 immediately indicates that surface tension dominates inertial forces. There are three locations in Figure 5.7 which correspond directly with the \( Re \) plot, the boundary interfaces and the bulk vapor-liquid interface. Comparing Figure 5.2 and Figure 5.7, the development of the three interfaces occurs at exactly the same distances from the MHE lid. This is particularly noteworthy since the equations for \( Re \) and \( We \) only have density and velocity in common. Comparing Equations 5.2 and 5.5 and removing like terms and constants, such as velocity and characteristic length, reveals magnitudes on the order of

\[
\frac{1}{\mu} \quad \text{and} \quad \frac{1}{\sigma}.
\] (5.6)
Since the viscosity and surface tension are programmed from fitted experimental data points, shown in Figure 4.2, and including the magnitude of both Figures 5.2 and 5.7, surface tension dominates viscosity for the MHE simulation.

![Figure 5.7: Weber number as a function of time and position for the simulated MHE channel](image)

### 5.1.4 Capillary Number

The last direct dimensionless number ratio compared for fluid flow is the viscous forces to the surface tension forces, given by the Capillary number, $Ca$. The $Ca$, comparing dynamic viscosity $\mu$, fluid velocity $V$, and interfacial surface tension $\gamma$, is given by

$$Ca = \frac{\mu V}{\gamma}. \quad (5.7)$$
During simulations this variable was isolated and calculated at each time step interval. This was done by running the simulation for the given time step and calculating the surface tension from the temperature using the trendline from Chapter 4, whereby the particular cell from the temperature array was determined by the three arrays from Chapter 4. $Ca$ as a function of time is shown in Figure 5.8. Note the magnitude of the $y$-axis. Since $Ca \approx 10^{-5}$ order of magnitude and the inherent porous nature of the MHE fabrication shown in Chapter 2, flow is dominated by capillary forces with negligible viscous forces contributing. Thus, the magnitude of the $Ca$ reinforces the $Re$, $Bo$, and $We$ numbers comparison by indication of surface tension dominance.

![Capillary number as a function of time for the three interface regions](image)

Figure 5.8: Capillary number as a function of time for the three interface regions

5.1.5 Fluid Flow Conclusion

The fluid flow dimensionless numbers, Reynolds, Bond, Weber, and Capillary, all point toward surface tension forces dominating fluid flow motion in the channel.
While the Bond and Capillary numbers are plotted as functions of time only, the Reynolds and Weber numbers both converge on the appearance of three regions of interest where phase change occurs. While it is intuitive that these regions exist due to boundary conditions and initial conditions, the direct comparison presents an interesting behavior requiring further analysis. In order to understand internal behavior of the MHE through simulation, heat transfer analysis must also be performed.

5.2 Heat Transfer Analysis

5.2.1 Prandtl Number

Moving directly from fluid flow to heat transfer analysis, a dimensionless number which compares the two is beneficial to determine which transport phenomena dominates in the MHE system. The Prandtl number, $Pr$, is a ratio of momentum diffusivity and thermal diffusivity comparing specific heat $c_p$, dynamic viscosity $\mu$, and fluid thermal conductivity $k$. It is given by Equation 5.8.

$$ Pr = \frac{c_p\mu}{k} $$

(5.8)

Figure 5.9 depicts $Pr$ for the MHE channel. All three variables for $Pr$ are discussed in the Chapter 4 section covering the working fluid modeling. Note the immediate similarity of Figure 5.9 and Figure 5.7. Interestingly, $Pr$ is only dependent on fluid characteristics, whereas $We$ has dependence on system channel geometry as well. However, just as Figure 5.2 and Figure 5.7 before, Figure 5.9 agrees on the three interface regions in exactly the same location relative to the MHE lid.
Figure 5.9: Prandtl number as a function of time and position for the simulated MHE channel

In addition to the comparison with previous dimensionless figures, the magnitude of $Pr$ indicates greater thermal diffusivity than momentum diffusivity throughout the channel. Since the fluid flow is laminar with low velocity, the bulk vapor and liquid components exhibit the smallest momentum diffusivity to thermal diffusivity. However, at the phase interfaces, the molecular motion due to phase interaction contributes to a larger influence by momentum diffusivity. The region which is larger in Figure 5.9 than Figure 5.2 and Figure 5.7 is the liquid/vapor behavior near the heat source. This can be explained strictly by the temperature array and how it affects both liquid and vapor. Even if the liquid is not boiling near the heat source, the temperature of the liquid through the MHE depth causes molecules to move more freely, allowing increased momentum transfer. Figure 5.10 shows the $Pr$ for the three interfaces as a function of time. Based on the simulation, thermal diffusivity is 6.86
times more influential than momentum diffusivity for the MHE lid and bottom at \( t = 1 \) whereas the thermal diffusivity is 5.66 times more influential than momentum diffusivity for the bulk vapor-liquid interface at \( t = 1 \). As steady-state conditions are reached, thermal diffusivity maintains dominance over momentum diffusivity for all three regions, though the influence diminishes to 5.46 for the bottom, 4.93 for the lid, and 4.87 for the bulk interface.

![Figure 5.10: Prandtl number as a function of time for the three interface regions](image)

5.2.2 Nusselt Number

The Nusselt number, \( \textit{Nu} \), provides the ratio of convective to conductive heat transfer occurring when there is heat transfer across a boundary. Since there are two boundaries supplying and removing heat, the \( \textit{Nu} \) is simulated for the entire channel depth. \( \textit{Nu} \) is typically given by comparing heat transfer coefficient \( h \), characteristic length \( L \), and fluid thermal conductivity \( k \) shown in Equation 5.9.
However, based on experimental testing, there is constant wall heat flux at both the source and sink surfaces. Constant heat flux at the boundary surfaces causes the energy equation can be reduced, combined with Fourier’s law of conduction, and solved for the heat transfer coefficient. The reduction yields \( h = \frac{4.364}{L} \) which therefore yields \( Nu = 4.364 \). Using the fluid flow conditions and the constant heat flux boundary condition, Shah and London published a correlation for thermally developing, hydrodynamically developed laminar flow, \( Re < 2300 \), where

\[
Nu = 4.364 + 0.0722 \left( Re Pr \frac{D_h}{L} \right), \quad Re Pr \frac{D_h}{L} < 33.3
\]  \hspace{1cm} (5.10)

which was used in the simulation for \( Nu \) [153]. Figure 5.11 displays the simulated results of the Shah correlation. The magnitude of \( Nu \) indicates convection is at least 4.364 times more influential on heat transfer than conduction for the entire channel. This is logical since air and working fluid have such low thermal conductivities even though fluid velocity in the channel is low. The behavior of Figure 5.11 is interesting at the bulk vapor-liquid interface, exhibiting multiple saddle points at most time step intervals. Therefore, the key locations of the bulk interface are shown in Figure 5.12 only as a function of time. Note that over time, the \( Nu \) slope begins to settle toward 0 meaning that the interface is steadying between 4.36503 and 4.36509 and the interface composition has mixed to become equal vapor and liquid.
Figure 5.11: Nusselt number as a function of time and position for the simulated MHE channel

Figure 5.12: Nusselt number as a function of time for the bulk vapor-liquid interface
5.2.3 Peclet Number

Based on the importance of convection in the simulation, given by the $Nu$, a further look into components of convection is beneficial to understanding transport phenomena in the channel. Convection is a combination of advective transport and diffusive transport. These two quantities are compared, with characteristic length $L$, fluid velocity $V$, and thermal diffusivity $\alpha$, using the Peclet number, $Pe$, given by Equation 5.11.

$$ Pe = \frac{LV}{\alpha} \quad (5.11) $$

$Pe$ compares the advective transport rate to the diffusive transport rate. Figure 5.13 shows the $Pe$ as a function of time and position in the channel. The magnitude of $Pe$ indicates large dependence on advection to the overall convection. However, at $t = 1$, the diffusive transport is dominant in the bulk vapor. This particular location and time experiences the largest diffusion of working fluid into the air due to large molecular movement down the concentration gradient. As time steps progress and the vapor becomes saturated, advection begins to supercede diffusion as the dominating transport mechanism.
Figure 5.13: Peclet number as a function of time and position for the simulated MHE channel

5.2.4 Condensation Number

The Condensation number, $C_0$, gives the ratio of number of molecules condensing on a surface compared to the total number of molecules striking the surface [154]. For the MHE simulation, the surface which experiences condensation is the MHE lid. The $C_0$ is determined, using heat transfer coefficient $h$, fluid thermal conductivity $k$, dynamic viscosity $\mu$, density $\rho$, and gravitational acceleration $g$, by the equation

$$C_0 = \frac{h}{k} \left( \frac{\mu^2}{\rho g} \right)^{\frac{1}{3}}, \quad (5.12)$$

which is different depending on the situation and geometry [155, 156]. Figure 5.14 shows how the MHE develops a steadying condensation film on the lid surface. The figure indicates the largest surface condensation occurring at early time steps. Due
to vapor saturation of working fluid, this initial data appears logical given the heat sink location and temperature of the lid. Over the time interval reaching steady-state conditions, note that $C_0 \to 0.215$. Therefore, roughly one-fifth of molecules striking the MHE lid, and as time progresses the condensate film, are condensing and contributing to the condensate film. It must be noted however that while a percentage of the vapor is condensing, the phase interface experiences constant molecules in the condensate film vaporizing into the saturated vapor region.

![Condensation number as a function of time for the simulated MHE lid](image)

Figure 5.14: Condensation number as a function of time for the simulated MHE lid

5.2.5 Fourier Number

The final heat transfer dimensionless number considered for the MHE channel is the Fourier number, $F_0$. The $F_0$ compares thermal diffusion transport rate the thermal storage rate of the medium with thermal diffusivity $\alpha$, characteristic time $t$, and characteristic length $L$. $F_0$ is determined by Equation 5.13.
\[ Fo = \frac{\alpha t}{L^2} \]  

(5.13)

As a function of time and position, \( Fo \) is shown in Figure 5.15. Note the overall difference in behavior versus all of the other heat transfer dimensionless plots. The overall magnitude of \( Fo \) shows that thermal diffusion dominates thermal storage, especially in the bulk liquid region. The interesting behavior occurs at the three phase interface locations. The lower \( Fo \) corresponding to the phase interaction at the MHE lid, bulk vapor-liquid interface, and MHE bottom demonstrates the energy storage via the continual phase interactions at the interfaces. For example, at the bulk vapor-liquid interface there is ever-evolving interaction between vapor condensing and liquid evaporating establishing a phase equilibrium. Therefore, some of the energy input into the system remains at the interface location causing the thermal storage rate to be larger than in the bulk vapor and liquid regions. Similarly for the MHE lid and bottom, there is a reduction in diffusion influence which indicates multi-phase interaction.
5.2.6 Heat Transfer Conclusion

The findings from the five heat transfer dimensionless numbers are: thermal diffusivity dominates momentum diffusivity, convection is the dominant transfer mechanism, and the three phase interface regions appear to increase thermal storage rates versus the bulk vapor and liquid regions. Considering the combination of convection, advection supplies the bulk heat transfer. Lastly, the MHE lid, kept at reduced temperatures with the heat sink, steadies at approximately 21% of molecules which contact the lid then accumulate in the condensate film. Overall, the heat transfer numbers agree with the fluid flow numbers, even when heat transfer numbers rely only on fluid characteristics. Time and position agreement reinforces conclusions reached by the fluid flow analysis concerning the three two-phase regions and the three interface movements during simulation.
5.3 Phase Change Analysis

5.3.1 Jakob Number

The Jakob number, $Ja$, is a rarely encountered dimensionless number, yet it is vital for direct contact processes that involve phase change [157]. For processes which involve boiling, $Ja$ gives the ratio of maximum sensible heat absorbed by the liquid to the latent heat absorbed by the vapor [158]. Condensing processes are characterized by the ratio of maximum sensible heat absorbed by the vapor to the latent heat absorbed by the liquid [158]. $Ja$ uses specific heat of liquid $c_{p,l}$, liquid density $p_l$, temperature difference $\Delta T$, heat of vaporization $h_{fg}$, and vapor density $\rho_v$. $Ja$ is given by Equation 5.14 [159, 160].

$$Ja = \frac{c_{p,l}p_l\Delta T}{h_{fg}\rho_v} \quad (5.14)$$

Since $Ja$ shows the amount of heat transfer occurring during phase change, Figure 5.16 give the time progression for $Ja$ at the MHE lid, MHE bottom, and the bulk vapor-liquid interface. In agreement with the surface tension dominance of the fluid flow analysis, $Ja \leq 16$ means that bubble departure at the MHE bottom is controlled strictly by surface tension forces [159]. The lid interface converges to 0.77. Therefore, approximately 44% of interface energy is absorbed via sensible heat through the vapor while 56% of the interface energy contributes to phase change via latent heat absorption by the liquid once steady-state conditions are reached. The boiling occurring at the MHE bottom converges to approximately 0.6. Conversely to the MHE lid, this means that 38% of interface energy is absorbed via sensible heat.
through the liquid and 62% of the energy is absorbed via latent heat through the vapor phase. The bulk interface converges to approximately 0.78. Similar in quantity to the MHE lid, 44% of the interface energy is absorbed via sensible heat while 56% contributes to phase change of both phases.

Figure 5.16: Jakob number as a function of time for the simulated MHE lid

5.3.2 Phase Change Conclusion

The use of the Jakob number demonstrates that the three regions experiencing multi-phase interaction have larger energy contributions to latent heat absorption. This indicates energy absorption via a constant temperature process, further reiterating phase change at that specific location. Therefore, the LBM simulation of the MHE channel supports the idea of interior operation of the MHE developing a boiling region and a condensing region where effective operation occurs as a continual cycling of working fluid phase change.
5.4 Conclusion

Analysis of fluid flow, heat transfer, and phase change in the simulated MHE channel via dimensionless numbers yields several interesting results. The primary behavior noted is that every dimensionless number plotted as a function of time and position agreed on the development and movement of the three phase interfaces, regardless of variable and system characteristic dependence. The overall result of the dimensionless number analysis is that channel behavior can be summarized as laminar, capillary-driven fluid flow, bulk heat transfer via convection effects, and effective boiling and condensing on both surfaces in contact with the heat source and heat sink, also summarized in Table 5.1.

The density, temperature, and pressure arrays are used in Chapter 4 to characterize phase change and internal operation of the MHE channel simulation. However, the collected analysis via dimensionless numbers, which is initially used to verify interior condition and behavior, is also verification of channel operation. While these numbers indicate the system characteristics they are intended to determine, the collectivized agreement of all numbers open potential avenues for future use of dimensionless numbers in simulation verification.

Table 5.1: Summary of MHE simulation behavior

<table>
<thead>
<tr>
<th>Transport Mechanism</th>
<th>MHE Channel Behavior</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid Flow</td>
<td>Laminar Flow Regime</td>
</tr>
<tr>
<td></td>
<td>Surface Tension Dominant</td>
</tr>
<tr>
<td>Heat Transfer</td>
<td>Thermal Diffusion Dominant</td>
</tr>
<tr>
<td></td>
<td>Convection Dominant</td>
</tr>
<tr>
<td></td>
<td>Thermal Sinks at Phase Interfaces</td>
</tr>
<tr>
<td>Phase Change</td>
<td>Latent Heat Greater Than Sensible Heat</td>
</tr>
<tr>
<td></td>
<td>Absorption at Phase Interfaces</td>
</tr>
</tbody>
</table>
CHAPTER 6

CONCLUSION

The present study set out to determine an efficient and low-cost fabrication method for the production of a copper-based micro-channel heat exchanger to recover waste thermal energy. In addition to the physical prototype required by the NASA proposal which provided funding for the development, the investigation sought to validate internal operation with a numerical model using the lattice Boltzmann method. Further dimensionless number analysis reinforced results from the numerical simulation. The importance of this study, after providing NASA with a working prototype which powers remote sensors with multiple energy scavenging mechanisms, allowed the creation of a simulation which used physical properties of system fluids without normalization thereby giving results which were immediately available for comparison against working fluid experimental data. Waste heat recovery has been researched to take advantage of thermodynamic inefficiencies, allowing for reduced operational costs, increased operational efficiency, and increased environmental sustainability. The study successfully accomplished all objectives including the creation of a cost-effective MHE fabrication technique, development of an energy efficient waste heat recovery device, and the creation of the numerical model using a statistical approach which allowed for efficient phase interface tracking during MHE operation.
Specific findings were chapter specific and were summarized with the respective chapters. However, there were results which bear reiterating at the end of the study. One of the first aims of the study was determining whether cost-effective general laboratory techniques were possible allowing the fabrication of the MHE. The technique developed, previously not found in literature, promoted an unexplored use of dry-film photoresist where film layers were stacked allowing micro-structure fabrication with greater heights and aspect ratios. Layering dry-films also provided an avenue for using low-cost office equipment and a power supply to produce repeatable samples of MHEs, shown for heights over 200μm. Therefore, successful fabrication of micro-structures was deemed available to any facility with general laboratory capabilities. While the low profile of a copper-based MHE transferred 87% of energy without using any working fluid, the use of a working fluid increased energy transference to over 95%. Heat transfer was enhanced in the device by the inherent porous nature of channel wall fabrication, increasing surface area by 18%. The overall effectiveness of the MHE resulted in progressive prototypes produced for the NASA modular energy harvesting device.

Having fulfilled the fabrication and experimental prototype aspect of the study, the focus changed from overall device operation to internal operation and phase change occurring during device use. The lattice Boltzmann method was a recently developed alternative to the more mature FEM and FDM techniques for solving multi-physics and computational fluid dynamics problems. The LBM required less computational resources and actively tracked the development of phase interfaces which occurred at the heat source, heat sink, and the bulk vapor-liquid interface. The simulated results,
compared with available working fluid data, confirmed that the MHE operated as intended. After completing the simulation analysis, additional simulation verification was completed through the use of 10 dimensionless numbers appropriate for the MHE. The simulation indicated that condensation at the MHE lid occurred developing a film thickness of 5\(\mu m\) which due to model setup corresponded to a 10\(\mu m\) film thickness in the experimental MHE operation. Bulk vapor-liquid interface movement occurred from 12\(\mu m\) from the MHE lid to 20\(\mu m\), thereby indicating the interface moved from 24\(\mu m\) to 40\(\mu m\) in the physical MHE testing. Therefore, the air/working fluid interface of the channel started at approximately 20\% of MHE depth and after saturation and phase change, the vapor/liquid interface moved down to 33\%. Interestingly, pressure conditions and the natural dominance of surface tension in the channel indicated potential superheating of working fluid with minimal bubble growth and release.

Moving beyond empirical results, there were three important additional results from the investigation. Keeping the copper-based MHE in prototype form eliminated the ability to use site glasses for experimental verification of phase change. However, using all available physical data for the air/working fluid mixture enabled using the computational model and subsequent analysis for justification of experimental results. This conclusion resulted in a faster prototyping process which future researchers can use for expedited projects. The use of physical parameters also introduced the possibilities of using dimensionless numbers in future simulated systems. Potential behavior indicators in the system, such as surface tension dominance, increased confidence that the employed numerical model operated similarly to initial design requirements.
As consequence of the study's outcomes, global adoption of the fabrication process was determined as financially practical for any facility. This was important to potentially increase study and use of MHEs in industry, academia, and possibly commercially. The robust, cost-effective, performance efficient MHE provided the immediate end user with a highly functional energy harvesting device which was intended for operation in multiple environments. In addition to the MHE, the LBM approach investigation allowed for immediate fluid alternatives to be modeled. The ease of transition between operational working fluids provided possibilities for research teams to quickly explore design alternatives.

Progressing past the current study, several recommendations for future study are apparent. Concerning fabrication, there are two avenues to explore further with the dry-film photoresist. The layering technique can be studied to produce taller micro-structures at potentially lower film resolutions. The second option examines other substrates, not necessarily metals, for fabrication of more varied micro-devices. Regarding the operational MHE, designs with narrower and taller channel walls can be explored for optimizing heat transfer efficiency. Additionally, the use of different working fluids for operational necessity, such as high heat flux conditions, needs to be investigated for increasing potential MHE operating conditions and environments. The progression of the numerical model should proceed to three dimensions to better understand transport phenomena occurring during operation, including influence of wicking through copper lattice pores. Increasing simulation dimensionality would also allow a more accurate analysis of phase change, especially for tracking interface motion, and behavior of the system via dimensionless number analysis. After exploring the
simulation with dimensionless numbers, this additional analysis technique appeared to be a reasonable step for modeling physical systems. While the use of these numbers in such a capacity has not appeared in publication to the extent found in the study, examination of these numbers in comparison with the EOS model was a highly successful process which other researchers could use in their studies.

Overall, the objectives set forth by the project to develop an effective fabrication method for producing an efficiently operational MHE was a success. Further analysis of internal operation using the lattice Boltzmann method and behavior within the system by dimensionless number analysis converged on the initial MHE design intentions. Though built and inspired for an end goal prototype at specific operating conditions, the three phases investigated herein established a procedure for development and analysis for future MHE use in various environments.
APPENDIX A

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COPPER PLATED MICROCHANNEL HEAT EXCHANGER FOR MEMS APPLICATION

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ABSTRACT
Demand for increased density circuit architecture, micro- and nano-scale devices, and the overall down-scaling of system components has driven research into understanding transport phenomena at reduced scales. One method to enhance transport processes is the utilization of mini-, micro-, or nano-channels which drive uniform temperature and velocity profiles throughout the system. This work specifically examines a unique heat exchanger. The exchanger is developed as a closed system, with 300µm width channels, fabricated entirely with copper.

The heat exchanger has been designed for widespread use in varied environments. Further, the exchanger is working fluid non-specific, allowing for different fluids to be specified for various temperature ranges. The system design can be used equally well as a standalone heat exchanger or coupled with another device to provide a thermal energy storage system.

Fabricating the heat exchanger with copper for the substrate as well as the channels themselves allows the exchanger to maintain a high thermal conductivity which aids in the fluid energy transfer. The exchanger was fabricated to be a closed system removing any excess equipment such as pumps. In testing, the exchanger showed thermal absorption of 2.2 kW given input of 2.63 kW and working fluid amounts of 37µl. The general design and use of copper in the exchanger allowed maximum absorption of 84% of the input with operation below the boiling point of the working fluid.

INTRODUCTION
Demand for increased density circuit architecture, micro- and nano-scale devices, and the overall down-scaling of system components has driven research into understanding transport phenomena at reduced scales. Effective removal of heat from micro-electro mechanical systems (MEMS) allows more efficiency and reliability of devices while providing an additional platform for useful power output [1]. Micro-channel heat exchangers (MHEs) use multiple fluid-filled channels which, at reduced scales, produce effective heat and mass transfer due to the dynamic boundary layers [2]. MHEs reduce system size while improving efficiency which reduces overall system operating costs [2]. Proper harvesting of MEMS waste heat necessitates an MHE that optimizes channel fluid dynamics while minimizing thermal resistance [1,3]. Thermal resistance is reduced using two-phase flows and materials with high thermal conductivities [1,3].

Traditionally, micro- and nano-scale devices have relied on silicon as substrate or bulk material of fabrication. However, the low thermal conductivity of silicon, when compared to metals such as copper, and its fragility in wafer form demand viable alternatives for more effective and varied usage in dynamic environments [4]. Metal-based MHEs would produce more robust systems while improving system thermal conductivity [1]. There are four broad fabrication classifications for processing micro-channels: photolithography, solidification, subtractive, and additive [5].

An important photolithography technique for metallic mi-


crostructures is LIGA (German for lithography, galvanof ormung (electroplating), and abformung (molding)) where thick photoresists are applied to substrates as molds and filled with electroplated metals [6]. Although LIGA produces high quality microstructures of high aspect ratios, it can be a high-cost process [1, 7]. An adaptation of LIGA involves using polyimide, widely used as COTS materials, as the applied molds for electroplating which is a low-cost alternative to LIGA [7]. Solidification techniques for metallic micro-channels using compression molding have been demonstrated with copper [4, 8]. In addition to surface reactions due to high temperatures and yield strength of mold insert concerns, compression molding requires expensive equipment to perform [1, 4, 8]. Subtractive processes involve milling micro-channels from a substrate [1, 5]. Milling has been demonstrated both with traditional metal bits and electro-erosion or micro-electrical discharge milling (µEDM) [1, 5, 9–11]. Fabrication challenges of milling include the loss of precision following mechanical wear. The high-cost of equipment if using the µEDM method or electro-erosion also presents milling challenges [1]. Additive techniques such as contact printing or vapor deposition are viable only for applications of minimal thickness and utilize high-cost equipment [6]. Sub-micron trenches have been electroplated at high aspect-ratios with copper which validates the proposed level of precision involved with MHEs [12]. An electroplating method provides an efficient, high-precision, low-cost method when coupled with a photolithography mask process.

In this work, a low-cost method for the fabrication of a copper MHE with minimal equipment using an adapted LIGA technique with electroplating is presented. The MHE has components as noted in Fig. 1. It is designed to operate as a closed system needing no external work mechanism. The closed system uses the temperature gradient between the MHE bottom and lid to facilitate the liquid-vapor cycling of the working fluid. The thermal transport is driven by using phase change of a working fluid. Micro-channels drive capillary action of the fluid as depicted in Fig. 2. The boiling working fluid travels to the MHE copper lid where condensation returns the fluid to either the micro-channels or the reservoir. The MHE is designed for multiple applications. There is no specified working fluid in the design. Fluid is selected based on boiling point requirements and device conditions. Copper was selected for the MHE due to its thermal properties, but any metal which can be electrodeposited could be used. Following fabrication of the MHE, the devices are evaluated in thermal testing for thermal absorption of simulated waste heat sources.

**EXPERIMENTAL METHODS**

**Fabrication**

The fabrication of MHEs was based on modified LIGA techniques focused on electroplating copper onto copper substrates. In general, to create the device from Fig. 1, a layer of copper was electrodeposited to provide the reservoirs and micro-channels structure. After the MHE bottom plate was developed, a sheet of copper was polished to produce the MHE lid. Though a closed system design, an access port was provided on the lid in order to add working fluid for testing.

First, copper 110 sheeting was purchased for substrate rigidity. A dry film patterning technique was employed to define channels for electroplating. The dry film manufacturer suggested that lamination of their dry film onto a substrate should be done with a hot roll at elevated temperature [13]. However, previous experiments have shown the possibility of dry film application with a hot flat iron [1]. Once the dry film was laminated onto the substrate, the photolithography mask was secured onto the dry film. After a development time of 20 minutes, the undeveloped film was removed from the substrate, the photolithography mask was secured onto the dry film. Though the recommended dry film exposure with a UV lamp of peak emission 360 to 380nm, experiments with the dry film proved that a fluorescent lamp developed the film successfully for 300µm width channels. After a development time of 20 minutes, the undeveloped film was removed from the substrate with the recommended K2CO3 solution [13]. Figure 3 depicts a sample with a developed dry film and substrate which was prepared for electroplating.

There were several suggested plating baths for copper in the
literature. However, accounting for chemical hazards and ease of preparation, the initial bath solution and plating conditions consisted of copper sulfate in a sulfuric acid mixture given in Table 1 [14]. Therefore, given a 300mL basis, and a plating area of 202.5mm², the information in Table 1 resulted in a bath of 288mL H₂O, 12.07mL H₂SO₄, 56.4g CuSO₄·5H₂O and a current density of 3 A/m².

After several iterations of plating currents ranging from 0.6A to 0.3A, an acceptable current for fabricating the channel areas was established at 0.4A. The 0.4A current yielded consistent, repeatable plating for the MHE geometry. This current produced channels as shown in Fig. 6 with a plating time of 1 hour. It was desired to determine a plating deposition rate in order to create channels of required height. Deposition rate was determined by varying plating time and measuring channel height with a Dektak 150 Surface Profiler. The first several samples were plated with a channel height of 60μm and 300μm channel width. To improve channel uniformity and repeatability, as well to prevent delamination, a further step was introduced into the fabrication process. Prior work indicated electropolishing the copper substrate would provide an improved and clean surface for deposition. Phosphoric acid was selected for the polishing step [15]. Table 2 shows final solutions and conditions for electropolishing copper with a 300g basis after determining proper current density for the surface area.

Once the MHE bottom was electroplated, a copper 110 lid was cut to bottom plate dimensions and polished with the phosphoric acid solution to enhance thermal contact with the bottom plate. Prior to bonding, an access port with diameter ¼ inch was drilled into the center of the lid. Centering the access port provided equal dispersion of working fluid, via the micro-channels, to the two reservoirs. The bottom plate and lid were then contacted with a press and bonded with epoxy on the vertical sides. Care was taken not to get epoxy on the external heat transfer (horizontal) surfaces.

Prior to testing with working fluid, the MHEs were baseline in a dry (no fluid) condition. Devices were tested with no fluid first and then increasing amounts of fluid were added to the MHE. The baseline measurement was performed to determine the energy lost to surroundings. The amount of fluid necessary to fill the MHE reservoirs and channels was 150μL. The vapor-liquid interface was desired so the MHE was not fully filled with working fluid. 3M Tm Novec HFE-7200 engineered fluid was selected as the working fluid for testing which has a boiling point of 76°C. Loading of the MHE fluid was performed by manually filling the MHE using a Gilson Pipetman pipette. Following filling of the MHE, the devices were ready for testing.
Test Setup

Tests were conducted to characterize energy transfer of the MHE. The fabricated MHEs were tested in a setup as depicted in Fig. 4. To minimize energy losses to the surroundings, the testing was performed on a piece of balsa wood. Omega k-type bare wire thermocouples (TC) were used to measure temperature at discrete points on the heater, heat sink, heat flux sensors and the MHE. TCs were centrally located between each layer as shown in Fig. 4. Omega lists the error of the TC as ±0.5°C. Heat flux to the test device was accomplished using an Omega KHLV-101/10-P resistance heater. The heater was connected to a power supply with a measured 1.1W maintained for each test. This power was selected to provide low energy input. A low energy power was selected to establish low temperature potential application for the MHE. To quantify the thermal energy transfer through the setup, an Omega HFS-4 thin film heat flux sensor (HFS) was placed between each layer as noted in Fig. 4. The nominal sensitivity of each HFS-4 has an error of ±10% based on Omega.

The layered test setup, shown in Fig. 4, was fabricated as a vertical stack of heater, heat flux sensor, MHE, heat flux sensor, and heat sink. The heater and heat flux sensors have the same area which allowed ease of stacking. The MHE, being elongated, was centered between the two flat heat flux sensors. Each layer had a TC above and below which was placed in the stack without additional tape so that the temperature results were not affected by extraneous materials. A laboratory jack was then used to press the test setup against the heat sink, which ensured good thermal contact between every layer.

This setup was maintained for all tests independent of the quantity of MHE working fluid. Care was taken to maximize thermal contact between setup layers by using Omega Omegatherm 201 thermal paste. Data was recorded and analyzed via a Lab-View program on a connected PC. The two HFS provided heat flux data which was used to calculate energy throughput and losses. Each HFS was calibrated and verified by Omega. The calibration number was used to convert the voltage signal received by Lab-View into an energy flux. The two HFS used for each test had calibration numbers of 1.9 $\frac{W}{m^2}$ and 1.8 $\frac{W}{m^2}$. The heat flux throughput allowed characterization of the MHE as described in the Results section.

RESULTS

Copper MHEs were fabricated as described in the Experimental Methods section. The suggested current density of $3 \text{A/cm}^2$, given surface area of the MHE, provided a suggested current of 0.61A from the literature [14]. This current density plated copper uncontrollably. Results of plating with the current lowered to 0.5A are presented in Fig. 5. There are many locations which indicated plating selectivity which caused disproportional channels. After the electropolishing step was introduced and the current lowered further to 0.4A, the MHE channels exhibited even plating at a deposition rate of 1 $\frac{cm}{min}$. The results of fabrication are shown in Fig. 6 which proved that well-defined MHEs of copper can be produced with minimal equipment.

The Experimental Methods section described testing setup performed after MHE fabrication. The resulting test data in Fig. 7 was generated with MHEs of 90$\mu$m channel height and 300$\mu$m channel width. Dry MHE readings indicated 2.63 $\frac{W}{m^2}$ of thermal flux entering the MHE and 2.55 $\frac{W}{m^2}$ leaving the MHE. Approximately 97% of energy supplied to the MHE transferred through the system. This information provided assurance that there was minimal energy lost to surroundings and good thermal contact of the MHE components and testing setup.
A partially filled MHE was tested with a volume of 37μL working fluid. This data, depicted in Fig. 7, indicated 2.2 kW of thermal flux throughput with working fluid. The reduction in performance was likely due to the lower thermal inputs of these initial tests. However, if the energy input is increased, the MHE performance is expected to be significantly improved with increased mass transfer and phase change.

Temperature profiles for the dry and wetted tests are shown in Fig. 8 and 9. The low energy input caused a heater temperature between 40°C and 50°C, too low to have significantly caused phase change of the working fluid based on the boiling point of 76°C. The temperature difference between each thermocouple was approximately 5°C in dry and wetted readings. The MHE Lid measurement showed a reduction of 2°C between dry and wetted tests without temperature change of the MHE bottom. Low energy input was achieved; however, selection of an alternative working fluid with reduced boiling point would improve the operation of these devices. Future tests are presently focused on increased heat input and device performance.

CONCLUSIONS

MEMS-based copper plated MHEs have been fabricated and tested for operation. Due to the large thermal conductivity of copper, fabrication focused on processes in which copper could be used for the entire MHE. Without the electropolish step, deposition was selective and channels delaminated from the substrate upon mask removal. Mask thickness set limits for channel height due to anisotropic deposition when mask height was exceeded. Fabrication of the MHE was performed at low cost with minimal equipment. Future work will focus on fabrication of varying channel heights and widths to further refine the fabrication process depending on MHE application.

The performance of the dry MHE indicated good testing setup with minimal energy loss to surroundings. However, energy input was not large enough to fully showcase the capability of the fluid-filled MHE. In both testing scenarios, however, the vast majority of input thermal energy was transferred from the simulated source. This showed the effectiveness of the use of copper substrate and fabrication techniques in general.

Future work is presently investigating the limits and performance characteristics of the devices at increased thermal inputs. Increasing energy input such that temperatures approach and exceed HFE-7200 boiling point will promote phase change within
the MHE. Another approach would consider testing under the same conditions using a different working fluid. Since the MHE is working fluid non-specific, another fluid could be selected with a boiling point appropriate to temperatures achieved with low power input.

The energy input necessary to promote phase change conditions and the use of thermally conductive materials will reduce thermal resistance of the MHE, allowing larger energy throughput. Knowing operating conditions for various working fluids will allow MHE flexibility with multiple applications dependent on device environment.

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REFERENCES
A COPPER MICROCHANNEL HEAT EXCHANGER FOR MEMS-BASED WASTE HEAT THERMAL SCAVENGING

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ABSTRACT

The growing necessity for increased efficiency and sustainability in energy systems such as MEMS devices has driven research in waste heat scavenging. This approach uses thermal energy, which is typically rejected to the surrounding environment, transferred to a secondary device to produce useful power output. This paper investigates a MEMS-based micro-channel heat exchanger (MHE) designed to operate as part of a micro-scale thermal energy scavenging system. Fabrication and operation of the MHE is presented. MHE operation relies on capillary action which drives working fluid from surrounding reservoirs via micro-channels above a heated surface. Energy absorption by the MHE is increased through the use of a working fluid which undergoes phase change as a result of thermal input.

In a real-world implementation, the efficiency at which the MHE operates contributes to the thermal efficiency of connected small-scale devices, such as those powered by thermoelectrics which require continual heat transfer. This full system can then more efficiently power MEMS-based sensors or other devices in diverse applications. In this work, the MHE and micro-channels are fabricated entirely of copper with 300µm width channels. Copper electro-deposition onto a copper substrate provides enhanced thermal conductivity when compared to other materials such as silicon or aluminum. The deposition process also increases the surface area of the channels due to porosity. Fabrication with copper produces a robust device, which is not limited to environments where fragility is a concern.

The MHE operation has been designed for widespread use in varied environments. The exchanger working fluid is also non-specific, allowing for fluid flexibility for a range of temperatures, depending on the thermal source potential. In these tests, the exchanger shows approximately 8.7 W/m² of thermal absorption and 7.6 W/m² of thermal transfer for a dry MHE while the wetted MHE had an energy throughput of 8.1 W/m². The temperature gradient maintained across the MHE bottom plate and lid is approximately 30 °C for both the dry and wetted MHE tests though overall temperatures were lower for the wetted MHE.

INTRODUCTION

The growing necessity for increased efficiency and sustainability in energy systems has driven research in waste heat scavenging. Over the past 30 years, the increasing miniaturization of devices and processing abilities has made previously unthinkable development and scaling possible. With this scaling down of devices, new avenues of waste harvesting possibilities, such as thermal scavenging, have been realized by researchers. Thermal scavenging can provide power for external sensors or provide a platform for structural health monitoring of systems. The key to taking advantage of these thermal harvesting possibilities is the same reason that sustainable alternatives are being sought out: inefficiency of existing systems.

Many standard thermodynamic cycles have low efficiencies where much of the thermal energy generated is rejected to the surrounding environment [1]. Harnessing the rejected thermal
energy via transduction mechanisms allows existing physical effects to power additional systems, such as MEMS devices [2]. Micro-channel heat exchangers (MHEs) use multiple fluid-filled channels which, at reduced scales, produce effective heat and mass transfer due to the dynamic boundary layers [3]. Maximizing harvesting of waste heat for MEMS devices with MHEs necessitates optimizing channel fluid dynamics while minimizing thermal resistance [3,4]. Thermal resistance is reduced using two-phase flows and materials with high thermal conductivities [3,4].

Traditionally, micro-scale devices have relied on silicon as substrate or bulk material of fabrication. However, the low thermal conductivity of silicon, when compared to metals such as copper, and its fragility in wafer form demand viable alternatives for more effective and varied usage in dynamic environments [5]. Adoption of technologies on the global scale requires low-cost materials and processes which can be implemented with minimal equipment infrastructure. Mass-produced metal-based MHEs would produce more robust systems, improved thermal conductivity, and allow usage throughout much of the globe [4]. Though there is numerous literature on heat spreaders and other capillary devices, including heat pipes, the exchanger in this work has been designed to use microchannels between two reservoirs composed entirely of copper [3,6-8].

There are five broad fabrication classifications for processing micro-channels: photolithography, expansion, solidification, subtractive, and additive [9,10]. While all five fabrication methods potentially include required expensive equipment or cleanroom laboratory conditions, there are low-cost options which would allow for widespread use of metal-based MHEs for thermal waste scavenging. One important photolithography technique for metallic microstructures is LIGA (German for lithography, galvanofomung (electroplating), and abformung (molding)) where thick photoresists are applied to substrates as molds and filled with electroplated metals [11]. An adaptation of LIGA involves using polyimide as the applied mold which is a low-cost alternative to traditional LIGA [12]. This adapted electroplating method produces an efficient, high-precision, low-cost method when combined with a COTS mask process.

In this work, a method for the fabrication of a copper MHE using an adapted, LIGA inspired technique is presented. The MHE has components as noted in Figure 1. It is designed for closed system operation whereby no external work mechanism is needed for fluid movement. Thermal and momentum transport are driven by using phase change of a working fluid. The micro-channels drive capillary action of the fluid such that the working fluid is dispersed over the heat source area, improving heat transfer to the working fluid. This mechanism is shown in Figure 2. The boiling working fluid travels to the MHE copper lid where heat transfer to the heat sink allows condensation to occur and the working fluid returns to either the micro-channels or the reservoirs. The MHE is designed for multiple applications in multiple environments. There is no specified working fluid in the operational design. Fluid is strictly selected dependent on boiling point requirements and conditions of device operation. Copper was selected for the MHE due to thermal properties; however, any metal which can be electrodeposited can be used.

The purpose of fabricating this particular MHE is to create a robust piece of equipment with minimal laboratory equipment while maintaining a high heat transfer capacity. Once made, such a device can be coupled with other devices, such as thermal storage systems. Heat exchanger development, along with materials research, are the cornerstones in developing effective and renewable thermal energy storage [13]. One challenge, when studying
the phase change materials (PCM) which are vital to the thermal energy storage, is that most PCM have low thermal conductivities leading to low charging and discharging rates [14]. The MHE studied in this work attempts to maximize structural heat transfer, while providing a platform capable for use in varied environments, through the use of copper.

The intentional ambiguity of design parameters, in fabrication and operation, makes the MHE multi-faceted, both in environmental condition and cost. Economic and situational demands may dictate the use of an alternative metal and working fluid. Following fabrication of the MHE, described in the following section, systems are evaluated in thermal testing for thermal absorption of simulated heat sources.

**EXPERIMENTAL METHODS**

**Fabrication**

The fabrication of MHEs from the modified LIGA technique concentrated on electropolishing copper onto copper substrates. To create the device shown in Figure 1, a 100-200μm layer of copper was electrodeposited onto a copper substrate to produce the reservoirs and micro-channels. Overall dimensions of the exchangers were 22 mm in width and 44 mm in length. After the MHE bottom plate was fabricated, another copper plate was polished for smoothness and used as the MHE lid. Since the system was designed for closed system operation, an access port was installed on the lid to allow addition of working fluid for testing.

A rigid substrate was necessary to limit plating delamination and plating cracking due to device handling. Copper 110 sheeting of 3.175 mm thickness was purchased (McMaster-Carr) for the substrate and lid construction. In order to create a process which minimizes equipment and fabrication time, dry film (Ordyl) was purchased for the micro-channel mask patterning. Electroplating and polishing was performed with solutions of copper sulfate/sulfuric acid and phosphoric acid (Alfa Aesar).

The copper substrate was initially cleaned with an acetic acid solution and rinsed. A flat iron was then heated on a hot plate to approximately 80 °C in order to laminate the dry film onto the substrate. Once the dry film was laminated onto the substrate, the 300μm width channel mask was secured to the laminate. Traditionally, a mask aligner is used to expose the film; however, in order to facilitate use outside of laboratory environments, a fluorescent light was used for exposure. After a development time of 15 seconds, the undeveloped film was removed from the substrate using Ordyls recommended potassium carbonate solution. Figure 3 depicts a sample with a developed dry film and substrate prepared for electropolishing.

Prior work indicated that electropolishing the copper substrate would provide an improved and clean surface for electropolating. Polishing the substrate creates a surface with enhanced uniformity, prevents delamination of plating, and allows for repeatability of MHEs. A phosphoric acid based solution was selected for the polishing step. Table 1 shows the final solution composition and conditions. The established mixture of copper sulfate and sulfuric acid was chosen for electropolating due to ease of preparation and accounting for chemical hazards. Table 2 shows the final solution composition and conditions for plating. The optimum current for the MHE dimensions was determined to be 0.4A. Deposition rate of copper was also a required parameter. Several samples were plated for varying times and a Dektak 150 Surface Profiler was used to measure the deposited channel heights. Using the fabrication method presented, the deposition rate of copper plating was determined to be approximately 1 μm/h. Samples were plated from 60μm in height to 200μm in height, all having channel widths of 300μm.

Once the MHE bottom was electropolated with a height of 120μm, a second copper substrate was polished and plated with a height of 120μm, which served as the MHE lid. The additional porosity and surface area of both the bottom plate and the lid were felt to enhance operation of the MHE. Prior to plating the lid, an access port with diameter 3 mm was drilled into the center of the lid. Centering the access port provided equal dispersion of the working fluid, via the micro-channels, to the two reservoirs. The bottom plate and lid were then contacted with a press and bonded with steel-based epoxy on the vertical sides. The relative smoothness of the plating enhanced thermal contact between the plates; the external seal covered the entire vertical sides of both plates to ensure working fluid did not escape during operation due to the porosity of the electropolating process. Examples of plated MHEs are presented in Figures 4 and 5.

Prior to testing with working fluid, the MHEs were baselined in a dry condition. Devices were tested with no fluid first and then working fluid was added to the MHE. The baseline measurement was performed to determine the energy lost to surroundings.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Working Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>H₂PO₄</td>
<td>157 ml.</td>
</tr>
<tr>
<td>H₂O</td>
<td>75 ml.</td>
</tr>
<tr>
<td>Current</td>
<td>0.045A</td>
</tr>
<tr>
<td>Temperature</td>
<td>55 °C</td>
</tr>
</tbody>
</table>

Table 1. ELECTROPOLISHING CONDITIONS FOR COPPER [15]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Working Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>CuSO₄.5H₂O</td>
<td>188 g</td>
</tr>
<tr>
<td>H₂SO₄</td>
<td>74 g</td>
</tr>
<tr>
<td>Current</td>
<td>0.4A</td>
</tr>
<tr>
<td>Temperature</td>
<td>40 °C</td>
</tr>
</tbody>
</table>

Table 2. ELECTROPLATING CONDITIONS FOR COPPER [16]
and absorbed by air inside the MHE. The amount of fluid necessary to fill the MHE reservoirs and channels of the bottom plate was 150µL. This value was used to determine an appropriate volume of working fluid for testing. The vapor-liquid interface was desired so the MHE was not filled entirely with working fluid. The selected volume of working fluid added to the MHE was 100µL. 3M™ Novec™ HFE 7200 engineered fluid was selected as the working fluid for testing with a boiling point of 76 °C. Loading of the MHE fluid was performed by manually fill the MHE using a Gilson® Pipetman pipette. Following filling of the MHE, the devices were ready for testing.

**Test Setup**

Tests were conducted to characterize energy absorption and transfer of the MHE and temperature profiles of MHE bottom and lid. The fabricated MHEs were tested in a setup depicted in Figure 6. To minimize energy losses to the surroundings, the testing was performed on a piece of balsa wood, which has a thermal conductivity twice that of air. Omega® k-type bare wire thermocouples (TC) were used to measure temperature at discrete points on the heater, heat sink, heat flux sensors, and the MHE. Omega® listed the error of the TC as ± 0.5 °C. The heat source
of the test device was provided by an Omega® KHLV-101/10-P resistance heater. The heater was connected to a power supply with a measured 3.6W maintained for each test. This power was selected to ensure the bottom of the MHE reached temperatures above the working fluid boiling point to cause the required phase change.

To quantify the thermal energy transfer through the setup, an Omega® HFS-4 thin film heat flux sensor (HFS) was placed between each layer as noted in Figure 6. The additional HFS underneath the heater was used to minimize energy loss through the balsa base by tracking heat flux sensor #1 from Figure 6 and maintaining a heat flux measurement of zero. The nominal sensitivity of each HFS-4 has an error of ± 10% according to Omega®.

Worth noting was a method used to minimize energy loss through the test setup underneath the heater. Heater #1 and heat flux sensor #1, from Figure 6, were included in order to offset heat output from heater #2 towards the setup base. The heat flux sensor between the heaters was measured during testing so that there was no heat flux between the heaters. This ensured that the majority of output from heater #2 was directed toward the MHE.

This setup was maintained for all tests independent of the volumetric quantity of MHE working fluid. Care was taken to maximize thermal contact between setup layers by using Omega® Omegatherm® 201 thermal paste. Data was recorded and analyzed via a Lab-View program on a connected PC. The two heat flux sensors (HFS #2 and HFS #3) provided heat flux data which was used to calculate energy throughput and losses. Each HFS was calibrated and verified by Omega®. The heat flux throughput and temperature data allowed characterization of the MHE as described in the Results section.

RESULTS

The results of prototype testing are included in this section. Tests were conducted to characterize the operation of the dry copper-based MHE versus a working fluid wetted MHE. The thermal testing reported how added thermal energy to the device altered heat flux and temperature measurements in order to characterize heat transfer and energy throughput.

Though the given energy input to the heater was 3.6W, resistive losses reduced energy input supplied to the MHE to 3.44W. This corresponded to a heat flux entering the MHE of 8.7 W/m². For the dry MHE, the output heat flux measured at 7.6 W/m² while the wetted MHE output was 8.3 W/m², depicted in Figure 7. The percentage of input flux which exited the MHE was 87.5% for
the dry MHE and 95.4% for the wetted MHE. Through the use of the working fluid, energy throughput through the device was increased by approximately 8%. Energy rejected from the sides of the MHE was assumed negligible versus the overall thermal energy passing from source to sink. This was based on the operating temperatures, heat flux, and side cooling that was restricted to radiation or free convection. Calculation of the side losses for both the dry and wetted MHE yielded less than 1% of the total input power to the device.

The temperature profiles of the dry and wetted MHE also yielded interesting results. The working fluid acted as a heat sink, which lowered the measured thermocouple temperatures an average of 9°C except for the actual heat sink. Both the dry and wetted MHE testing yielded a heat sink temperature of 37°C. The temperature data for the dry and wetted MHE tests are shown in Figures 8 and 9. While the working fluid acted as a heat sink for the entire system, it benefited the temperature difference of the MHE bottom and top an average of 3°C.

The underlining key to these tests was the contribution to efficiency through the use of the working fluid within the MHE. In the dry tests, there was a small pocket of air between the two plates which reduced the thermal conductivity and the energy throughput of the MHE. However, after adding working fluid the pocket was filled approximately two-thirds with fluid. The vapor-liquid interface allowed greater conductivity but also introduced phase change. The capillary action of the micro-channels pulled liquid over the heat source. Vaporization of fluid caused micro-channels to pull additional liquid and the temperature of the MHE lid indicates condensation of working fluid back to the reservoirs to begin the cycle anew.

SEM images of the plated copper are shown in Fig. 10 and 11. The estimated porosity was determined to be 15%. This estimation was performed using a linear bulk volume measurement. Given the volume of the designed MHE, the fabricated sample was massed and converted to volume using the density of copper. It was calculated that the sample's volume was 85% of the theoretical volume of the MHE, thus giving a porosity of 15%. The porous structure increased device surface area and enhanced contact area for the working fluid. Using phase change allowed working fluid to inundate the entire device, not just the capillary micro-channels. Though the system is closed, this structure promoted a highly interactive vapor-liquid interface which increased overall heat transfer.

CONCLUSIONS
Fabrication and operation of a small-scale, MEMS-based MHE has been investigated. The MHE acted in a closed system with operation based on development of capillary micro-channels which provided pump-like working fluid motion across a heated surface, inducing phase change of the fluid. This operation provided a more efficient transfer of energy across the copper device.

Fabrication of the copper device was based on electroplating techniques that produced channel widths of 300 µm with heights of 120 µm. The fabrication methodology proved that devices can be manufactured with minimal equipment and laboratory necessities. This added benefit was meant to potentially increase the global adoption and use of such devices, even in remote areas of the world.

The introduction of working fluid improved the energy throughput of the device by approximately 8%. With a energy flux supply of $8.7 \text{W/m}^2$, a dry device showed output flux of $7.6 \text{W/m}^2$ whereas the wetted device showed output flux of $8.3 \text{W/m}^2$. These
results demonstrated the effectiveness of using working fluid and phase change within the devices. Use of working fluid, coupled with the development of copper techniques allowed a device to be fabricated entirely of copper. This increased thermal conductivity of the device when compared to established materials such as silicon and aluminum.

Future work will investigate the ability to decrease the capillary channel width of the copper exchangers. This should produce devices with greater capillary capability compared to the 300μm width channels reported in this work. Development of alternative channel geometries and the control of plating porosity may be advantageous to improving heat transfer across the device. Further progress on these devices promote adoption of waste heat scavenging programs and potentially opens diverse avenues of applicability for global development.

ACKNOWLEDGEMENTS

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REFERENCES


EXPLORATION OF PHASE CHANGE WITHIN A CAPILLARY FLOW DRIVEN SQUARE MICRO-CHANNEL USING LATTICE BOLTZMANN METHOD AND EXPERIMENTAL BOUNDARY CONDITIONS

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ABSTRACT

Previously published research examined the overall efficiency of heat transfer through a copper plated micro-channel heat exchanger. However, since the device is sealed and composed entirely of copper, understanding the phase change, temperature field, and density field of the working fluid is difficult empirically. Given that the efficiency was shown to be greatly increased by the working fluid phase change, this understanding within the device is important to designing devices of greater efficiency and different working fluids. One method of determining device and component performance is numerical modeling of the system.

Fluids that undergo phase change have long frustrated those attempting to successfully numerically model systems with acceptable stability. Over the past twenty years, the lattice Boltzmann method (LBM) has transformed the simulation of multicomponent and multiphase flows. Particularly with multiphase flows, the LBM "naturally" morphs the phase change interface throughout the model without excessive computational complexity. The relative ease with which LBM has been applied to some multicomponent/multiphase systems inspired the use of LBM to track phase change within the previously recorded experimental boundary conditions for the copper plated heat exchanger.

In this paper, the LBM was used to simulate the evaporation and condensation of HFE-7200 within a capillary flow driven square micro-channel heat exchanger (MHE). All initial and boundary conditions for the simulation are exactly those conditions at which the empirical data was measured. These include temperature and heat flux measurements entering and leaving the MHE. Working fluid parameters and characteristics were given by the manufacturer or measured during experimental work. Once the lattice size, initial conditions, and boundary conditions were input into MATLAB®, the simulations indicated that the working fluid was successfully evaporating and condensing which, coupled with the capillary driven flow, allowed the system to provide excellent heat transfer characteristics without the use of any external work mechanism.

Results indicated successive instances of stratified flow along the channel length. Micro-channel flow occurring due to capillary action instead of external work mechanisms made differences in flow patterns negligible. Coupled with the experimentally measured thermal characteristics, this allowed simulations to develop a regular pattern of phase interface tracking. The

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agreement of multiple simulations with previously recorded experimental data has yielded a system where transport properties are understood and recognized as the primary reasons for such excellent energy transport in the device.

**INTRODUCTION**

Multiphase flow is a fundamental phenomenon necessary for many primary functions in life. While understood in practice and use, multiphase flow is not fully understood theoretically. In order to properly understand what occurs during phase change in many systems, researchers have turned to numerical modeling and simulation. The traditional method to model the physical fluctuations at the phase change interface is to perform macroscopic discretizations of continuum equations, most notably solving for the Navier-Stokes equations. However, the nonlinearity and constant variations of phases at the interface, where different phases are continually combining, separating, and changing, presents difficulties for macroscopic schemes. In the physics discipline of statistical mechanics, it is known that behavior of fluids can be determined through statistical analysis of ensemble systems on the microscopic scale. Though the evolution of complex systems on the microscopic scale involves too many variables, these systems can be expanded artificially to the “mesoscopic” scale where the physics of the system is maintained.

In 1986, Frisch, Hasslacher, and Pomeau presented a Boolean case for using lattice-based gas cellular automata (LGA) to simulate nonlinear fields such as the Navier-Stokes equations [1]. This led to the development of a computational method named the lattice Boltzmann method (LBM). The LBM is based on the continuous Boltzmann equation which is then discretized in time and phase space [2, 3]. When researchers expanded the LBM to macroscopic systems, the Navier-Stokes equations were recovered [1, 3, 4]. Soon after, studies began examining the use of LBM to simulate multiphase and multi-component systems, most notably by Shan and Chen. Their paper introduced the interparticle potential as a means of adding attraction and repulsion to the lattice elastic conditions [4]. Over time, Shan and Chen’s pseudo-potential model inspired several other models including Yuan and Schaefer, Zhang and Chen, and Zeng, who all incorporated particle forces into equations of state [5]. Initially, these LBM models incorporated a density distribution function which is then substituted into a given equation of state to provide the pressure function which gives the phase change interface. However, when temperature differences play a vital role in the system and boundary conditions, an additional temperature distribution function is needed. Fusing these two distribution functions with the chosen equation of state can present difficulties which is why many researchers have tested models using normalized parameters, such as the gas constant, and arbitrary equation of state parameters [5–8]. Unfortunately, for most fluids, this generalization is not valid to determine the phase change interface within a realistic system.

Implementing the lattice Boltzmann scheme requires a thorough understanding of the system in question as well as the properties of the working fluid. In order to simulate the system properly, the physical system must be numerically modeled equivalently and the lattice network must be of sufficient resolution with an appropriate time step [9]. These constraints will assist in representing the correct physics for the system. While in many cases the physical system units are nondimensionalized prior to discretizing the lattice, it is not absolutely necessary [9]. The model presented in this work maintained all units which allowed for comparison with parameter calculations and experimental data.

In this work, an MHE square channel is modeled with prior experimental boundary conditions applied and the lattice Boltzmann method. The channels have been designed for closed system operation whereby no external work mechanism is needed for fluid movement. Thermal and momentum transport are driven by the capillary flow within each channel as well as the phase change of the working fluid. The mechanism of operation is given in a previous work [10]. Applying the appropriate time steps and channel geometry allowed for results which reached steady-state at approximately the same time as the original experimental work.

**EXPERIMENTAL METHODS**

**Model Description**

Prior to examining specific performance of the proposed model, a general description of the model construction is given along with all fundamental equations. Since the system has multiple phases, there is a need to establish a density distribution function and a temperature distribution function in order to fully capture the particle evolution through lattice space and time. The LBM describes a single component or multi-component system by these evolutions of the particle distribution functions. The density distribution function evolves using the traditional BGK collision operator as given in Equation 1. A corresponding equilibrium distribution function relates the relaxation of the system, which is given in Equation 2. The last term in Equation 1 introduces a body force term as presented by Gong and Cheng [6]. This force term includes three forces which change associated velocity terms during collisions, the interparticle interaction force, the interaction force between surfaces and fluids, and the gravitational force [6].

\[
\mathcal{f}_i(x + e_\alpha \delta_\alpha t + \delta_t) - \mathcal{f}_i(x,t) = \frac{1}{\tau} (\mathcal{f}_i(x,t) - \mathcal{f}_i^e(x,t)) + \Delta \mathcal{f}_i(x,t)
\]  

(1)
Analogous to the density distribution functions evolution equation, the temperature distribution function has an evolution equation given by Equation 3. The equilibrium function, given by Equation 4, is very similar to the density equilibrium function. The main difference between the two distribution functions is the last term of each equation. While the density evolution incorporates a body force term, the temperature evolution uses an energy source term. Each distribution has with it an associated velocity, \( e_i \), at position \( x \) and time \( t \). The weighting coefficients, \( \omega_i \), for both equilibrium functions is based on statistical probability for particle movement.

\[
g_i(x + e_i \delta t + \delta t) - g_i(x, t) = -\frac{1}{\tau_T} (g_i(x, t) - g_i^e(x, t)) + \delta_i \omega_i \phi
\]  

\[
g_i^e = \omega_i \tau_T \left[ 1 + \frac{e_i \cdot U}{c_i^2} + \frac{(e_i \cdot U)^2}{c_i^2} - \frac{U^2}{2c_i^2} \right]
\]  

There are many different LBM schemes based on system dimensionality and potential locations for the particles to move. For the proposed model the D2Q9 lattice scheme was chosen due to system symmetry. The D2Q9 scheme is shorthand for two dimensional, nine discrete velocities within the momentum discretization [7]. Figure 1 depicts the velocities for the D2Q9 scheme. There are nine potential velocities and each has an associated probability, given in Equation 5.

\[
\omega_i = \begin{cases} 
\frac{4}{15}, & i = 0 \\
\frac{1}{12}, & i = 1, 2, 3, 4 \\
\frac{1}{10}, & i = 5, 6, 7, 8
\end{cases}
\]  

Before initializing the model with the density and temperature distribution functions, information given by 3M \(^{TM}\) was acquired concerning the working fluid in question, HFE-7200. The large amount of 3M published data allowed for the use of two equations which correlate the density, temperature, and partial pressure of HFE-7200 [11]. This information was used to transform temperature data for boundary conditions into density data. Since there was a direct linear equation associating temperatures with specific densities, the difficulty in fusing two distribution functions was simplified to include only the density distribution function values. The two equations are given in Eq. 6 and 7 below [11].

\[\ln P = 22.289 - 3752.1(1/T)\]  

\[P = 1.4811 - 0.0023026(T - 273.15)\]

Once the density distribution function is established and the temperature array is also found for the model, the pressure field must be determined. Since the pressure gives the phase change interface, it is the overall objective of this paper. In order to determine the pressure, an equation of state is typically used to associate all three variables. The Peng-Robinson equation of state was chosen due to stability and accuracy for most models [5, 6]. Equation 8 gives the Peng-Robinson equation of state. The next section describes how the system was tested and the lattice parameters for model simulation.

\[P = \frac{\rho RT}{1 - b \rho} - \frac{a \rho^2 \tilde{e}(T)}{1 + 2b \tilde{c} - b^2 \tilde{c}^2}\]
Performance Setup

The initial step in pursuing an LBM simulation is the selection of the program which will solve the model. Given the objectives of observing the density, temperature, and pressure fields and the arrays which will develop during the LBM, MATLAB was selected. From the experimental data published previously, the first step for the program is to determine the boundary conditions of the system as well as the initial conditions for the fluid in channel. Symmetry of the channel and the bi-directionality of the fluid flow suggest periodic horizontal boundary conditions. The fixed top and bottom plating provides for the traditional LBM bounce back condition. While these boundary conditions apply to the momentum equations, there are also two additional boundary conditions which account for the heat sink and source. Therefore, temperature boundary conditions must also be set. This presents difficulty in programming where an additional temperature distribution function must be defined. However, one benefit of using a manufactured working fluid is the OEM data sheets and information which potentially provide alternative avenues to establishing a second distribution function. The boundary conditions and initial conditions for the channel model are given in Figure 2.

The physical micro-channel had dimensions of 20mm long, 300μm wide, and 120μm in height. Therefore, the ratio of length to height is approximately 166. While this ratio would simulate conditions along the entire channel, symmetry and bi-directionality of fluid motion due to capillary action allowed the physical ratio of the program to run at half of the channel size. Thus, the ratio used in the program is approximately 80. For scaling, the height of the simulation was halved such that LY was set to 60μm. Therefore, the horizontal lattice was set to 4800μm to give the proper dimensionality of the simulated channel. Temperature from the experimental testing fixed the bottom plate temperature at 85°C and the top plate temperature at 50°C. Time steps were tried to measure the steps necessary to ensure steady-state conditions. Following application of boundary and initial conditions along with channel dimensions for the simulation, working fluid properties were considered to provide the LBM-generated pressure field.

Product information for 3M Novec 7200 fluid includes Eq. 6 and 7 which provide correlation between vapor pressure, density, and temperature [11]. Given these equations, the temperature boundary conditions can be transformed into part of the density array for the lid and bottom boundary conditions. Now that the boundaries are established for the density array, the channel initial conditions must be fixed. Since 80% of the channel is filled with working fluid and 10% is air, the inner array has initial densities of 1.434 for Novec 7200 and 0.0012 for air at ambient temperature. After setting up the static system, fluid motion must be accounted for in the channel. In the two-dimensional system, the horizontal velocity component can be found by measurement. Testing provided a flow velocity along the channel at 20mm/s. At ambient temperature, 3M published viscosity allows calculation of the Reynolds number to be 975.6, giving laminar micro-channel fluid flow. Gravity and buoyancy effects must also be accounted for in the flow profile. However, Cheng and Wu published Eq. 9 [12]. For the system in question, channel diameter is much less than the equation requires, revealing that gravity effects on the symmetry flow should be negligible. Establishing the physical system, boundary and initial conditions, and fluid momentum within the system allows solutions of LBM collisions. This is coupled with the two OEM equations and the equation of state to determine phase transitions within the channel.

\[ d_e = 0.224 \lambda_e \] (9)

In order to develop the pressure field, the parameters for the Peng-Robinson equation of state must be determined. Table 1 was generated from calculation and working fluid information from 3M. Note that all units were maintained due to requiring modeling results in appropriate units. Taking all conditions and the physical system parameters into account, simulations were performed to solve for the phase change interface transitions. Though the program is continually being developed, the following results suggest promising simulations of the phase change transition regions within the micro-channel.
RESULTS

The initial simulations at the dimensions of 4800 µm long and 60 µm height indicated that with laminar flow conditions and negligible gravity effects, the program speed is increased by decreasing the length dimension without sacrificing accuracy. Therefore, the subsequent simulations were conducted with 60 µm height and 240 µm length. The pressure array was plotted with a color map which is default in MATLAB. The system in Fig. 3 is initialized at time \( t = 1 \) and develops through time \( t = 360 \).

The initial image, and subsequent figures, can be described as follows. The yellow line at approximate height 48 µm is the liquid-vapor interface at equilibrium prior to any applied heat. The teal area under the interface is the region of liquid-vapor mixture with greater concentration of liquid. Bulk liquid is the large dark blue region. The lighter blue region at the bottom of the figure is the region experiencing the initial energy input from the heat source. Above the equilibrium interface, there is an orange region which is the liquid-vapor mixture with greater concentration of vapor. Bulk vapor is the dark red region, while the lighter red at the highest region is the region experiencing the initial energy output from the heat sink.

After time \( t = 90 \), there is a shift downward for the vapor-liquid equilibrium interface. The vapor-liquid mixture layers have all expanded and the bulk liquid and bulk vapor layers have contracted given the added energy throughput to the system. At time \( t = 180 \), the process has executed similarly, expanding the vapor-liquid mixture regions while contracting the bulk vapor and liquid regions.

After time \( t = 270 \), the simulation has not approached steady-state. The vapor-liquid equilibrium interface continues to fluctuate and broaden while moving toward the base of the channel as additional bulk fluid is boiling near the top plate. It is also apparent that the condensation on the top plate has slowed due to the bulk vapor moving upward. Both vapor-liquid mixture region above and below the equilibrium interface have broadened and the vapor-liquid mixture at the heat source continues to increase in size.

From the time of \( t = 320 \) onward, the plot begins to indicate steady-state flow with only minute fluctuations in each region. The percent change maintains less than 1% of the pressure array readings. Figure 3(e) shows the channel at \( t = 360 \). The equilibrium interface is in similar position and channel width to the plot at \( t = 270 \). The largest difference is the vapor-liquid mixtures below the equilibrium interface. It has expanded several microns downward. The mixture at the heat source has also roughly doubled in width suggesting that nucleation of working fluid has optimized at steady-state. The vapor region has steadied, though the condensation at the heat sink has slowed such that there is little gradient between the top plate mixture and the bulk vapor flow.

While the model is being continuously improved, these initial results indicate a strong vapor-liquid equilibrium interface which fluctuates over several microns but reaches steady conditions after approximately \( t = 320 \). The vapor-liquid mixtures on either side of the equilibrium interface increase in size over time until they are roughly the size of the equilibrium interface. Those three transition regions cover approximately 12 µm of the simulation which corresponds to 25 µm of the experimental tests. While the bulk vapor and liquid regions contract, they maintain the majority of the channel volume. The top plate condensation appears to slow as the larger working fluid volume begins to vaporize and the bottom plate evaporation tends to increase with a nucleation zone thickness of approximately 10 µm for the simulation and 20 µm for the experimental model.

CONCLUSIONS

The phase change transition regions, within one square micro-channel heat exchanger, were explored through the use of the lattice Boltzmann method. This computational method coupled with working fluid properties allowed a model to be constructed which accounted for body force terms and phase change at the transition regions. Experimental results were used to implement the boundary conditions for temperature while experimental analysis yielded fluid flow conditions which could be simplified for the program, such as bi-directionality and critical diameter. The simplification of the physical system model allowed for the focus to placed on incorporating the density and temperature distribution functions and the final pressure array, given by the Peng-Robinson equation of state.

Including the temperature boundary conditions produced progression of the equilibrium interface downward in the channel and indicated that the equilibrium conditions increased to broaden the interface in the channel. Stratification within the program appeared correct, where the vapor-liquid mixture regions surrounding the interface also began to broaden until steady-state. The vaporizing mixture at the heat source interface expanded prior to reaching steady-state; however, the difference in temperature between the two copper plates was not large enough to maintain an analogous interface on the condensing plate. While there was still pressure drop at the top plate interface, it did not have the same width and large difference as the bottom plate.

Table 1. Peng-Robinson EOS parameters [11]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a )</td>
<td>3.6705 ( \frac{MPa}{K} )</td>
</tr>
<tr>
<td>( b )</td>
<td>0.00016 ( \frac{K}{MPa} )</td>
</tr>
<tr>
<td>( R )</td>
<td>8.314 ( \frac{K}{MPa} )</td>
</tr>
<tr>
<td>( T_r )</td>
<td>483.15 K</td>
</tr>
<tr>
<td>( P_r )</td>
<td>2.01 MPa</td>
</tr>
</tbody>
</table>
Figure 3. Progression of channel simulation phase change interface versus time.
Future work will investigate combining experimental data with the program model. As the model develops more accuracy, a better progression of the phase change regions will become more apparent. There should also be larger data sets for which to correlate the density/temperature array with the pressure array in order to gain greater insight into the transitions regions. Running the program on a computer system with larger memory would also allow for the entire channel to be simulated, giving a larger picture of the channel fluid interaction. Further progress on this model promotes design alterations which will take advantage of knowing how the phase change transition regions change over time. With this knowledge, better experimental testing can occur in order to develop more efficient, higher performing devices.

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