DESIGN AND PERFORMANCE CHARACTERIZATION OF HIGH POWER DENSITY AIR-COOLED COMPACT CONDENSER HEAT EXCHANGERS

BY

NICHOLAS IAN MANISCALCO

DISSESSATION

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Doctoral Committee:
Professor William P. King, Chair, Director of Research
Professor Anthony M. Jacobi
Professor John G. Georgiadis
Professor Gregory S. Elliott
ABSTRACT

The current investigation reports the design, fabrication, and characterization of ultra-compact air-cooled condenser heat exchangers with exceptional power density performance in excess of 100 W/cm³. This exceptional heat exchanger performance is attributed to the implementation of high-speed compressible air flow in circular microchannels and two-phase condensation flow of refrigerant in rectangular minichannels to achieve high heat transfer performance in a compact design. These advancements are enabled by recent developments in micro-electric discharge machining to produce very high surface area compact heat exchanger devices.

Air-cooled heat sinks have been identified as pivotal to the future of thermal management of microelectronics in the 21st century. Air-cooling systems provide clear advantages in overall ease of integration due to its availability and abundance over forced liquid cooling systems. The enhancement of forced air flow is especially important because it generally represents the dominant thermal resistance of the condenser. The implementation of high-speed compressible air flow in microchannels permits investigation of flow conditions with exceptionally high heat transfer coefficients. This investigation explores turbulent flow regime conditions for air-side Reynolds numbers 8,000 < Re < 20,000, in high-density parallel arrangements of circular copper microchannels, 355 and 520 μm in diameter.

Two-phase heat transfer technology is one of the most efficient methods of waste heat removal in high power electronics cooling. It is especially advantageous in applications where size, weight, and efficiency are important factors. Two-phase active cooling systems consist of an evaporator and a condenser. Heat transfer performance in the evaporator is typically much higher than in the condenser; consequently the condenser is the limiting component of the entire cooling system. Improvements in condenser technology enable electronics systems to operate at a higher power while reducing the overall cooling system size and weight. The condensation phase change process in microchannels with high aspect ratios yields the formation of a thin film condensation layer on the heat transfer wall, resulting in high heat transfer coefficients with little pressure drop penalty. It is therefore of utmost importance that condensation phenomena, especially in high aspect-ratio microchannels be experimentally investigated. This investigation employs condensation in a parallel array of high-aspect ratio, 0.5 mm x 2 mm, rectangular minichannels.
Compact cross flow heat exchangers are manufactured using novel micro-electro-discharge machining to produce high-density, high aspect ratio microchannels in a copper alloy. The heat exchanger performance is characterized for single-phase liquid and phase-change condensation of a refrigerant. The heat exchangers are operated using single-phase liquid flow of dielectric refrigerant R245fa at 80 °C and high-speed flow of air at ~25 °C to demonstrate a power density performance of nearly 70 W/cm³. The heat exchangers are operated using two-phase condensation of R245fa at 80 °C and high-speed flow of air at ~25 °C to achieve power density performance > 175 W/cm³ and overall thermal resistance < 0.27 K/W.

Test methodologies were implemented for determination of the thermal-hydraulic performance of these novel devices. Modeling and characterization of this system were implemented using well-known methods and the results are compared with the corresponding literature for microchannel fluid flow and heat transfer. The study of this system demonstrates an advancement in the state-of-the-art in power density performance of compact air-cooled heat sinks. Power dissipation rates > 1 kW and an overall thermal resistance of < 0.05 K/W are projected with scaling of these methodologies in a 10 cm³ device.
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**Roman Symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$</td>
<td>speed of sound</td>
<td>[m/s]</td>
</tr>
<tr>
<td>$A$</td>
<td>area</td>
<td>[m$^2$]</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat capacity at constant pressure</td>
<td>[J/kg-K]</td>
</tr>
<tr>
<td>$c_v$</td>
<td>specific heat capacity at constant volume</td>
<td>[J/kg-K]</td>
</tr>
<tr>
<td>$COP$</td>
<td>coefficient of performance</td>
<td></td>
</tr>
<tr>
<td>$D_h$</td>
<td>hydraulic diameter</td>
<td>[m]</td>
</tr>
<tr>
<td>$e$</td>
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</tr>
<tr>
<td>$f$</td>
<td>fanning friction factor</td>
<td></td>
</tr>
<tr>
<td>$F$</td>
<td>LMTD correction factor</td>
<td></td>
</tr>
<tr>
<td>$G$</td>
<td>mass flux</td>
<td>[kg/m$^2$-s]</td>
</tr>
<tr>
<td>$h$</td>
<td>heat transfer coefficient</td>
<td>[W/m$^2$-K]</td>
</tr>
<tr>
<td>$i$</td>
<td>specific enthalpy</td>
<td>[J/kg-K]</td>
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<tr>
<td>$j$</td>
<td>Colburn factor</td>
<td></td>
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<tr>
<td>$K$</td>
<td>dynamic recovery factor</td>
<td></td>
</tr>
<tr>
<td>$K_e$</td>
<td>entrance pressure loss coefficient</td>
<td></td>
</tr>
<tr>
<td>$K_e$</td>
<td>exit pressure loss coefficient</td>
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</tr>
<tr>
<td>$K(x)$</td>
<td>incremental pressure defect</td>
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<tr>
<td>$k$</td>
<td>thermal conductivity</td>
<td>[W/m-K]</td>
</tr>
<tr>
<td>$Kn$</td>
<td>Knudsen number</td>
<td></td>
</tr>
<tr>
<td>$L$</td>
<td>channel length</td>
<td>[m]</td>
</tr>
<tr>
<td>$Ma$</td>
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<td>$m$</td>
<td>mass flow rate</td>
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</tr>
<tr>
<td>$N_c$</td>
<td>number of channels</td>
<td></td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
<td></td>
</tr>
<tr>
<td>$P$</td>
<td>pressure</td>
<td>[Pa]</td>
</tr>
<tr>
<td>$P$</td>
<td>thermal effectiveness</td>
<td></td>
</tr>
<tr>
<td>$P_W$</td>
<td>wetted perimeter</td>
<td>[m]</td>
</tr>
<tr>
<td>$P_0$</td>
<td>stagnation pressure</td>
<td>[Pa]</td>
</tr>
</tbody>
</table>
\( \Delta P \) differential pressure \[ \text{Pa} \]

Po Poiseuille number

Pr Prandtl number

\( Q \) heat rate \[ \text{W} \]

\( Q' \) heat flux \[ \text{W/m}^2 \]

\( r_h \) hydraulic radius \[ \text{m} \]

\( R \) thermal resistance \[ \text{K/W} \]

\( R \) heat capacity rate ratio

\( R \) specific gas constant \[ \text{J/kg-K} \]

\( r \) recovery factor

Re Reynolds number

\( S \) conduction shape factor \[ \text{m}^{-1} \]

\( T \) temperature \[ \text{K} \]

\( T_0 \) stagnation temperature \[ \text{K} \]

\( UA \) overall thermal conductance \[ \text{W/K} \]

\( u \) velocity \[ \text{m/s} \]

\( V \) volume \[ \text{m}^3 \]

\( \dot{V} \) volumetric flow rate \[ \text{m}^3/\text{s} \]

\( W' \) pumping power \[ \text{W} \]

\( x^+ \) dimensionless axial distance

**Greek Symbols**

\( \alpha \) aspect ratio, void fraction

\( \beta \) fin parameter

\( \gamma \) heat capacity ratio

\( \eta_f \) fin efficiency

\( \theta \) dimensionless temperature

\( \lambda \) mean free path

\( \mu \) dynamic viscosity \[ \text{kg/m-s} \]
\( \nu \)  
kinematic viscosity  
\([\text{m}^2/\text{s}]\)

\( \rho \)  
density  
\([\text{kg/m}^3]\)

\( \sigma \)  
area ratio

\( \chi \)  
fin parameter

**Subscripts**

\( a \)  
absolute

\( aw \)  
adiabatic wall

\( C \)  
cold

\( c \)  
cross-sectional

\( cv \)  
control volume

\( CF \)  
counter flow

\( char \)  
characteristic

\( cond \)  
conduction

\( exp \)  
experimental

\( f \)  
fluid

\( fg \)  
liquid-vapor

\( fr \)  
frontal area

\( g \)  
saturated vapor

\( H \)  
hot

\( in \)  
inlet

\( LM \)  
logarithmic mean

\( LO \)  
liquid only

\( m \)  
mean

\( max \)  
maximum

\( me \)  
measured

\( min \)  
minimum

\( out \)  
outlet

\( p \)  
probe

\( ref \)  
refrigerant

\( s \)  
surface
$sat$ saturation
$VO$ vapor only
$w$ wall
$W$ wetted
CHAPTER 1: INTRODUCTION

1.1 Motivation

Thermal management in electronics has been a key driver of heat transfer research for over 30 years [1-3]. Energy conversion and utilization are continuous but ever increasing processes for sustainability and economic development. Heat exchangers are a universal component for achieving energy conservation through enhanced heat transfer [4]. The issue of thermal management in both military and commercial systems has become increasingly important due to more powerful electronic processing capabilities and the resulting heat generation. Over the past decade, average heat dissipation in chips and printed circuit boards has more than doubled [5, 6]. Modern electronic components are being asked to maintain higher performance at a lower heat tolerance under more demanding climatic conditions. Issues such as increased energy demands, space limitations, and material savings have highlighted the need for miniaturized light-weight heat exchanger to high heat transfer for a given heat duty [1-4]. As heat exchangers employing conventional channels (>3 mm) approach their margins for utility, microchannels (<1 mm) represent the next step in heat exchanger development.

Thermal management of processors is an important field of study devoted to engineering the removal of heat from these devices. Natural air and forced convection of air over the chip are no longer capable of maintaining process temperatures below acceptable values. Heat transfer at the electronics source is the primary driver of power electronics’ performance [7]. As such, a majority of research on compact heat exchangers and phase-change electronics cooling has focused on evaporation and boiling near the electronics being cooled [8-11]. Boiling in microchannels for heat removal of computer chips has been shown to yield the lowest pumping power and highest heat dissipation rates. Combining these technologies with vapor compression refrigeration to enable high heat fluxes has been called for in the literature [12]. Saturation conditions and flow rates can be controlled refrigeration systems to achieve a desirable range of operation.

The majority of research in microchannel heat sinks focuses on heat removal from the chip at the evaporator. As a result of this emphasis, the compactness and performance of the condenser has been subsequently neglected. In a complete thermal system, the condenser and the evaporator must work in concert; it is therefore essential that their capabilities be well matched. Space
restrictions impose a limit on the size and weight of refrigeration systems and are one of the most challenging aspects of implementing these technologies. It is therefore imperative that the compact design of all components in the cycle be considered in order to advance the technology [12].

The current research aims to bring compact condenser technology to parity with evaporator technology. Almost all existing research on compact condensers has been driven by the HVAC and automotive industries rather than electronics thermal management [13]. The development of new applications in thermal management requires rapid cooling in confined spaces. Electronics cooling, aerospace, MEMS for power electronics cooling, and space thermal management have increased the number of applications requiring high heat transfer rates and fluid flows in relatively small passages [1].

High-flux thermal management is a primary design concern for advanced defense devices found in radars, directed-energy laser, and microwave weapon systems and avionics [14]. While these devices follow the trend of escalating power density of commercial electronics, heat fluxes from defense devices are projected to exceed 1000 W/cm² [15]. This level of heat dissipation exceeds the current trend in capabilities being developed by today's most advanced dielectric liquid cooling systems, highlighting the need to develop novel cooling technologies. Defense electronic systems fail to reach maximum device performance due to high thermal resistances [16]. Thermal management hardware in advanced electronic systems accounts for a large fraction of the volume, weight, and cost which restricts the efforts to transfer emerging electronic components to portable applications. A considerable amount of research effort has been concentrated on using microchannel heat sinks to reduce total thermal resistance. Today's most effective cooling solutions are usually based on microchannel coolant flows and jet impingement [17]. Decreasing the hydraulic diameter has been a common tactic employed to significantly increasing the convective heat transfer coefficients in order to achieve high performance power density dissipation. Microchannel heat sinks are ideal for compact light-weight applications. However the two key drawbacks associated with these devices are the high pressure drops and large temperature gradients across the device.

Water, although cheap and abundant is ill suited for two-phase cooling of most microelectronic applications because most silicon based microelectronics have a maximum junction temperature around 85 °C, which lower than the boiling point of water at atmospheric conditions. Therefore, low pressure dielectric refrigerants which are inherently safer when being
used near electrical connections must be studied. Implementation of low pressure dielectric refrigerants offers the advantage of high heat fluxes at practical operating conditions.

This investigation presents an experimental analysis for compact microchannel condensers for a miniature scale refrigeration system. The thermal-hydraulic performance of microchannel condensers with high-speed air flow is enhances power density performance in a compact cross-flow heat exchanger with single-phase and condensation phase-change refrigerant-side heating.

1.2 Objectives of Work

Heat transfer coefficients in microchannels are very high due to their small hydraulic diameters. However previous researchers have been limited by low flow rates due to the high pressure gradients incurred at these reduced channel diameters. For a given temperature rise, the ability of the fluid stream to remove heat is limited by the thermal capacity of low fluid flow rates. Therefore in order to improve the overall cooling performance multiple microchannels with short lengths are recommended to reduce the overall pressure drop while exploiting the benefits of high heat transfer and high flow rates. The incorporation of multiple channels in parallel with short path lengths leads to a larger dominance of developing region effects when the heat transfer rate is high. An overall pressure drop reduction will enable higher velocities to be explored, which provides for the investigation of turbulent flow conditions with high heat transfer coefficients in microchannels.

The main objective of this work is to design, fabricate, and characterize ultra-compact air-cooled condenser heat exchangers with performance that exceeds a power density of 100 W/cm³ and a heat load 1 kW with an overall thermal resistance >0.05 K/W. The project additionally aims to achieve heat flux at the refrigerant-air interface of 1 kW/cm² or higher, at as large a coefficient of performance as possible. The two-fluid exchanger incorporates high speed air flow through metal microchannels to condense a high temperature dielectric refrigerant flowing though microchannel passages in cross-flow. Modeling and characterization of this system was performed using well-known methods and then compared with the corresponding literature on microchannel flow and heat transfer. The study of this system provides broad and useful insights for advancing the start-of-the-art in design and operation of compact air-cooled heat sinks.

The objectives of this work is can be broken into two main areas. The first area is focused on modeling and design of the heat exchangers. This part also describe the fabrication of the devices and the integration of the heat exchangers in the test facility. The second part of this work
is focused on the thermal and hydraulic performance characterization of the devices. The objectives of the work are summarized as follows:

1. Design and Experimental Methods
   a. Model heat exchanger performance
   b. Select heat exchanger designs to meet performance metrics
   c. Fabricate the heat exchangers
   d. Construct test facilities

2. Heat exchanger characterization
   a. Characterize thermal-hydraulic performance of air side
   b. Characterize single-phase thermal-hydraulic performance of refrigerant-side
   c. Characterize single-phase thermal-hydraulic performance of refrigerant-side
   d. Compare performance metrics with reported literature
   e. Summarize conclusions and impact of work

1.3 State-Of-The-Art in Heat Sink Technology

Research in the field of thermal fluidics at the microscale level has been steadily increasing due to rapid growth in technological applications requiring high rates of heat transfer in small volumes. Recent advances in microelectromechanical systems (MEMS) and advanced very large-scale integration (VLSI) technologies have led to significant increases in the packing densities of these devices and allow heat fluxes of up to 200 W/cm² to be generated [18]. The automotive, aerospace, chemical reactor, MEMS, and cryogenic industries are responsible for most of the research and development of microchannel heat exchangers technologies. Thermal duty and energy efficiency requirements have increased during this period and space constraints have become more restrictive. The trend has been to focus on increasing the heat transfer rate per unit volume [19]. Microchannel heat exchangers are well suited to meet the demands of these applications due to their light weight, compactness, and high heat transfer performance.

The goal of thermal management is to remove the thermal energy dissipated within a component and reject it to the ambient surroundings. Various physical phenomena have been utilized to provide cooling in a wide variety heat exchanger configurations. Commonly applied methods in high flux thermal management include natural convection, radiation, forced air convection, forced liquid convection and evaporation. Although forced liquid cooling systems are capable of
accommodating higher heat loads, air-cooling systems provide clear advantages in overall ease of integration due to its availability and abundance. Liquid-cooled systems are useful in large systems for transporting heat from a locality however the issue of rejecting this heat to the ambient is generally not suitable for portable electronic systems due to size, weight, and complexity restrictions. Air-cooled heat sinks can inherently reduce the overall size and cost of a system by eliminating the need for external connections. Air-cooled heat sinks have been identified as playing a pivotal role in the future of thermal management of microelectronics in the 21st century [20].

A forced convection mechanism accounts for heat transfer in almost all compact heat exchangers. In traditional air-to-liquid cross-flow heat exchangers, the air-side generally accounts for about 80% of the total thermal resistance, although in some cases it can account for more than 90% [21]. The enhancement of forced air flow is especially important because it usually represents the dominant thermal resistance. In order to minimize the size and weight of an exchanger the thermal conductance on both sides of the exchanger should be well matched [12]. The heat transfer coefficient for gases is generally one or two orders of magnitude lower than those for water, oil, and other liquids. As a result of this limitation, the heat transfer surface on the air-side must usually have a much larger area highlighting the demand for a more compact design.

Staats [22] reviewed the thermal performance of a number of commercially available air-cooled heat sinks. Results were summarized for 20 heat sinks aggregated from a review website, Frostytech [23], who performed the thermal testing. The thermal performance of the Sandia Cooler [24] and the expected performance of Staats work which was nicknamed “PHUMP” were also included. The different heat exchanger designs can be normalized by Eq. (1.1) which defines a normalized volume, $\tilde{V}$, as a dimensionless measure of how much volume, $V$, the heat sink occupies relative to the heat input surface area, $A_h$, over which the heat load is applied, to give a measure of the heat sinks compactness. A 150 W load was applied to the tested heater blocks with a precision power resistor and the temperature rise was measured using an embedded thermocouple to measure the temperature rise above the ambient and calculate the thermal resistance. The Sandia cooler was reported as having the best performance with a normalized volume of ~0.3, and a thermal resistance of ~0.15 K/W. Recently published experimental results on the PHUMP which employs an impeller motor to cool a single condenser loop heat pipe with a condenser 26 cm³ (10.2 x 10.2
x 0.25 cm) in size, dissipated a heat load of 200 W, (a power density of 7 W/cm³) and an overall thermal resistance of 0.177 K/W [25].

\[ \bar{V} = \frac{V}{A_h^{2/3}} \]  \hspace{1cm} (1.1)

1.3.1 Micro-Manufacturing of Heat Exchangers

The unique characteristics of a compact extended surface make it possible to manufacture components having different orders of magnitude of surface area densities. These advanced techniques provide flexibility in the distribution of surface area between the two sides of a compact heat exchanger as warranted by the design. This enhancement leads the realization of substantial cost, weight, and volume savings [26]. In 1981 Tuckerman and Pease [1] demonstrated increased heat transfer performance with heat-sinking silicon-based microchannels anodically bonded to Pyrex cover plates. Many subsequent heat transfer investigations have been conducted using silicon based microchannels [15, 27, 28]. In heat sinking the use of silicon has been largely based on the use of mature fabrication techniques for producing high-aspect ratio microscale structures. However, silicon is not an optimal material for cooling devices with regard to its thermal and mechanical properties compared to other materials such as copper, silver, aluminum, or diamond.

Metals such as copper and aluminum possess higher bulk thermal conductivities than silicon [29]. Metal based microchannel heat exchangers also promise increased mechanical robustness. Various techniques for fabricating metal based heat exchangers including micromilling [30], wire micro electrical discharge machining [31], micropower injection molding [32], microcasting [33], and laser-beam direct writing [33] have been studied. Cross flow heat exchangers have also been fabricated from precision-cut metal foils that are stacked and bonded together [34]. Lightweight metal foams have been used in the design of compact microchannel heat exchangers [12, 35-37]. Metal-based microchannel heat exchangers are of particular interest to the aerospace industry due to the combination of high heat transfer performance and the improved mechanical integrity over silicon devices. With the aim of reducing size and cost, microchannel heat exchangers have been demonstrated to achieve performances for surface area per unit volume as high as 1500 m²/m³ [19].

Micromachining technology using photolithography on silicon substrates is the most widely used process for fabricating microstructures, but it is limited by choice of working material...
and aspect ratio. Deep X-ray lithography using synchrotron radiation beam processing has been used to produce very high aspect ratio, three-dimensional sub-micron structures with high accuracy. These structures are limited in their maximum thickness and require special facilities for fabrication. X-ray lithography has been demonstrated to create high-aspect ratio microstructures with feature sizes down to 20 μm and a height exceeding 300 μm [38]. Minimum feature sizes of 7 μm and an aspect ratio of 3.2 were demonstrated by batch mode of sinker μEDM using stainless steel and titanium coated on silicon microstructures formed by deep reactive ion etching over a 5 mm x 5 mm area [39].

Micro-electrical discharge machining is a manufacturing process that selectively removes an electrically conductive material by plasma discharge. A metal electrode tool, having a shape that is the inverse of the desired part, is brought near the surface of a conductive part. The tool and part are separated by a dielectric liquid and an electrical bias applied between the tool and part generates a plasma discharge. An electrical current pulse is driven across the gap in the dielectric liquid which consumes some of the tool and part, leading to the creation of features in the part [40]. An increase in pulse energy increases the plasma temperature, alters in the heat distribution pattern between the two electrodes, which in turn increases tool wear rate [41]. Careful control and monitoring of the operating parameters have allowed parts to be made at lower voltages than conventionally possible, thereby minimizing the tool wear and machining time [42]. Proprietary advancements in parameter control are responsible for high-throughput machining of high-density high aspect ratio microstructures into a copper part, thus enabling the manufacturing of the compact condensers utilized in this project.

1.4 Single-Phase Flow in Microchannels

Due to their high surface area to volume ratios, microchannels have been a target of research aimed at increasing compactness and performance of compact heat exchangers. In general, improving heat transfer performance requires either increasing the heat transfer coefficient or increasing the surface area participating in heat transfer. Increasing the surface area per unit volume by shrinking system sizes can lead to remarkable heat transfer enhancements [43]. Taking advantage of this approach requires knowledge of the prevalent governing equations and boundary conditions at these scales.
The compressible Navier-Stokes equations are the governing conservation laws for mass, momentum and energy. These laws assume that the fluid is Newtonian. The external volume forces such as gravitational and magnetic forces are generally negligible in the case of gas microflows because the volume over surface ratio decreases with the characteristic length.

The Reynolds number, Re, as defined in Eq. (1.2), is a dimensionless quantity defined as the ratio of inertial forces to viscous forces. The Re number is frequently used to help predict flow patterns between different fluid flow situations. Low Re numbers indicate that the flow is laminar, and as the value of the Re increases, the flow becomes turbulent. The Mach number, Ma, is a dimensionless quantity representing the ratio of the speed of an object or a fluid to the local speed of sound as defined in Eq. (1.3). The speed of sound, \(a\), is determined from Eq. (1.4). For an ideal gas, the specific heats are constant and their ratio is defined by the index, \(\gamma\), given in Eq. (1.5). The specific gas constant, \(R\), defined in Eq. (1.6), is the difference between the specific heats defined in Eq. (1.7) and Eq. (1.8).

\[
\begin{align*}
\text{Re} &= \frac{\rho u L_c}{\mu} \quad (1.2) \\
\text{Ma} &= \frac{u}{a} \quad (1.3) \\
a &= \sqrt{\gamma RT} \quad (1.4) \\
\gamma &= \frac{c_p}{c_v} \quad (1.5) \\
R &= c_p - c_v \quad (1.6) \\
c_v &= \left. \frac{\partial e}{\partial T} \right|_v = \frac{1}{\gamma - 1} R \quad (1.7) \\
c_p &= \left. \frac{\partial h}{\partial T} \right|_p = \frac{\gamma}{\gamma - 1} R \quad (1.8)
\end{align*}
\]

1.4.1 Scaling Effects in Microchannels

Single-phase heat transfer in microchannels and minichannels can generally be described by standard theory and correlations. However, phenomena and scaling effects that are often negligible in macrochannels may need to be accounted for in microchannels and minichannels. In his chronological review of 90 papers on published experimental results, Morini [44] observed that
many of the proposed heat transfer correlations only predict their own experimental data well. It was noted the deviations between the behaviors of fluids through microchannels with respect to macrochannels were decreasing as time progressed. This trend reasoned to be explained by improvements in fabrication, measurement techniques, and interpretation of data including minor losses and scaling effects. When scaling effects are properly accounted for, the classical fluid dynamics theory and correlations seem to be in reasonable agreement with the experimental data [18, 45-47]. It was recommended that entrance effects, viscous heating, temperature dependent properties, surface roughness, rarefaction, and compressibility be checked for significance when considering gas flow in microchannels.

A large scatter in published results for fluid flow and heat transfer characteristics has been reported in microchannels [5]. According to conventional theory, continuum based models for duct flow should be valid for Knudsen numbers lower than 0.01. Contrary to this prediction, early experimental investigations of microchannel flow were shown to find discrepancies between the standard models and the microchannel flow measurements. Researchers conducting experiments with liquids reported friction factors greater than what was predicted by theory [28, 48-51]. Wu and Little [52] found friction factors to be much greater than those predicted by Shah and London [53] for laminar flows and slightly greater than those predicted by the Blasius equation [54] for turbulent flow. It was also reported that transition to turbulence occurred much earlier than expected [55, 56]. Kurokawa et al. [57] report an increase in critical Reynolds number and friction factor for a fluid in a microchannel when studying the effects of acceleration and deceleration of an incompressible fluid. It was found that increased acceleration of the fluid resulted in an increase of both the critical Reynolds number and friction factor.

Chung et al. [58] found that their test results for water were consistent with the predictions of Shah and London [53] and the test results for nitrogen gas were well correlated with standard theory as long as compressibility effects were taken into account. Analytical and experimental work conducted by Arklic et al. [59] showed that by accounting for the effects of compressibility of gaseous flows, their measured results agree with the Navier-Stokes equations. Kohl [27] conducted an experimental investigation of microchannel flow with internal pressure measurements and concluded that friction factors for microchannels could be accurately determined from data for standard large channels. Kohl attributed the large inconsistencies in previously published data to instrumentation errors and failure to account for compressibility
effects and entrance effects. Vijayalakshmi [60] studied the effects of compressibility and transition to turbulence on flow through long microchannels ranging in diameters from 60 – 211 μm and found that there are no special micro-scale effects, including early transition to turbulence, when the effects of compressibility were appropriately accounted for.

Several criteria have been proposed for classifying microchannels versus minichannels in the literature [61-63]. While it is not always suitable to differentiate minichannels and microchannels by a specific hydraulic diameter such as 1 mm, this is a common, yet inconsistent, definition adopted by researchers [19]. Microchannel heat exchangers have been broadly defined as devices incorporating at least one fluid flow passage with typical dimensions between 1 μm and 1 mm [64].

Rarefaction effects must be considered when the length scale of the channels is on the order of the average distance a molecule travels between two collisions, known as the mean free path, λ. In order for the continuum approach to be valid the frequency of the intermolecular collisions within the sampling volume must be high enough. This implies that the mean free path of the molecules must be small relative to the characteristic length of the sampling volume. The ratio of the mean free path to the characteristic length of the control volume is referred to as the non-dimensional Knudsen number given in Eq. (1.9). The Knudsen number is related to the Reynolds number and the Mach number in Eq. (1.10).

\[
Kn = \frac{\lambda}{L_{char}} \quad (1.9)
\]

\[
Kn = k_2 \sqrt{\gamma} \frac{Ma}{Re} \quad (1.10)
\]

When Kn > 10⁻³ rarefaction effects in the gas must be taken into consideration. Flow is considered to be in the following regimes depending on Kn: continuum flow for Kn < 10⁻³; slip flow for 10⁻³ < Kn < 10⁻¹; transition flow for 10⁻¹ < Kn < 10; and free molecular flow for Kn > 10. Flow in the continuum regime may be accurately described by the compressible Navier-Stokes equations, equation of state of an ideal gas, and classical boundary conditions that express the continuity of temperature and velocity between the fluid and the wall [43]. It is possible that for very small microchannel diameters, usually less than 10 μm, flow may be in the slip-flow regime. Sparrow and Lin [65] investigated slip flow in microchannels and found that the Nusselt number decreases with increasing Knudsen number for both constant temperature and constant heat flux.
boundary conditions. This reduction is strongly influenced by a temperature jump that needs to be considered. Kavenhpour [66] showed that the Nusselt number was substantially reduced for slip flow when compared with continuum flow for a developing compressible flow.

In conventionally sized channels, gas flow is assumed to be incompressible as long as \( \text{Ma} < 0.3 \) while compressibility effects are accounted for \( \text{Ma} > 0.3 \). However, this criterion is a necessary but insufficient condition to allow the flow to be considered compressible. At the microscale there is significant variation in the density of gases due to the large pressure drops resulting from the surface friction inside the microchannels [67]. If the compressibility of the flow can be neglected and the flow is developed the energy and the momentum equations are uncoupled, longitudinal temperature gradients are constant and longitudinal velocity gradients are zero. If these assumptions are not satisfied however, as is usually the in microchannel configurations, analytical or semi-analytical solutions are not possible and numerical simulations are required [68].

In gaseous flows the static temperature of the fluid decreases due to the conversion of thermal energy to kinetic energy. Therefore, the static temperature, or bulk temperature, of the gaseous flow in a microchannel is generally not a suitable characteristic temperature for analyzing the heat transfer. The total temperature, or stagnation temperature accounts for the total energy in the fluid flow. Two-stream gas-to-gas micro-heat exchangers were investigated numerically by Miwa et al. [69] for parallel-flow and counter-flow arrangements. Their results emphasized that the static temperature is not suitable for a characteristic temperature of heat transfer from the hot fluid to the cold fluid. The factors that determine the temperature distribution of the fluid are convective heat transfer, heat released due to viscous dissipation, and cooling due to expansion of the gas.

Viscous dissipation in microchannels may influence the fluid viscosity at the wall. Viscous dissipation effects increase rapidly with a decrease in channel dimensions [70]. Xu et al. [71] modeled viscous dissipation in microchannels and found that the velocity profile was modified due to viscous dissipation. Viscous effects significantly reduce the pressure along the flow direction thereby varying the gas density which changes both the velocity and temperature profiles which affect the heat transfer [72]. The heat released due to viscous dissipation can lead to flow instability [73], transition to turbulence [74], and oscillatory motions [75]. Due to a reduction in viscosity at higher temperatures, the resulting friction factors are predicted to be lower in liquids when viscous effects were present. Judy et al. [76] concluded that viscous heating can influence
heat transfer in microchannels depending on the boundary conditions and should be accounted for. For flows with a constant wall temperature boundary condition, the Nusselt number increased due to viscous heating effects to a number that is independent of the Brinkman number. For flows with a constant wall heat flux boundary condition, the Nusselt number decreases as the Brinkman number increases.

The heat flux from the wall of macroscale channels decreases monotonously along the channel length due to change in fluid temperature. Asako et al. [77] studied the effect of heat transfer characteristics of gaseous flows in microchannels and noted that the temperature fall due to the expanding gas result in additional heat transfer near the channel outlet when the flow is fast. In the case of large temperature differences, the additional heat transfer rate due to this temperature fall is small compared to the normal heat transfer rate. Asako [77] surmised that the heat transfer of gaseous flow in microchannels could be predicted from correlations for incompressible flow in a conventional sized channel when the temperature difference between the wall and the stagnation temperature of the fluid is greater than 50 K. For temperature differences less than 50 K, it was noted that the additional heat transfer rate becomes significant and the heat transfer for the gaseous flow in the microchannel could not be predicted from the incompressible macroscale correlations.

If the thermal conductivity of the gas is extremely high, the gas temperature at the outlet recovers to the wall temperature. In actuality however, only a slight recovery is observed. Accurate evaluation of the recovery factor is required when calculating the heat transfer rate from the difference in temperature between the bulk fluid temperature and the wall temperature. In the particular investigation of this dissertation the heat transfer rate was not calculated from temperature surface measurements. Instead, the heat transfer rate was determined by measurements of the total energy of the flow upstream and downstream of the heat exchanger. Due to the high flow speeds encountered, the effect of recovery factor on the thermocouples is discussed in Section 2.5.

Due to the relatively short lengths employed in microchannels, the influence of the entrance effects cannot be neglected. Entrance region effects in laminar heat transfer become more significant at higher Reynolds numbers [43]. In developing flow the velocity profile is changing with downstream position and the only component of velocity is in the axial direction. The friction factor is not constant when the velocity profile is changing. As a result, large frictional effects result from the difference in velocity between the wall and the core of the flow at the entrance of
the duct, and the friction factor is high. The friction factor diminishes throughout the entrance region as the flow develops and reaches an asymptotic value in the fully developed region.

In compact heat exchanger design consideration must be given to the fact that the fluid transport properties may vary considerably with temperature which can effect flow-friction and heat transfer results. Such property variations distort both velocity and temperature profiles. For gases, thermal conductivity, viscosity, and density all vary considerably with temperature. In liquids, the only property varying with the temperature is viscosity.

Since gas properties vary in a similar manner with absolute temperature, all fluid properties can be evaluated at the convenient mixed mean temperature and the effects of property variation over the flow section can be expressed as a function of the absolute temperature ratio \( T_m/T_w \). Analysis and experiment indicate that for most cases of interest the temperature-dependent properties at a given Reynolds number can be expressed as a simple power of this ratio. Values for the exponents are given for various situations based on flow conditions [78]. For fully developed turbulent flow in a circular tube with a heating gas \( N=0.5 \) and \( M=0.1 \) are recommended for use in adjusting the friction factor and Nusselt number in Eq. (1.11) and Eq. (1.12) respectively [79].

\[
\frac{f}{f_m} = \left( \frac{T_m}{T_w} \right)^M \\
\frac{N_u}{N_{u_m}} = \left( \frac{\mu_m}{\mu_w} \right)^N
\]

The only important property varying with temperature for a liquid is viscosity. The recommended equations for adjusting friction factor and Nusselt number are respectively given in Eq. (1.13) and Eq. (1.14). The recommended exponents in evaluation of these equations are \( N=0.14 \) and \( M=0.4 \) for a laminar flow of a cooling liquid [80].

\[
\frac{f}{f_m} = \left( \frac{\mu_m}{\mu_w} \right)^M \\
\frac{N_u}{N_{u_m}} = \left( \frac{\mu_m}{\mu_w} \right)^N
\]

Kandlikar et al. [81] investigated the effect of surface roughness on heat transfer and pressure drop of laminar flow in smooth and rough stainless steel tubes 1.07 and 0.62 mm in diameter. The results indicated that effect of changes in relative roughness on pressure drop were
minimal, but the heat transfer in the thermal entry region showed a distinct dependence on surface roughness. The relative roughness values for microchannels are expected to be higher than the limit of 0.05 used in the Moody diagram [82]. Kandlikar et al. [83] recommends using a constricted flow area when calculating the friction factor by considering the area reduction due to protruding elements. Using this reduction it was found that the friction factor plateaued to a value of 0.042 for relative roughness values $0.03 \leq \varepsilon/D_h \leq 0.05$ [43].

The transition from laminar-to-turbulent flow has been reported to occur in microchannels at Re $< 2300$. Schmitt et al. [83] showed that while a transition to turbulence in smooth rectangular channels occurred between 2000 and 2300, the transition Reynolds number in roughened channels were a function of the relative roughness. Equations were proposed to account the roughness effects based on their experimental data.

1.5 Two-phase Condensation in Rectangular Minichannels

Condensation heat transfer in microchannels and minichannels is of great practical significance in the development of next generation ultra-compact high power density thermal systems. Due to the electronics cooling industries interest in heat removal at high heat fluxes a considerable generation of literature on the effects of single-phase flow, pressure drop and heat transfer in pool and convective boiling in microchannels has been generated [1, 8, 9]. By contrast, there have been fewer studies conducted on the measurement of pressure drop and heat transfer coefficients during condensation in micro- and mini-channel geometries. Condensation and boiling are reciprocal processes similar in nature. However, nucleation sites are not present in condensation which account for a substantial difference in the underlying physics of the phenomena. The phase change process in microchannels with high aspect ratios yield the formation of thin film condensation layer on the heat transfer wall, resulting in high heat transfer coefficients with little pressure drop penalty. It is therefore of utmost importance that condensation phenomena, especially in high aspect ratio microchannels be experimentally investigated [84-86].

Convective heat transfer coefficients associated with two-phase flow are generally an order of magnitude larger than single-phase heat transfer coefficients under similar conditions [5]. Two-phase flow lends itself to applications where single-phase flow would impose high pressure losses and thus require a high pumping power. Phase-change at small scales is been generally estimated using correlations developed in macrochannels. However, conventional models may fail to account for flow phenomena and interfacial shear behavior specific to microchannel flows and can
therefore deviate significantly from the measured data [85]. Although prediction of two-phase flow characteristics in microchannels is an area of much recent research [10, 85-93] there is no consensus of a unified model in the literature.

Two-phase flows exhibit a high rate of heat removal at a nearly constant temperature giving them an advantage over single-phase flows in compact heat exchangers. Much like single-phase flows, two-phase flows are generally characterized by investigating inertial, viscous, and pressure forces, however additional attention to interfacial tension forces, wetting behavior of the liquid on the surface, and liquid-vapor momentum exchange in the flow are also relevant. Flow regimes in two-phase flows are much more complex than in single-phase flows. Single-phase flows are adequately distinguished in general by the evaluation of the Reynolds number to be in either the laminar, transitional, or turbulent flow regime. In two-phase flow regimes the interactions of the liquid-vapor interface are observed in reference to the mass flux, heat flux, quality, saturation conditions, and thermophysical properties of the fluid. Two-phase flow mechanisms in horizontal flows are also slightly different than in vertical flows due to stratification. In horizontal flows at very low quality, bubbly flow is often observed. As the quality increases, small bubbles coalesce to form plug-type bubbles. Stratified flow is observed at a higher qualities and low flow rates due to separation of vapor in the upper part of the tube. A further increase in quality and flow rate can introduce instabilities in the smooth liquid-vapor interface causing wavy flow. At high flow rates, the waves span across the entire width of the tube forming large slug-type bubbles. Apart from these flow regimes, annular flows are also observed at high vapor velocities and moderate liquid flow rates. Finally, mist flows and dry-out regions are observed at very high qualities in two-phase flow. Flow models and flow regime maps are available to predict pressure drop and heat transfer for two-phase condensation flow in horizontal tubes [43, 63, 87, 92].

Heat transfer coefficients in condensers increase with an increase in mass flux and quality and with a decrease in the tube diameter. The heat transfer coefficient decreases as the liquid film becomes thicker with a decreasing vapor quality. Actual heat transfer coefficients in a condenser are dependent on inlet conditions, temperature of the fluid, the inlet superheat, and the wall subcooling [91].

As the hydraulic diameter decreases the effects of surface tension supplants the effects of gravity as the dominant force in condensation flows. This facilitates plug and slug flow at high qualities as the hydraulic diameter is decreased. Surface tension stabilizes the waves. The increase
in surface tension has been attributed to an increase in annular flow patterns in smaller hydraulic diameter tubes [43].

The study of high aspect ratio microchannels is important in phase change inducing process since these channels may be manipulated by the design to yield formation of thin film condensation on the heat transfer wall resulting in high heat transfer coefficients with little pressure drop penalty. Al-Hajri et al. [84] conducted an experimental parametric study of two-phase condensation for R134a and R245fa in a single high aspect ratio rectangular microchannel with a hydraulic diameter of 0.7 mm (0.4 mm x 2.8 mm) and length of 190 mm. The study investigated the effects of saturation temperatures 30-70 °C, mass flux 50-500 kg/m²s, and inlet superheats 0-20 °C on the average heat transfer coefficient and overall pressure drop in the microchannel. The results of the study submit that for the same operating conditions refrigerant R245fa demonstrated 25% better heat transfer, but had almost 100% higher pressure drop than R134a.

1.6 Structure of Thesis

This dissertation is divided into the following six chapters:

Chapter 1 introduces the motivation and objectives of the work that will be explored in the experimental investigation. An overview of the current state-of-the-art in heat sink technology is given and basic relevant information regarding the thermal-hydraulic performance and scaling effects for single-phase and two-phase phenomena in microchannels is provided. Finally, the chapter highlights the novelty of the study and the impact of the work.

Chapter 2 discusses the engineering design and manufacturing processes of employed for fabricating the heat exchangers through novel electro-discharge-machining. The fabricated device characteristics are provided and expected performance is modeled using conventional and finite element methods. The experimental apparatus for air-side and refrigerant-side test facilities are described, experimental methods, data reduction methodologies, and uncertainty in experimental data are also presented.

Chapter 3 reports the thermal-hydraulic analysis for single-phase gas flow in circular microchannels, single-phase liquid flow in rectangular mini-channels. The chapter provides an overview of the experimental performance data, reduction methods and presents the experimental pressure drop and heat transfer results of the study. Measured air-side and refrigerant-side pressure drop data is decomposed into constitutive components and experimentally obtained friction factors
are compared to the prevailing models and theories in the literature. Average heat transfer coefficients are determined using the air-side convection coefficients determined by the modified-Wilson plot method from the single phase to obtain two-phase convection coefficients and compared to the predictions in the literature. The overall thermal performance of the devices are compared to the conventional models using theoretical predictions and a 3-D finite element simulation of the performance. Finite element performance is evaluated by application of the experimentally obtained boundary conditions to a computer aided design (CAD) model of the heat exchangers evaluated in COMSOL Multiphysics.

Chapter 4 reports the performance and thermal-hydraulic analysis for the compact heat exchanger devices with phase change condensation on the refrigerant-side. The chapter provides an overview of the experimental performance data, reduction methods and presents the experimental pressure drop and heat transfer results of the study. Measured refrigerant-side and pressure drop data is decomposed into constitutive components and compared to the prevailing models and theories in the literature. Average heat transfer coefficients are determined from the air-side convection coefficients determined by the modified-Wilson plot method from the single phase to obtain two-phase convection coefficients and compared to the predictions in the literature. The overall thermal performance of the devices are compared to the conventional models using theoretical predictions and a 3-D finite element simulation of the performance. Finite element performance is evaluated by application of the experimentally obtained boundary conditions to a computer aided design (CAD) model of the heat exchangers evaluated in COMSOL Multiphysics.

Chapter 5 introduces scaling the techniques employed in this research in a 10 cm$^3$ heat exchanger which will be the subject of future work. The fabricated device is shown and the expected performance of the heat exchanger configuration is simulated by applying convection coefficients to the given geometry in COMSOL Multiphysics.

Chapter 6 summarizes the scientific impact and significance of the research effort. Conclusions and recommendations for future directions in the field of high power density heat sink performance are provided.
CHAPTER 2: DESIGN AND EXPERIMENTAL METHODS

2.1 Overview

Flow passage dimensions in convective heat transfer applications have been diametrically shifting towards smaller dimensions to fulfill the growing demand for high heat transfer performance in power electronics. The shift toward higher heat loads and smaller hydraulic diameters highlights the demand for advanced manufacturing to produce compact devices and the need to design and characterize systems at these compact scales. This chapter outlines the engineering design, manufacturing, and design of experiments for characterization of compact air-cooled heat exchangers with dielectric refrigerant heating. The design space is defined by applying well understood design methodologies using macroscale correlations in a one-dimensional fin model of the heat exchanger. A three-dimensional heat exchanger simulation was performed to validate the one-dimensional resistance model by applying boundary conditions to a computer aided design models of heat exchangers in using COMSOL. The advanced manufacturing of high density high aspect ratio micro heat exchangers are described. An overview of the engineering design concept, heat transfer model, advanced manufacturing process, and experimental methods are presented. The physical characteristics, manufacturing tolerances, and surface roughness characterization of the fabricated devices are provided.

2.2 Modeling Heat Exchanger Performance

The air-side compact heat exchanger design space was initially explored by utilizing experimental correlations for the case of gas flow inside parallel circular tubes with abrupt-contraction entrances published by Kays and London [78]. Kays and London aggregated a combination of analytical and experimental flow data and published curves providing the friction factor, $f$, and Colburn factor, $j$, as a function of Reynolds number to predict the pressure drop and heat transfer coefficients respectively. The generalized curves are distinguished by the length to diameter ratio, $L/D$. The heat transfer data was further specified for a constant temperature versus a constant heat flux boundary condition. Fluid property adjustments to these results are applied via the property ratio method discussed in Chapter 1. The convection coefficient is determined from the Colburn factor, and Stanton number, using Eq. (2.1)-(2.3). The pressure drop is determined from the friction factor using Eq. (2.4). Figure 2.1 shows the pressure drop and heat transfer

18
coefficient predictions for a range of circular microchannel diameters as a function of Reynolds number based on the Kays and London experimental correlations. Smaller channel sizes are projected to increase performance and cost of operation via overall pressure drop.

\[
j = StPr^{2/3}
\]

(2.1)

\[
St = \frac{Nu}{RePr}
\]

(2.2)

\[
h = \frac{Nuk}{D_h}
\]

(2.3)

\[
\Delta p = f \frac{L \rho v^2}{D_h 2}
\]

(2.4)

Figure 2.1 Shows the predicted A) pressure drop and B) heat transfer coefficient for different circular channel diameters as a function of Reynolds number as predicted by Kays and London experimental correlations.

Two-phase flow performance was estimated using convectional correlations for condensation in macrochannels. In their experimental investigation of condensation heat transfer coefficients for circular microchannels Bandhauer et al. [93] concluded that many of the available macro- models and correlations over-predict their pressure drop data and under-predict their heat transfer data for micro- and mini- channels. Since it was considered prudent to over-estimate the pressure drop and under-estimate the convection coefficient for the sake of over designing the devices to meet performance metrics, the large tube models were implemented. The refrigerant-side design space was defined using the classical Friedel [94] and Akers [95] models to predict the respective pressure drop and heat transfer performance for two-phase condensation of di-electric
refrigerants. Refrigerant R134a and R245fa were both considered during this preliminary investigation. Refrigerant R245fa was ultimately chosen based on its favorable thermal characteristics and low saturation pressure. At a saturation temperature of 80 °C R245fa and R134a have respective saturation pressures of 781 and 2635 kPa. Furthermore, R245fa exhibits favorable heat of vaporization, it is nontoxic, nonflammable, noncorrosive, and environmentally benign. While R134a is widely used and exhibits a favorable specific heat it was not considered to be of practical application given the demand in saturation conditions in this investigation. A recent performance study showed that for condensation in rectangular microchannels refrigerant R245fa demonstrated 25% better heat transfer but had almost a 100% higher pressure drop than R134a [84].

A theoretical analysis was conducted considering R245fa at a saturation temperature of 80 °C flowing through a single microchannel with a hydraulic diameter of 800 µm. The absolute value of the frictional pressure gradient was determined as a function of quality for a range of mass fluxes using the Freidel model as seen in Figure 2.2A. The pressure drop increases with increases in mass flux and quality. The heat transfer coefficient was determined using the Akers model as a function of quality for a range of mass fluxes as seen in Figure 2.2B. The heat transfer coefficient increases with an increase in mass flux and quality. Based on this analysis, we expect to achieve heat transfer coefficients in excess of 10,000 W/m²K on the refrigerant-side for two-phase condensation.

![Figure 2.2](image)

**Figure 2.2** A) The predicted frictional gradient predicted by Friedel model and B) convection coefficient predicted by Akers model for refrigerant R245fa in an 800 µm channel is shown.
2.2.1 Model Comparison

As explained in the previous section, the air-side heat exchanger convection coefficients are modeled using Kays and London experimental correlations and the two-phase refrigerant-side heat transfer coefficients are established using the predictions of the Akers model. The heat transfer coefficients from these models are applied using a 1-D fin analysis considering the respective convection coefficients, transfer areas, fin efficiency and conduction resistance of the geometry in order to determine the overall heat transfer coefficient. The overall heat transfer coefficient is used to evaluate the heat exchanger performance. In this manner we may rapidly evaluate the heat transfer performance for many heat exchanger geometries. The results of this analysis allow heat transfer and pressure drop capacities to be considered against manufacturing capabilities in order to select optimal parameters for the heat exchanger performance.

The model for heat transfer in the devices is shown in Figure 2.3A. Heat is convectively transported from the refrigerant, conducted through the solid copper walls and transferred via convective transport to the air. In the configuration shown the heat transfer model of the heat exchanger is comprised of 4 air-side fins with adiabatic tips of length \( L_f \) and fin efficiency \( \eta_f \). A temperature distribution of a 2-D cross section of a simulated device for the modeled configuration is presented in Figure 2.3B. Convective boundary conditions at the fluid temperature are prescribed to the respective air-side and refrigerant-side surfaces. The top and bottom surfaces are adiabatic. This model is mathematically represented by the total thermal resistance model given by Eq. (2.5) which is used in conjunction with the log mean temperature difference of the air and refrigerant transfer fluid to determine the heat dissipation rate in the device with Eq. (2.6). Equations (2.6)-(2.8) may be iterated to determine the temperature of the air leaving the exchanger.

\[
R_{tot} = \frac{1}{h_{ref} A_{ref}} + \frac{L}{k_W A} + \frac{1}{\eta_f h_{air} A_{air}} \tag{2.5}
\]

\[
Q = \frac{\Delta T_{LM}}{R_{tot}} \tag{2.6}
\]

\[
\Delta T_{LM} = \left(\frac{T_{ref,sat} - T_{air,in}}{T_{ref,sat} - T_{air,in}}\right) - \left(\frac{T_{ref,sat} - T_{air,out}}{T_{ref,sat} - T_{air,out}}\right) \tag{2.7}
\]

\[
T_{air,out} = T_{air,in} + \frac{Q}{\dot{m} c_p} \tag{2.8}
\]
For a given heat exchanger volume there is an intrinsic tradeoff between utilization of volume on the air-side versus refrigerant-side. For optimal utilization of volume, it is recommended that the overall thermal resistance of the air-side and refrigerant-side be well matched [12]. Two-phase heat transfer coefficients on the refrigerant-side are predicted to be 3-5 times larger than those on the air-side, therefore the air-side should have correspondingly larger transfer area.

Internal finning of the heat exchanger is considered by investigating different configurations of refrigeration cross channels within the confines of 1 cm³ cube. Figure 2.4 depicts the effect of adding more fins with shorter lengths for devices with 400 μm and 600 μm diameter channels with center-to-center pitch offsets as a function of air-side Reynolds number. This illustrates the advantage of employing more fins with shorter lengths to optimize fin efficiency on the air-side. The advantage of doing this needs to be weighed against the disadvantage of having smaller transfer area due to the manufacturing tolerance which must be maintained between the two sides. The tolerance ensures there is no leakage between the air-side and refrigerant-side channels, which would render the devices inoperable. Smaller channel sizes are were to increase performance but also increase cost of operation and manufacturing.
Figure 2.3 A) The heat transfer model is comprised a heat exchanger having 4 fins with adiabatic tips of length $L_f$. B) The cross sectional temperature distribution of the modeled heat exchanger.
Figure 2.4 The of fin efficiency as a function of Re is shown for 400 and 600 μm diameters on a 450 and 650 μm pitch for different fin lengths.

Figure 2.5A shows the expected power for a heat exchanger with air-side holes 400 μm in diameter and a pitch offset of 450 μm using the 1-D model. This configuration is evaluated for a range of refrigerant heat transfer coefficients from 5000-14000 W/m²K as a function of air-side convection coefficients. The merits of a single refrigeration cross channel were compared to the double cross channels configuration, clearly demonstrating the relative enhancement of finning the device. The higher fin efficiencies of the shorter fins contribute to the enhanced overall heat transfer performance.

The power density performance of the heat exchangers are evaluated by investigating the steady state heat transfer response using COMSOL Multiphysics. The same heat transfer coefficients and temperature boundary conditions applied in the 1-D model are applied to three-dimensional solid model created using computer aided design software (SolidWorks) to investigate multidimensional heat transfer effects. A surface area integration of the temperature distribution is employed to provide the overall convective heat transfer, or dissipated power, of the 3-D heat exchanger. Several internal finning configurations of different heat exchangers were explored in COMSOL. Figure 2.5B-C displays the steady-state temperature distribution of the model with single and double refrigerant-side cross channels. The power density performance predictions of the 1-D and the 3-D model for heat exchangers with a single and double refrigeration channel are compared in Table 2.1 for different boundary conditions. Both of these models have the same total transfer area and the same volume. The circular air-side holes are 400 μm on 450 μm pitch offset.
Figure 2.5. A) Power density for an air-side channel diameter of 400 µm over a range of refrigerant side convection coefficients are shown as a function of the air side convection coefficient for 1 and 2 refrigeration cross channels (CC). Heat exchangers with B) one cross channel and C) two refrigerant side cross channel were modeled in COMSOL using the same boundary conditions applied in the 1-D model.

The model with a single refrigerant cross channel is effectively partitioning the air-side of the heat exchanger into two fins with adiabatic tip boundary conditions while the model with two cross-channels is effectively partitioning the air-side of the heat exchanger into four fins with adiabatic tips conditions. Air-side boundary conditions of 2000 W/m²-K at 30 °C and refrigerant channel boundary conditions 5-15 kW/m²-K at 80 °C were applied to these geometries.
Table 2.1 Comparison between 1-D fin model to the 3-D COMSOL model for several air side and refrigerant side boundary conditions.

<table>
<thead>
<tr>
<th>Boundary Conditions</th>
<th>Single-Channel</th>
<th>Double Channel</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>h&lt;sub&gt;air&lt;/sub&gt;</td>
<td>T&lt;sub&gt;air&lt;/sub&gt;</td>
</tr>
<tr>
<td>A</td>
<td>2,000</td>
<td>30</td>
</tr>
<tr>
<td>B</td>
<td>2,000</td>
<td>30</td>
</tr>
<tr>
<td>C</td>
<td>2,000</td>
<td>30</td>
</tr>
</tbody>
</table>

The 3-D COMSOL models show agreement with the 1-D resistance model to within 10% for the range of tested geometries and convection coefficients. Increasing the number of refrigeration cross channels increased the air-side fin efficiency of the heat exchanger at the expense of having less air-side channels for cooling. This is a compromise in the number of air-side holes due to the need to maintain manufacturing tolerances between the two-sides to avoid mixing of the fluids. The heat exchanger with two 0.5 mm x 9 mm channels exhibited > 40% higher heat transfer than a heat exchanger with the same transfer area and boundary conditions with a single 1 mm x 9 mm channel. This performance enhancement showed strong agreement with the 1-D model.

Achieving the heat transfer performance predicted in the heat transfer models is predicated on achievement of high speed air flow in the microchannels. While gas rarefaction is not relevant at this scale, compressibility effects, and viscous effects are significant. A 1 cm long column of air was simulated in a computational fluid dynamics program (ANSYS Fluent) for 400 and 600 μm diameter channels for the case of an adiabatic wall and an isothermal wall at 80 °C to solve the compressible Navier-Stoke equations. Since continuum flow conditions are valid, the no-slip velocity boundary condition and the wall temperature conditions are applied. Figure 2.6 shows a cross section of the resulting static pressure, static temperature, velocity magnitude, and fluid density profiles for a 400 μm diameter air column. The upstream stagnation pressure in the simulation is 150 kPa at an inlet temperature of 293 K and the downstream condition is a static pressure of 101 kPa. The results of this analysis show that for the conditions considered the friction factors for the air-side column were over the predictions for the friction factor provided by Kays and London for inlet static pressures up to 150 kPa. The friction factors resulting from simulations of the isothermal wall condition were approximately 10% higher than those with the adiabatic wall boundary condition. The friction factors provided in this analysis were utilized in determination of the predicted pumping power required for the devices.
Based on the results of the initial investigation, a heat exchanger design configuration utilizing two-cross flow channels was chosen. Figure 2.7 depicts the cross flow compact heat exchanger design concept. The heat exchanger utilizes two unmixed working fluids in a cross flow configuration. Air flows through the circular microchannels and a dielectric refrigerant flows through the rectangular minichannels in cross-flow. The air-side consists of an array of circular microchannels of diameter, $D$, and a center-to-center pitch offset spacing, $P$. The refrigerant side consists of parallel rectangular minichannels having a cross sectional aspect ratio of $\alpha=a/b$.

![Diagram](image)

Figure 2.6 A) 1 cm long circular air column with a diameter of 400 μm is simulated using FLUENT for an A) adiabatic wall and an B) isothermal wall at 80 °C. A 2-D cross section of the static pressure, static temperature, velocity magnitude, and fluid density are plotted as a function of the channel length for the case where the upstream stagnation pressure is 150 kPa at an inlet temperature of 293 K and the downstream outlet condition is 101 kPa. The no-slip condition at the channel walls is employed.
Figure 2.7 The cross flow compact heat exchanger design concept. A) Air flows through the circular microchannels and a dielectric refrigerant flows through the rectangular minichannels. B) The air side consists of an array of circular microchannels of diameter $D$ and a center-to-center offset $P$. The refrigerant side consists of parallel rectangular minichannels with a cross sectional aspect ratio $\alpha = a/b$. 
2.3 Micro-EDM Manufacturing of Micro-Heat Exchangers

Micro-electro-discharge machining (micro-EDM or μEDM) is a non-traditional machining technology that has been gaining popularity as a new alternative for fabricating microstructures. The non-contact process requires little force between electrode and work piece and is capable of machining ductile, brittle, or super hardened-materials. The main advantages of μEDM are its low cost set up, high accuracy, and large design freedom. Compared to etching or deposition techniques μEDM has the advantage of being able to create complex three-dimensional shapes with high aspect ratios. There are many operating parameters that affect the μEDM process which can make it difficult to maintain high accuracy of these features. The fabrication of micro-electrodes employed by the machine is also an important parameter and active area of research development [96]. With appropriate parameter regulation, it is possible for μEDM to achieve high precision and high quality machining.

Micro-EDM can be used to make microstructures from any electrically conductive material including metal alloys. While copper has desirable thermal characteristics, it is notoriously difficult to machine and EDM. The heat exchangers utilized in this investigation are manufactured from tellurium copper alloy (C14500). The addition of tellurium enhances the strength of copper allowing for higher machining speeds and a longer tool life than pure copper, while preserving the alloys desirable thermal characteristics. Copper C14500 has a machinability rating scale of 85% compared to pure copper which has a rating of 20%. The C14500 alloy has a thermal conductivity of 354.6 W/m²-K compared to pure copper which has a value of 396.9 W/m²-K at 20 °C. The composition of the C14500 alloy is > 99.9% copper, silver, and tellurium, 0.4-0.6% of which is tellurium, and 0.004-0.012% phosphorus by weight [97].

High aspect ratio parallel microchannels were simultaneously created in a high-density honeycomb arrangement with sinker μEDM machining using a single carbon electrode tool. The carbon electrode tool has a shape that is the inverse of the desired channel shape and length. The non-contact nature of μEDM makes it possible to use a very long and thin electrode for the machining. The electrode tool is brought near the surface of the conductive part separated by a dielectric liquid. Plasma discharge consumes some of the tool and part as an electrical current pulse is driven across the gap in the dielectric liquid [40]. An increase in pulse energy increases the plasma temperature, which alters in the heat distribution pattern between the electrodes, and in turn increases the tool wear rate [41].
Conventional μEDM typically has a high electrode wear rate ratio and low material removal rate. The wear of the electrode must therefore be compensated by either changing the electrode or by preparing longer electrodes from the beginning of fabrication. Changing the electrode during machining is not considered feasible for producing reliable structures since re-clamping the tool electrode reduces accuracy and adds to the set-up time. Proprietary advancements at EDM Department Inc. have directly led to minimization of the voltages employed during the machining of these devices which in turn minimized the tool wear rate and allowed the devices to be made with a single carbon tool electrode.

Minimization of the tool wear rate allows a high-density honeycombed array of high-aspect-ratio microchannels to be manufactured into a copper block yielding very high surface area per volume heat exchangers. The circular air-side microchannels are formed first and the rectangular minichannels are formed second employing the same methodology with rectangular electrodes. A characteristic overview of the two designs are summarized in Table 2.2 and values for the transfer area, cross sectional area, surface to volume ratio ($A/V$) and aspect ratio ($L/D_h$) are given based on measurements taken with an optical microscope.

<table>
<thead>
<tr>
<th>Device</th>
<th>Mean Hole Diameter [μm]</th>
<th>Std. Dev. of Diameter [μm]</th>
<th>Mean Pitch Offset [μm]</th>
<th>Std. Dev. of Pitch Offset [μm]</th>
<th>Number of Air Channels</th>
<th>Number of Refrigerant Channels</th>
<th>Surface Area/Volume [m2/m3]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>520.2</td>
<td>57.4</td>
<td>640.8</td>
<td>15.7</td>
<td>150</td>
<td>8</td>
<td>3370</td>
</tr>
<tr>
<td>B</td>
<td>355.1</td>
<td>17.4</td>
<td>450.9</td>
<td>25.8</td>
<td>300</td>
<td>8</td>
<td>4270</td>
</tr>
<tr>
<td>C</td>
<td>482.4</td>
<td>20.23</td>
<td>671.0</td>
<td>24.9</td>
<td>600</td>
<td>14</td>
<td>2111</td>
</tr>
</tbody>
</table>

Table 2.2 Summarizes the two 1-cm$^3$ and one 10-cm$^3$ heat exchanger designs manufactured using μEDM manufacturing.

As seen in Figure 2.8 Device A comprises 150 circular air-side channels with an average diameter of 520 μm configured in a high density honeycomb array with an average center-to-center offset pitch spacing of 640 μm and Device B comprises of 300 circular air side channels with an average diameter of 355 μm configured in a high density honeycomb array with an average center-to-center pitch offset spacing of 450 μm. Both Device A and B have refrigerant-side geometries comprised of 8 minichannels configured into two rows of 4 horizontal rectangular minichannels separated by 3.5 mm. Each refrigeration channel has a cross sectional aspect ratio of 0.5 mm x 2 mm and is separated by 0.5 mm center-to-center pitch offset. Device A has a surface area to volume ratio of 3370 m$^2$/m$^3$ and Device B has a surface area to volume ratio of 4270 m$^2$/m$^3$. 
ratio of 4270 m$^2$/m$^3$. Aspect ratios of the air-side channels are 19.61 and 28.7 for Device A and Device B respectively.

Figure 2.8 Photographs of 1 cm$^3$ devices and air-side microscope images. A) Device A and B) Device B. C) Photographs of the assembled manifold (left) air side and (right) both sides.
Figure 2.9 shows the surface roughness profile of a μEDM manufactured air-side channel cross section and a summary of the roughness profile parameters which was optically characterized using a profile roughness measurement tool (Alicona). A 6.2 mm scan of a cross section has an average roughness profile of 2.09 μm, a root mean square of the roughness profile of 2.6 μm, maximum peak height of 8.0 μm, maximum valley height of 8.7 μm, the maximum peak to valley height observed in the roughness profile is 16.8 μm and the mean spacing of profile irregularities is 163.84 μm. Kandlikar et al. [83] proposed a method for evaluating the equivalent roughness of a surface profile, ε by adding mean of the maximum profile peak height and the floor distance to the mean line. The measured mean of the maximum profile peak height is 8.7 μm and the floor distance to mean line was graphically estimated from the profile scan to be 7.1 μm. Based on this evaluation methodology the equivalent roughness of the scanned profile is assessed to be ε = 15.8 μm. Given this approach the relative roughness, ε/Dh, for Device A and Device B are 0.03 and 0.045 respectively.
Figure 2.9 Surface roughness profile for the inside of a circular cross section 6.2 mm in length is shown along with a table of summarizing the characteristic parameters of the roughness profile.

2.4 Experimental Design

The heat exchanger performance is experimentally characterized by supplying air and refrigerant flows at various operating temperatures and air flow rates for single-phase and two-phase refrigerant heating. The test facility can be utilized to characterize the performance of fabricated heat exchangers and investigate hydraulic and heat transfer phenomena at previously unexplored ranges. The test bench provides versatility intended for performing heat transfer and fluid flow research on different heat exchanger geometries over a wide range of experimental conditions. The air-side and refrigerant-side facilities were constructed using macro-scale tubes that integrate to various components of the system. The test facility incorporates high-accuracy compact data acquisition (cDAQ’s) interface the hardware with LabView (National Instruments).
LabView is used to simultaneously operate the experimental equipment and read all of the sensors. The cDAQs were integrated into a 9174 4-slot USB chassis designed for small, portable, mixed-measurement systems. The thermocouple readings were measured by two 9213 mV reading cDAQs. Pressure, density, and mass flow rate measurements were taken with a 9208 mA reading cDAQ. The electronic pressure regulator actuator is controlled by a 9263 voltage writing cDAQ and monitored with a voltage reading 9215 cDAQ. All experimental data in this investigation was obtained a rate of 1 Hz.

2.4.1 Air-Side Test Facilities

The air-side line is depicted schematically in Figure 2.10A. High pressure air was generated by two compressors that is treated by respective air dryers and fed into a 142 m³ compressed air storage tank farm. The two compressors are operated in parallel and have respective mass flow rate capacities of 0.7 kg/s and 0.75 kg/s that are capable of supplying > 1 kg/s of dried air at a constant pressure of 689.5 kPa. The air flow was accessed in the laboratory by a series of two 4” gate valves. The air-line was reduced from 4” down to a 1/2” with a series of reducing pipe fittings. Compressed air was sanitized with a particle filter to remove particles down to 5 µm and a coalescing filter to remove oil and mist down to 0.01 ppm and particles down to 0.01 µm. The filters each have maximum flow capacities of 51.9 liters/s at 689.5 kPa such that they would not inhibit air flow to the test section. A high flow precision electronic air regulator with a regulating pressure range from 0 – 862 kPa and a flow capacity of 47.2 liters/s at 689.5 kPa was installed upstream to control the upstream pressure during experimentation.

The air mass flow rate was measured with a CMF 100 Elite Series Coriolis-effect flow sensor (Micro Motion Inc.) along with a 2700 series transmitter. The flow meter was placed upstream of the manifold and test section in order to improve accuracy of the measurement. Coriolis-effect flow meters are more accurate when measuring high density gases than low density gases. Air flow is controlled with a downstream valve and air is exhausted through a high flow muffler with a maximum flow capacity of 43 liters/s. The muffler is employed to dampen the noise generated by air and has a noise reduction capacity of 17-22 dB.

Pressure and temperature measurements were made with absolute pressure transducers and T-type sheathed thermocouples. The thermocouples are calibrated using an ice-point bath to improve the relative uncertainty to ± 0.1°C from the manufactures rating of ± 1.0°C. The absolute pressure and static temperature in the flow were measured upstream and downstream of the test
section at points A and D respectively. Temperature measurements were also made upstream and
downstream of the manifold at points B and C, respectively. The uncertainty of temperature,
pressure, and mass flow rate measurements are given in Table 2.3. The components were
connected by tube compression fittings (Swagelok) used on 9.5 mm OD, 0.8 mm thick smooth
copper tubes cut to length. The entire apparatus was wrapped with 12.7 mm thick foam rubber
insulation to reduce heat leaks.

Figure 2.8C, shows how the manifold was designed to integrate with the heat exchanger
into the test facility. The manifold is comprised of four-pieces each consisting of steel cage with
Garolite interfaces capable of withstanding the high pressures and temperatures. Garolite has
excellent chemical resistance at high temperatures, high impact strength and a low thermal
conductivity, all of which serve to thermally insulate the heat exchanger an isolate it from its
surroundings. Teflon gaskets were hand-crafted using Teflon tape to interface between the Garolite
seal and the heat exchanger to ensure a hermetic seal between the two interfaces. The heat
exchangers were manually assembled into the test manifold using a custom jig designed to ensure
proper alignment. The heat exchanger-manifold assembly was held in place by threaded rods,
washers, and hex nuts used to keep the seals under a compressive load. The manifold
accommodates a 1 cm3 exchanger and can be connected to and disconnected from the respective
air and refrigerant side apparatuses with relative ease using the compression fittings connections.
Each part of the manifold has 3.175 mm ports for sheathed thermocouples to be attached.

The thermophysical properties of air including specific heat, viscosity, conductivity, and
Prandtl number were evaluated as a function of temperature at the respective points in the flow
using the embedded functions in engineering equation solver assuming air behaves as an ideal gas.

2.4.2 Refrigerant-Side Test Facilities

A closed recirculating loop was utilized to supply refrigerant flow to the heat exchanger.
Figure 2.10B depicts the refrigerant loop design concept schematically. The refrigerant was
circulated using a magnetically driven GB series gear pump (Micropump Inc.) coupled with a P35
gear set coupled to a 1 HP motor (Baldor Inc.). A VS1ST series variable frequency drive, (VFD),
was used to control motor rpm, thereby controlling the mass flow rate of the circulating refrigerant
in the loop. The refrigerant mass flow rate and density were measured with a CFM 025 Elite Series
Coriolis-effect flow sensor (Micro Motion Inc.) along with a 2700 series transmitter. Refrigerant
was heated in a 3 kW capacity brazed-plate heat exchanger evaporator (GEA Process Engineering
The evaporator was heated by an external 3 kW capacity circulator bath (Thermo Scientific Inc.) using Sil-180 silicone oil as a working fluid which can operate at temperatures up to 200°C. The fluid was expanded in a throttling valve used to lower the pressure upstream of the test section. The static temperature and gage pressure were measured upstream and downstream of the test section at points 1 and 2 respectively. A second valve throttling was placed downstream of the test section to further expand the fluid. The flow passes into a 1 kW capacity brazed-plate heat exchanger condenser (GEA Process Engineering Inc.) in order to subcool the refrigerant. The condenser was connected to an external 1 kW recirculating chiller (Thermo Scientific Inc.) using 50/50 ethylene glycol/water mixture as a working fluid. A sample cylinder (Swagelok) with an internal volume of 1 liter, was used as a receiver in the closed loop. The system is charged and discharged with refrigerant R245fa via a service valve upstream of the gear pump. A single zone refrigerant monitor (Kele) was located in the laboratory to sense for refrigerant leaks.

The flow circulates from the exit of the pump, through the flow sensor, evaporator, test section, condenser, receiver, and back to the pump inlet. The fluid temperature was measured using T-type sheathed thermocouples placed at the inlet of the brazed plate evaporator at point 1, upstream of the test section at points 2 and 3, downstream of the test section at points 4 and 5, and downstream of the brazed plate condenser at point 6. The gage pressure was measured upstream of the evaporator at point 1, upstream of the test section at point 2, and downstream of the test section at point 3. A differential pressure transducer was placed on either side of the test section at point 2 and 5. The absolute pressure in the laboratory was monitored using an external pressure transducer. The uncertainty of temperature, pressure, and mass flow rate measurements are given in Table 2.3. The components were connected by tube compression fittings (Swagelok) used on 9.5 mm OD, 0.8 mm thick smooth copper tubes cut to length. The uncertainty of temperature, pressure, and mass flow rate measurements are given in Table 2.3. The components were connected by tube compression fittings (Swagelok) used on 9.5 mm OD, 0.8 mm thick smooth copper tubes cut to length. The entire apparatus was wrapped with 12.7 mm thick foam rubber insulation to reduce heat leaks.

The thermophysical properties of R245fa including specific heat, density, viscosity, thermal conductivity and Prandtl number were evaluated using the embedded functions of engineering equation solver (EES) as a function of pressure, temperature, and quality.
<table>
<thead>
<tr>
<th>Variable</th>
<th>Calibration Range</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>0 to 100°C</td>
<td>± 0.1°C</td>
</tr>
<tr>
<td>Air Pressure</td>
<td>0 to 690.5 kPa</td>
<td>± 0.25 % FS</td>
</tr>
<tr>
<td>Air Mass Flow Rate</td>
<td>5 to 56 g/s</td>
<td>± 1 %</td>
</tr>
<tr>
<td>Refrigerant Pressure</td>
<td>-101.3 to 1034.2 kPa</td>
<td>± 0.25 % FS</td>
</tr>
<tr>
<td>Refrigerant Pressure Drop</td>
<td>0 to 6.9 kPa</td>
<td>± 0.25% FS</td>
</tr>
<tr>
<td>Refrigerant Mass Flow Rate</td>
<td>0 to 605 g/s</td>
<td>± 0.1 % FS</td>
</tr>
</tbody>
</table>

Table 2.3 Measurement uncertainty of instruments.
Figure 2.10 Schematic of experimental setup for A) air-side and B) refrigerant-side test facilities.

2.4.3 Experimental Procedure

Air flow to the laboratory was supplied by two 300 HP compressors (Ingersoll Rand) which were accessed through a series of mechanical and pneumatically actuated valves. The upstream
air-side pressure was set using an electronic regulator, and the back pressure and flow rate were controlled by a precision gate valve downstream of the test section.

Single-phase liquid conditions were attained on the refrigerant-side by adjusting the system charge between 2.4 – 2.6 kg. Adjustments to the charge were made to ensure the system pressure exceeded the saturation pressure at the given operating temperature. Under this condition the refrigerant was subcooled at the inlet and outlet ensuring liquid-only single-phase flow. The requisite amount of charge depends on the saturation temperature, refrigerant mass flux, and heat load on the air-side. Two-phase conditions were achieved by using less charge in the system than in single-phase flow, between 2.3 – 2.4 kg, to reduce the pressure in the system such that it was lower than the saturation pressure, and the refrigerant was observed to be either two-phase or superheated at the inlet.

The desired experimental refrigerant-side conditions were realized by making a series of adjustments to the system. Results were obtained by adjusting refrigerant charge of the system, refrigerant mass flux and temperature in the evaporator.

The refrigerant charge was adjusted via an access valve connected to a supply tank of R245fa. The refrigerant mass flow was adjusted using the variable frequency drive to control the rate at which the motor ran the gear pump, between 2 – 15 Hz. The silicone oil temperature on the recirculating heater was adjusted from values of 65-95 °C to control the temperature and saturation conditions in the system. The throttling valves were adjusted by lowering the pressure and increasing the temperature of the refrigerant expanding across the valve. The recirculating chiller was set to run with ethylene glycol water mixture at 20 °C had a thermal capacity of 1 kW to ensure the refrigerant was fully subcooled to avoid cavitation in the gear pump.

To obtain single-phase heat transfer results the inlet and outlet refrigerant temperatures were kept at least 5 °C subcooled to ensure that no vapor bubbles were present in the system which could lead to two-phase enhancements in the device. An energy balance between the air-side and refrigerant-side was confirmed to ensure that the refrigerant-side was not exhibiting any two-phase characteristics. An indication of two-phase flow would manifest itself in the energy balance between the two fluids. An energy balance between the two fluids was maintained throughout experimentation to ensure single-phase conditions prevailed.

For two-phase flow experimentation the refrigerant-side saturation conditions at the inlet and outlet of the heat exchanger were kept slightly superheated. The degree of superheat was
monitored using pressure and temperature measurements in the manifold. Flow entering the heat exchanger was only slightly superheated while the refrigerant exiting the exchanger was either be two-phase flow or slightly subcooled. The degree of superheat at the inlet was minimized such that the flow exiting the heat exchanger would not be subcooled. These conditions were evaluated for the compact condenser geometries in Table 2.4. It is important to note that the areas given in the table are adjusted from the total areas of the manufactured exchangers to effective area participating in heat transfer and fluid flow. This adjustment is necessary since some of the channels are obstructed when the device is connected in the manifold.

<table>
<thead>
<tr>
<th></th>
<th>Device A</th>
<th>Device B</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Air-Side</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$D_h$</td>
<td>520 μm</td>
<td>355 μm</td>
</tr>
<tr>
<td>$N_T$</td>
<td>150</td>
<td>300</td>
</tr>
<tr>
<td>$A$</td>
<td>18.04 cm$^2$</td>
<td>24.07 cm$^2$</td>
</tr>
<tr>
<td>$A_c$</td>
<td>21.24 mm$^2$</td>
<td>20.09 mm$^2$</td>
</tr>
<tr>
<td>$L/D_h$</td>
<td>19.61</td>
<td>28.7</td>
</tr>
<tr>
<td><strong>Refrigerant-side</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$D_h$</td>
<td>800 μm</td>
<td>800 μm</td>
</tr>
<tr>
<td>$N_T$</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>$A$</td>
<td>6.0 cm$^2$</td>
<td>6.0 cm$^2$</td>
</tr>
<tr>
<td>$A_c$</td>
<td>8.0 mm$^2$</td>
<td>8.0 mm$^2$</td>
</tr>
<tr>
<td>$L/D_h$</td>
<td>12.8</td>
<td>12.8</td>
</tr>
</tbody>
</table>

Table 2.4 Summary of the air-side and refrigerant-side characteristics of 1 cm$^3$ manufactured heat exchangers.

2.5 Experimental Methods

The law of conservation of energy is the first law of thermodynamics, it can be mathematically defined for an open system with multiple inlets and outlets by Eq. (2.9). This mathematical representation of the Navier-Stokes energy equation deals with rates of energy flow and is applicable at an instant in time. The equation physically states the time rate of change of total energy within the control volume is equal to the difference between the net heat transfer to the control volume and the net power produced by the control volume; plus the difference in energy flowing into and out of the control volume [98].

\[
\frac{dE_{cv}}{dt} = Q_{cv} - W_{cv} + \dot{m}_{in} \left( i_{in} + \frac{u_{in}^2}{2} + gz_{in} \right) - \dot{m}_{out} \left( i_{out} + \frac{u_{out}^2}{2} + gz_{out} \right)
\]  

(2.9)

Energy is transferred in the exchanger from a hot refrigerant stream to a cool air stream flowing through the device. The energy transferred between the two fluids in the exchanger is
evaluated considering the reduction of the energy equation for each fluid stream. Under steady state conditions the total energy transferred from the refrigerant is equal to the product of the mass flow rate and the change in enthalpy of the refrigerant per Eq (2.10). This form of the equation assumes the flow is steady, no work is done, the kinetic energy is negligible, and potential energy change is negligible since the flows are horizontal.

\[ Q_{ref} = \dot{m}_{ref}(i_{out} - i_{in}) \quad (2.10) \]

The change in energy between any two points in the refrigerant flow can be calculated using Eq. (2.11) regardless of whether there is phase change or not. In the case of single-phase liquid flow the refrigerant is treated as an ideal liquid and the energy equation is further reduced to Eq. (2.12) whereby the enthalpy is equal to the product of the specific heat of the fluid and the change in temperature.

\[ Q_{ref,SP} = \dot{m}_{ref} c_{p,ref} (T_{out} - T_{in}) \quad (2.11) \]

In the case of induced phase change in the refrigerant, the change in enthalpy of the fluid is given by Eq. (2.12) whereby the heat rate depends on the mass flow rate, the latent heat of vaporization, and the change in quality. Evaluation of this form of the equation requires a full energy balance to be carried out on the external oil circulator and ethylene glycol circulator determine how much energy was rejected to the air-side of the heat exchanger. This full energy balance was not carried out in this manner in favor of an energy balance on conducted on the air-side.

\[ Q_{ref,TP} = \dot{m}_{ref} i_{fg}(x_{in} - x_{out}) \quad (2.12) \]

The energy leaving the refrigerant-side is transferred to the air-side in the exchanger. All of the operating conditions explored in the investigation are well below critical pressures and well above critical temperatures, therefore the ideal gas law remains valid. The compressibility factor was evaluated for all experimental conditions explored in the investigation to ensure the validity of this assumption. Changes in the kinetic energy of the air flow cannot be neglected for the high flow rates investigated. Both the fluid enthalpy and kinetic energy must be evaluated at a given location in the flow to determine the total energy in the fluid. Evaluation of the kinetic energy requires calculation of the fluid velocity which can be determined by substituting the ideal gas law, Eq. (2.13), into the continuity equation, Eq. (2.14), to yield Eq. (2.15). The speed of sound for a perfect gas is a function of the gas constant, ratio of specific heat, and the measured temperature.
The air-side Mach number at a point is the ratio of the fluid velocity to the speed of sound per Eq. (2.16).

\[ \rho_i = \frac{p_i}{RT_i} \]  
\[ \dot{m}_i = \rho_i u_i A_{c,i} \]  
\[ u_i = \frac{\dot{m}_i}{\rho_i A_{c,i}} \]  
\[ M_i = \frac{u_i}{\sqrt{\gamma RT_i}} \]  

The energy equation, Eq. (2.9), is reduced for the air-side by assuming a steady, compressible flow, with negligible potential energy change, and negligible external volume forces Eq. (2.17). For an ideal gas, the specific heat is only a function of temperature, therefore the change in specific enthalpy can be simplified using Eq. (2.18) to obtain Eq. (2.19).

\[ Q_{\text{air}} = \dot{m}_{\text{air}} \left[ (i_{\text{out}} - i_{\text{in}}) + \left( \frac{1}{2} u_{\text{out}}^2 - \frac{1}{2} u_{\text{in}}^2 \right) \right] \]  
\[ c_p(T) = \frac{di}{dT} \]  
\[ Q_{\text{air}} = \dot{m}_{\text{air}} \left[ c_{p,\text{air}} (T_{\text{out}} - T_{\text{in}}) + \left( \frac{1}{2} u_{\text{out}}^2 - \frac{1}{2} u_{\text{in}}^2 \right) \right] \]  

The same form of the air-side equation may alternatively be obtained from the definition of stagnation temperature, given in Eq. (2.20), to yield the result presented in Eq. (2.21). The energy balance is not directly measuring the change in energy along the microchannel with a surface temperature measurement it is not directly affected by viscous changes to the fluid at the microchannel wall. This form of the energy equation remains valid for use in evaluation of the air-side energy performance based on thermodynamic evaluation of the fluid upstream and downstream of the test section as long as the gas may be considered ideal.

\[ T_{0,i} = c_{p,i} T_i + \frac{u_i^2}{2} \]  
\[ Q_{\text{air}} = c_{p,\text{air}} (T_{\text{0,\text{out}}} - T_{\text{0,\text{in}}}) \]
2.5.1 Experimental Measurements

Pressure does not change through the boundary layer in a perpendicular direction to the wall [99]. Therefore the local value of static pressure in a moving stream may be measured by sensing the pressure in the direction that is normal to the flow streamline. The air-side and refrigerant side static pressures were respectively measured with absolute and gauge pressure transducers adjacent to the fluid flow direction. The refrigerant-side pressure drop for single-phase flow was also measured with a differential pressure transducer adjacent to the fluid flow direction.

The static temperature is defined as the temperature that would be sensed by an instrument moving along at the local fluid velocity. The stagnation temperature is defined as the temperature of the fluid if it were to be reversibly and adiabatically brought to rest at a point. Fundamentally, in liquids the stagnation and static temperatures are equal [100]. In gas flows it is convenient to think of the kinetic energy of a high velocity fluid being converted into thermal energy by reversibly and adiabatically bringing the flow to rest at a point. This is true at normal pressures and temperatures such that the velocity of the gas at the surface interface is zero because of the effects of viscosity. Practically however, in high speed compressible flows, the kinetic energy of the fluid is not completely converted into thermal energy because some of the energy ends up being dissipated due to the viscosity of the fluid [100]. The thermocouple measurement will therefore read a temperature somewhere between the true static and stagnation temperatures. It is common practice to simply increase the uncertainty of the thermocouple reading in order to account for this effect at the expense of a larger experimental error in the velocity calculation. However, this error can be reduced by applying a recovery factor to correct the temperature measured by thermocouple junction to the stagnation temperature of the moving fluid for high speed gas flows [101].

When air passes over the probe, a boundary layer is formed over its surface creating velocity and temperature gradient. The velocity gradient gives rise to shear stress resulting in fluid friction and heat dissipation. This will make the probe feel a temperature above the stagnation temperature. The temperature gradient in the boundary layer also gives rise to a loss of heat from the probe since the probe is at a higher temperature than the flow. These two competing effects are considered by evaluation of the Prandtl number. The fluid Prandtl number, given by Eq. (2.22) represents the ratio of momentum to thermal diffusivity. If the Prandtl number of the fluid is unity these two effects will exactly cancel each other out. For gases, $Pr < 1$, for air $Pr \sim 0.72$ up to
moderately high temperatures. Hence the heat conduction from the probes surface to the air is
dominant over the frictional temperature rise and the temperature recorded by the probe is less
than the stagnation temperature [102].

\[ Pr = \frac{c_p \mu}{k} \quad (2.22) \]

The relationship between temperature and velocity for thermocouples with a known
recovery factor is given by Eq. (2.23) where the probe temperature is equal to the adiabatic wall
temperature, \( T_{aw,i} \), representing the real equilibrium temperature of the stationary probe with respect
to the fluid flow. The adiabatic wall temperature for a compressible gas is determined by the
relation given in Eq. (2.23) where \( u^2/2c_p \), is the dynamic temperature and \( r \) is the recovery factor.

\[ T_{aw,i} = T_i + r \frac{u_i^2}{2c_p} \quad (2.23) \]

The recovery factor can be determined from Eq. (2.24) gives the fraction of the kinetic
energy recovered as thermal energy.

\[ r = 1 - \frac{T_{0,i} - T_{aw,i}}{u_i^2/2c_p} \quad (2.24) \]

The stagnation temperature is related to the adiabatic wall temperature by Eq. (2.25) [100].

\[ T_{0,i} = T_{aw,i} + \frac{(1 - r)u_i^2}{2c_p} \quad (2.25) \]

The recovery factor is not a constant value but depends on the character of the flow on the
surface, the flow regime, and the thermal properties of the medium. The recovery factor for
compressible flow on a flat wall is \( r = Pr^{1/2} \approx 0.85 \) for a laminar boundary layer and \( r = Pr^{1/3} \approx 0.9 \)
for a turbulent boundary layer.

Ferneilus [101] reported a value of \( r = 0.815 \) for 1.59 T-type sheathed thermocouples for
Ma > 0.4. Moffat et al. [103] reported recovery factors for beaded thermocouple junctions of round
wire for wires normal to the flow \( r = 0.68 \pm 0.07 \) for wires parallel to the flow \( r = 0.86 \pm 0.09 \).
Total temperature probes are specifically designed to bring a fluid adiabatically to rest either with
a shield or by placing a shroud downstream of a thermocouple exist. These probes can become
quite complicated and are both larger and more expensive than un-shrouded thermocouples [100].

While the adiabatic wall temperature treatment is convenient a real thermocouple probe
does not always stagnate a moving gas effectively. Although there will be stagnation at a point,
most temperature sensing probes indicate a mean temperature, thus the effect of the entire probe
gallery must be considered [104]. It is not necessarily any more realistic to consider an idealized
probe than it is to consider a fluid in which the effects of viscosity and thermal conductivity are
neglected due to the uncertainty in applying a constant recovery factor. A dynamic correction
factor, \( K \), can be defined to correct for the effects of a real gas flowing over a diabatic probe that
attempts to stagnate the flow by Eq. (2.26) where \( T_p \) is the equilibrium temperature sensed by the
probe. The temperature indicated by the real probe may expressed in terms of the deviation from
the ideal stagnation temperature given by Eq. (2.27) [105].

\[
T_{p,i} = T_i + K \frac{u_i^2}{2c_p} \tag{2.26}
\]

\[
T_{0,i} - T_{p,i} = (1 - K_i) \frac{u_i^2}{2c_p} \tag{2.27}
\]

The dynamic recovery factor is affected by variations in the surrounding temperature and
the Mach and Reynolds numbers. At low velocities the probe temperature will be influenced more
by conduction along the stem and radiation than by convective heat transfer. Figure 2.11 (left)
qualitatively describes the effect of variation between the temperatures for high velocity
compressible flow [102]. Figure 2.11 (right) shows that the convective heat transfer increases with
increasing velocity driving the dynamic correction factor toward the overall recovery factor [105].
Benedict [106] measured dynamic correction factors in air flow for various probe type geometries
including half-shielded probe, a pencil type probe, and a bare wire probe. It was concluded that
the half-shielded thermocouples were the least sensitive to radiation and Mach and Reynolds
numbers effects and had a constant \( K \approx 0.96 \). The bare wire type was less sensitive to radiation
and Mach and Reynolds number effects and a dynamic correction factor ranging from \( 0.42 < K < 0.82 \).
The pencil type thermocouple exhibited the most sensitivity to radiation and Mach and Reynolds
number effects, and had a \( K \) value ranging from \( 0 < K < 0.82 \). Both the bare wire type
and pencil type devices converge to the result obtained by Ferneilus for subsonic \( Ma > 0.4 \).
2.5.2 Experimental Uncertainty

The experimental errors present in the testing platform include the inherent errors in the measurement devices, errors in the instrumentation calibration, and systematic effects which change the magnitude of the quantity being measured. Systematic errors from the temperature rise in the system which are not reflected in the enthalpy rise of the air-flow. Natural convection from the top and bottom surfaces of the chip are negligible due to thermal insulation of the heat exchangers. The natural convection coefficient is approximately three orders of magnitude smaller than the forced convection of the exchanger fluids. Conduction heat transfer from the manifold is minimized via the Teflon and Garolite gaskets which insulate the copper from the steel cage. The maximum operating temperatures explored in this investigation are well below those in which radiation heat transfer gains significance. Radiation heat transfer from a 1 cm² copper surface at
80°C to the ambient are less than 0.05 W. The experimental uncertainties for the measurement devices are summarized in Table 2.3.

The geometry of the manufactured copper heat exchangers are presented in Table 2.2. The channel diameters and pitch offsets were measured by viewing the microchannel heat sinks under a high-power microscope equipped with a workstation measurement tool. Channels obstructed by the manifold interface which were known to not participate in heat transfer were not included in the performance analysis. The channel diameter along the length of the tube was not characterized and therefore assumed to be uniform.

Heat exchanger convection coefficients were extracted via the modified Wilson-plot method which determines individual heat transfer resistances from an overall resistance by regression and extrapolation. Unfortunately, uncertainties in the Wilson-plot method are not as straightforward as in the single sample method. In the single sample method the refrigerant-side convection coefficient is accounted for by a correlation and the conduction resistance is accounted for by a simple analytical expression. In this configuration, the air-side convection coefficient is determined from a simple iteration between the fin efficiency and air-side convection coefficient. By contrast, for the Wilson-plot method, measurement uncertainties of the refrigerant-side and air-side temperatures and flow rates all have an impact the slope and ordinate intercept of found from regression of multiple data sets. A single slope and ordinate intercept result from the regression of a Wilson curve, it is unclear what impact these factors have on the uncertainty of the resulting convection coefficients. El Sherbini et al. [107] recommended using a Chi-squared statistical goodness of fit strategy for minimizing uncertainty of Wilson plot results. It was also recommended to more points closer to the x-axis and greater relative spread improve the error in the ordinate intercept. This method was applied to minimize the uncertainty in calculation of heat transfer coefficients.

2.6 Conclusions

This chapter provides an overview of the modeling techniques employed to characterize the design space. The design and manufacturing for three μEDM manufactured copper compact heat exchangers are presented. Experimental design for air-side and refrigerant-side test facilities utilized to characterize the compact heat exchangers is described. Experimental procedures for single-phase and two-phase refrigerant heating over a range of operating conditions are detailed.
The experimental methodology including measurement strategy and uncertainty analysis is discussed.

Smaller channel diameters can accommodate the integration of more channels over a given area at the expense of increase manufacturing cost and are expected to have higher heat transfer coefficients. Smaller channel sizes are projected to increase performance and cost of operation by due to higher hydraulic pressure drops. The results of this analysis allow heat transfer and pressure drop capacities to be considered against manufacturing capabilities in order to select optimal parameters for the heat exchanger performance.
CHAPTER 3: SINGLE PHASE PERFORMANCE

3.1 Overview

Fluid flow and heat transfer in internal macroscale channels have been extensively analyzed and its theory and results are considered well-established with accepted correlations. Flow passage dimensions in convective heat transfer applications have been diametrically shifting towards smaller dimensions to meet the growing demand for high heat transfer performance in power electronics. This shift toward smaller channel sizes is accompanied by the inherent tradeoff of higher pressure drops per unit length. In dealing with flows in minichannels and microchannels there are not any expected fundamental changes from the continuum approximation employed in microfluidic applications [108]. Although, as mentioned in the previous section, scaling effects may be relevant in certain conditions and should be checked.

Air-side pressure drops were measured and the data are reduced to account for inlet, momentum, and acceleration effects in order to produce experimentally determined friction factor data for compressible air-flow in circular microchannels with hydraulic diameters of 335 and 520 μm. The experimentally obtained data agree with the predictions of the adiabatic compressible Fanno flow model within 15%. The measured friction factor for both devices are shown to decrease with increasing Reynolds numbers. The measured friction factors for the device with the smaller hydraulic diameter of 335 μm are less than those measured in the 520 μm channels.

Single-phase liquid refrigerant-side pressure drop for developing flow in eight parallel copper channels having a hydraulic diameter of 0.8 mm and an aspect ratio of 0.25 was obtained. Flow development are shown to be relevant for the short channel length. A measured apparent friction factor for the configuration is reported and shown to be in agreement with the predictions of Phillips [109] for developing laminar flow in microchannels having an aspect ratio of 0.25. Developing flow effects are relevant in determination of the pressure drop characteristics in these devices and should be thusly incorporated into design of system when operating with single-phase liquid only fluid flow in short rectangular microchannels.

The heat transfer performance of the heat exchangers is evaluated by application of the first law of thermodynamics to the free stream of the fluid flow in order to calculate the total energy between the upstream and downstream positions. The next step in the thermal characterization is to obtain average convection heat transfer coefficients for air-side and refrigerant-side flow in the
device. The obtained experimental results are used to characterize deviations from the predictions of conventional theory.

3.2 Air-Side Pressure Drop

Flow of compressible gases in straight duct is based on the momentum theorem and is calculated from Eq. (3.1) which can be integrated to give the pressure drop along the length of the channel given by Eq. (3.2) [52].

\[
\frac{dP}{dx} = -f \left( \frac{L}{2D_h} \frac{G^2}{\rho} + G \frac{du}{dx} \right) \tag{3.1}
\]

\[
P_1 - P_2 = \frac{G^2}{2} \left[ f \left( \frac{L_{12}}{D_h \rho_m} + 2 \left( \frac{1}{\rho_2} - \frac{1}{\rho_1} \right) \right) \right] \tag{3.2}
\]

The flow in each microchannel of the heat exchanger is considered as one-dimensional flow in a duct of constant cross-sectional area. Both the effects of heat exchange and shear stress at the wall resulting from the changes in viscosity are important when considering momentum effects of single-phase air flow in the circular microchannels.

The pressure drop includes additional losses at the channel entrance and exit under experimental conditions where the pressure is experimentally measured in the inlet and outlet plenums. The pressure drop of the air-side test section was decomposed into four components: entrance effect, acceleration effect, core friction, and exit effect. The measured pressure drop \( \Delta P \) is a function of the Reynolds number, \( Re \), mass flux, \( G \), density, \( \rho \), cross-sectional area, \( A_c \), total transfer area, \( A \), and geometry-dependent coefficients \( K_c \), \( K_e \), and \( \sigma \) [78]. The air-side mass flux was determined by dividing the measured air mass flow rate by the total cross sectional area of the microchannels using Eq. (3.3)

\[
G_{air} = \frac{m_{air}}{A_c} \tag{3.3}
\]

Contraction and expansion pressure loss was accounted for with coefficients for flow between inlet and outlet manifolds and the microchannels. The pressure drop coefficients \( K_c \) and \( K_e \) were determined graphically from the ratio of the cross sectional area to the frontal area, \( \sigma \), given in Eq. (3.4) in conjunction with Figure 3.1 from Kays and London [78]. Abdelall et al. [110] show that the loss coefficients associated with single-phase flow in microchannels are comparable to those for macrochannels with the same area ratios. The density of the air is determined from
pressure and temperature measurements via the ideal gas law, Eq. (2.13), at points A and D. The air pressures at points B and C are determined with pressure estimates considering isentropic expansion between points A and B, and C and D, respectively.

$$\sigma = \frac{A_c}{A_{fr}}$$  \hspace{1cm} (3.4)

Figure 3.1 Contraction and Expansion loss coefficients for flow between inlet and outlet manifolds and the microchannels (A) $\alpha_c < 1$ and (B) $0.1 \leq \alpha_c \leq 1.0$. [Image adapted from [78]]

<table>
<thead>
<tr>
<th></th>
<th>Device A</th>
<th>Device B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_h$</td>
<td>520 $\mu$m</td>
<td>355 $\mu$m</td>
</tr>
<tr>
<td>$N_T$</td>
<td>150</td>
<td>300</td>
</tr>
<tr>
<td>$A_c$</td>
<td>18.04 cm$^2$</td>
<td>24.07 cm$^2$</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>0.31</td>
<td>0.29</td>
</tr>
<tr>
<td>$K_c$</td>
<td>0.3</td>
<td>0.3</td>
</tr>
<tr>
<td>$K_e$</td>
<td>0.49</td>
<td>0.51</td>
</tr>
</tbody>
</table>

Table 3.1 Air-side hydraulic characteristics and loss coefficients.

The frictional pressure drop is calculated by subtracting the entrance, exit, and momentum components from the experimentally measured pressure drop, Eq. (3.5).
\[ \Delta P_{\text{meas}} = \frac{G_{\text{air}}^2}{2 \rho_B} \left[ \frac{(K_c + 1 - \sigma^2)}{\Delta P_{\text{ent}}} + 2 \frac{(\rho_B}{\rho_C} - 1) \Delta P_{\text{mom}} \right] + f \left( \frac{A}{A_c \rho_m} - \frac{(1 - \sigma^2 - K_c)}{\Delta P_{\text{exit}}} \frac{\rho_B}{\rho_C} \right) \tag{3.5} \]

Figure 3.2 shows the measured pressure drop, including the decomposition into the previously mentioned components, for Device A and Device B. The pressure drop is dominated by core friction as expected. The negative pressure drop at the exit represents expansion of the air at the channel outlet. The key parameter that was determined from Eq. (3.5) is the experimentally determined friction factor, \( f \), which was resolved from Eq. (3.6) and Eq. (3.7).

\[ \Delta P_{\text{core}} = \Delta P_{\text{meas}} - \Delta P_{\text{ent}} - \Delta P_{\text{mom}} - \Delta P_{\text{exit}} \tag{3.6} \]

\[ f = \frac{\Delta P_{\text{core}}}{\rho_m A_c} \frac{A_c}{G_{\text{air}}^2 \frac{A}{A}} \tag{3.7} \]

The core pressure drop also contains the contribution due to developing flow in the entrance region. Shah and London [53] recommend Eq. (3.8) to assess the pressure drop associated with developing flow.

\[ \Delta P_{\text{dev}} = K_\infty \frac{G_{\text{air}}^2}{2 \rho_m} \tag{3.8} \]

The coefficient \( K_\infty \) is the asymptotical value of the incremental pressure drop in circular ducts and can determined from the following expression in Eq. (3.9) [111].

\[ K_\infty = 1.2 + \frac{38}{Re} \tag{3.9} \]

Evaluation of Eq. (3.8) and (3.9) show that 15-17% of the pressure drop in the core was due to the effect of flow development.
Figure 3.2 A) Air-side pressure drop measurements for Device A and B as a function of Re. B) The measured apparent friction factors are compared to the isothermal compressible Fanno model as a function of Re.
An approximation of the friction factor was determined from the Fanno models neglecting heat transfer. The experimentally determined friction factors were compared to the isothermal compressible flow model given by Eq. (3.10) and the adiabatic compressible flow Fanno model given by Eq. (3.11) [68, 112, 113].

\[ \frac{f(l^*)}{D_h} = \left( \frac{1 - \gamma M^2}{\gamma M^2} + \ln(\gamma M^2) \right) \]  \hspace{1cm} (3.10)

\[ \frac{4f(l^*)}{D_h} = \frac{1 - M^2}{\gamma M^2} + \frac{\gamma + 1}{2\gamma} \ln \left[ \frac{[(\gamma + 1)/2]M^2}{1 + [(\gamma - 1)/2]M^2} \right] \]  \hspace{1cm} (3.11)

In the above equations, \( l^* \) is the choking length. The channel length, \( l \), may not be long enough to choke, the supporting relation given in Eq. (3.12) is utilized when the flow is not choked at the exit.

\[ \frac{4f(l_1 - l_2)}{D_h} = \frac{4f(l^* - l_2)}{D_h} - \frac{4f(l^* - l_1)}{D_h} \]  \hspace{1cm} (3.12)

The isothermal compressible flow model can be simplified using the ideal gas law as given by Eq. (3.13) [113]. As previously discussed, \( P_1 \) and \( P_2 \) were adjusted from the measurement for to eliminate entrance, exit, and momentum effects.

\[ \frac{4f(l^* - l)}{D_h} = \left( \frac{P_1^2}{RT_2 \gamma} - 2 \ln \left( \frac{P_1}{P_2} \right) \right) \]  \hspace{1cm} (3.13)

The adiabatic compressible flow model can be further simplified using the ideal gas law as given by Eq. (3.14) [114].

\[ \frac{4f(l_1 - l_2)}{D_h} = \frac{P_2^2}{RT_1 G^2} \left( 1 - \frac{P_2^2}{P_1^2} \right) \]

\[ + \frac{\gamma + 1}{2\gamma} \ln \left[ \frac{T_1 2P_2^2 \gamma + (\gamma - 1)G^2RT_2}{T_2 2P_1^2 \gamma + (\gamma - 1)G^2RT_1} \right] \]  \hspace{1cm} (3.14)

Friction factors were obtained for the case of no heating in the devices. Equations (3.13) and (3.14) were shown to agree with one another within ±5%. The measured friction factor was in agreement with the isothermal compressible friction factor calculation ±5% and the adiabatic Fanno friction factor calculation ±11%.

The friction factor can be used to calculate the fluid flow performance of other gases through channels of this size and shape. The friction factors for Device A and Device B agree with the adiabatic Fanno model to within 15% and with the isothermal compressible flow model to
within 1%. The measured friction factors for both devices are shown to decrease with increasing Reynolds numbers. The measured friction factors for Device B having smaller hydraulic diameter are less than those measured in Device A. This result is in agreement with the expectation that friction factors are to decrease with decreasing channel size [115]. Friction factors are expected to decrease further for channels with decreasing channel depths [53]. One method for improving the measurement of friction factors in microchannels would be to obtain pressure measurements inside the channel itself; this method eliminates the need to assume loss coefficients, however it is not well suited for practical applications.

The air-side pumping power is a critical parameter for evaluation of the devices as it describes the energy required for the system to achieve its performance. The air-side pumping power is the product of the volumetric flow rate, Eq. (3.15), and the associated overall pressure drop. The air-side volumetric flow rate is calculated by dividing the measured mass flow rate by the average density of the fluid along the channel. The total pumping power for the system is given by Eq. (3.16) which also includes refrigerant-side pumping power. The refrigerant-side pumping power is negligible in comparison to air-side pumping power in the current configuration.

\[ \dot{V}_{\text{air}} = \frac{m_{\text{air}}}{\rho_{\text{m,air}}} \]  
\[ \dot{W}_{\text{tot}} = \Delta P_{\text{me,air}} \dot{V}_{\text{air}} + \Delta P_{\text{me,ref}} \dot{V}_{\text{ref}} \]

Kohl [27] investigated friction factors for gases in microchannels assuming a constant cross-sectional area and a constant friction factor along the channel comparing both adiabatic and isothermal flow with friction, Eq. (3.13) and (3.14) and found that the numerical models for each case agreed within 1%, well within the uncertainty of the experiment. In order to accurately predict friction factor for compressible flows the analytical results from Schwartz [116] were recommended as an initial approximation. A numerical model is required to provide a more detailed investigation of the effects of compressibility and heat addition on friction factor in the heat exchangers [68].

3.3 Liquid Refrigerant-Side Pressure Drop

One-dimensional flow of an incompressible fluid in a smooth circular pipe based on the continuum assumption for Newtonian liquid flows in minichannels and microchannels forms the basis for pressure drop analysis in internal flows. The pressure gradient and the wall shear stress
given by Eq. (3.17) are related by the Fanning friction factor, $f$, in Eq. (3.18). The friction factor is widely reported in the literature due to its ability to consistently represent the momentum transfer process of fluid flow in a similar manner with the heat and mass transfer process requirements.

$$
\tau_W = \mu \frac{du}{dy} \bigg|_{W} \quad (3.17)
$$

$$
f = \frac{\tau_W}{(1/2)\rho u_m^2} \quad (3.18)
$$

The friction factor is dependent on the flow regime, wall geometry, flow development, and surface finish of the walls. The frictional pressure drop, $\Delta P$ over the channel length, $L$, is obtained by Eq. (3.19).

$$
\Delta P = \frac{2f \rho u_m^2 L}{D_h} \quad (3.19)
$$

For non-circular flows the diameter is replaced by the hydraulic diameter calculated by Eq. (3.20) where $A_c$ is the channel cross sectional area and $P_W$ is the wetted perimeter. The hydraulic diameter of a rectangular channel is computed by Eq. (3.21) where the channel width, $a$, and the channel height, $b$, are defined in Figure 2.7B.

$$
D_h = \frac{4A_c}{P_W} \quad (3.20)
$$

$$
D_h = \frac{2ab}{a + b} \quad (3.21)
$$

In the design of microchannel heat sinks it is necessary to predict the overall pressure drop. The friction factor for developed laminar flow in a rectangular cross section is a function of the channels cross sectional aspect ratio. The cross sectional channel aspect ratio is defined as the ratio of the short side to the long side $\alpha_c = a/b$. The Pousille number is related to the friction factor, $f$ and the Re by Eq. (3.22). Shah and London [53] provide a relation for fully developed Poiseuille number in rectangular microchannels as a function of $\alpha_c$ given in Eq. (3.23). For the aspect ratio of 1:4 the equation predicts $Po=18.23$.

$$
Po = f Re \quad (3.22)
$$

$$
Po = fRe = 24(1 - 1.3553\alpha_c + 1.9467\alpha_c^2 - 1.701\alpha_c^3 + 0.9564\alpha_c^4 - 0.2537\alpha_c^5) \quad (3.23)
$$
Simultaneously developing laminar flow is a common occurrence in microchannels due to the short entrance lengths employed. Under this circumstance the fanning friction factor is replaced by an apparent friction factor $f_{app}$ to account for the effect of the developing region. In a rectangular duct with 1:4 aspect ratio the $f_{app}Re = 42.85$ is obtained from Phillips’ [109] model for laminar flow in the entrance region of a rectangular duct.

The pressure drop is decomposed into the four components described in Eq. (3.24). The flow experiences a sudden contraction at the inlet which causes the flow to separate and undergo an irreversible free expansion. In the core the refrigerant experiences skin friction and a density change due to cooling. At the exit there is another irreversible free expansion. The entrance loss coefficient, $K_c$, and the exit loss coefficient, $K_e$, obtained from Figure 3.1 based the ratio of the core free flow to the frontal cross sectional areas, $\sigma$. Table 3.2 gives the refrigerant-side characteristics, area ratio, and loss coefficients. The loss coefficients assume a fully developed uniform velocity profiles at the inlet. However, a less than fully developed velocity profile at the core exit result in smaller $K_c$ and larger $K_e$ than those obtained for the fully developed flow situation [109].

$$\Delta P = \frac{\rho u_m}{2} \left[ K_c + K_e + \frac{4f_{app}L}{D_h} \right]$$ (3.24)

<table>
<thead>
<tr>
<th>Refrigerant-side</th>
<th>Device A</th>
<th>Device B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_h$</td>
<td>800 μm</td>
<td>800 μm</td>
</tr>
<tr>
<td>$N_T$</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>$A$</td>
<td>6 cm$^2$</td>
<td>6 cm$^2$</td>
</tr>
<tr>
<td>$A_c$</td>
<td>8 mm$^2$</td>
<td>8 mm$^2$</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>0.09</td>
<td>0.09</td>
</tr>
<tr>
<td>$K_c$</td>
<td>0.8</td>
<td>0.8</td>
</tr>
<tr>
<td>$K_e$</td>
<td>0.97</td>
<td>0.97</td>
</tr>
</tbody>
</table>

Table 3.2 Refrigerant-side hydraulic characteristics and loss coefficients.

Figure 3.3 shows the single-phase liquid only refrigerant-side pressure drop measurement decomposition and the determined apparent friction factor. The pressure drop is broken down into its constitutive terms including entrance, frictional, momentum, and exit effects. The apparent friction factor is determined from the frictional pressure term and compared with Phillips’ four sided heating flow model for developing laminar flow in rectangular channels with an aspect ratio of 0.25 [109].
Phillips provided tabular data from his model for apparent friction factor as a function of the non-dimensional length, $x^+$, which is given in Eq. (3.25). Phillips predicted fully developed flow to be attained at different $x^+$ values, with low aspect ratio ducts reaching development earlier. Phillips’ laminar friction factor for a 0.25 aspect ratio channel was obtained by interpolation of the 0.2 and 0.5 aspect ratio data to get Table 3.3 which can be numerically integrated to determine the average theoretical friction factor in the channel.

$$x^+ = \frac{x}{D_n} \frac{1}{Re}$$

\[(3.25)\]

<table>
<thead>
<tr>
<th>$x^+$</th>
<th>$f_{app Re}$</th>
<th>$x^+$</th>
<th>$f_{app Re}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>142</td>
<td>0.04</td>
<td>24.48</td>
</tr>
<tr>
<td>0.001</td>
<td>111</td>
<td>0.05</td>
<td>23.23</td>
</tr>
<tr>
<td>0.003</td>
<td>66.08</td>
<td>0.06</td>
<td>22.38</td>
</tr>
<tr>
<td>0.005</td>
<td>52.33</td>
<td>0.07</td>
<td>21.83</td>
</tr>
<tr>
<td>0.007</td>
<td>45.13</td>
<td>0.08</td>
<td>21.4</td>
</tr>
<tr>
<td>0.009</td>
<td>40.45</td>
<td>0.09</td>
<td>21.05</td>
</tr>
<tr>
<td>0.01</td>
<td>38.73</td>
<td>0.1</td>
<td>20.75</td>
</tr>
<tr>
<td>0.015</td>
<td>33.13</td>
<td>0.2</td>
<td>19.33</td>
</tr>
<tr>
<td>0.02</td>
<td>29.93</td>
<td>&gt;1</td>
<td>18.2</td>
</tr>
<tr>
<td>0.03</td>
<td>26.35</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.3 Laminar flow friction factor in the entrance region of a rectangular duct with an aspect ratio of 1:4 interpolated table from Phillips model [109].

The experimentally obtained apparent friction factors agree with Phillips model to within the range of experimental uncertainty. A distinct transition to turbulence was not observed for investigated Reynolds numbers. Comparison of the measured friction factor to turbulent flow friction factor data were not able to predict the measured data. The flow is considered to satisfy the hydrodynamic conditions for developing laminar flow in a rectangular microchannel for aspect ratio of 0.25.
Figure 3.3 A) Single-phase refrigerant-side pressure drop decomposition for Device A and B as a function of mass flux at 60 and 80 °C. B) Experimentally determined refrigerant-side friction factors as a function of refrigerant-side Reynolds number.
3.4 Heat Exchanger Performance

Energy is transferred in the exchanger from a hot refrigerant to cold air. The energy transferred between the two fluids in the heat exchanger is evaluated considering the reduction of the energy equation for each fluid stream. The change in energy between any two points in single-phase refrigerant flow is calculated using Eq. (3.26) assuming the flow is steady, no work is done, kinetic energy, and potential energy are negligible.

$$Q_{ref,SP} = \dot{m}_{ref} c_{p,ref} (T_{out} - T_{in})$$  \hspace{1cm} (3.26)

The change in energy between any two points in the air flow can be calculated by Eq. (3.27) assuming a steady, compressible flow, with negligible potential energy change, and negligible external volume forces [34].

$$Q_{air} = c_{p,air} (T_{0,out} - T_{0,in})$$  \hspace{1cm} (3.27)

An energy balance on the heat exchanger is satisfied for a control volume including the cross flow heat exchanger assuming the exchanger is well insulated, there is no heat loss to the surroundings by the equality given in Eq.(3.28).

$$Q_{air} = Q_{ref}$$  \hspace{1cm} (3.28)

Figure 3.4A-B shows the measured performance for the 1 cm$^3$ heat exchanger devices for single-phase refrigerant flow. Device A is characterized with inlet temperatures of 60 and 80 °C and Device B is characterized for refrigerant inlet temperatures of 80 °C. Power density performance for heat exchanger device and B are shown as a function of refrigerant-side mass flux. The heat exchanger power density performance slightly increased as a function of the air-side Reynolds number. For Device A heat rates of 25 – 42 W and 39 – 69 W were achieved with single-phase refrigerant flow at 60 °C and 80 °C, over the refrigerant-side of mass fluxes 330 – 750 kg/m$^2$-s and air-side Reynolds numbers 9000 – 20,000. For Device B heat rates of 42 – 69 W were achieved for single-phase refrigerant flow at 80 °C for refrigerant flow mass flux range 330 – 750 kg/m$^2$-s and air-side Reynolds number range 8000 – 13,500. The additional performance enhancement of increasing the refrigerant-side mass flux was more pronounced than adjustments to the air-side flow rate due signifying the dominance in refrigerant-side thermal resistance. The air-side and refrigerant-side convection coefficients were similar however the ratio of the air side transfer area to the refrigerant-side transfer area 3 and 3.33 for Device A and B respectively.
Figure 3.4C depicts the total thermal conductance, $UA$, for two heat exchangers operated with single-phase refrigerant-heating for the conditions described above. The thermal conductance was a strong function of the refrigerant-side mass flux. The thermal conductance of Device A is higher when operated at 80 °C than at 60 °C as expected by the presence of a larger temperature gradient. The thermal conductance of Device B is larger than the thermal conductance of Device A for 80 °C operating temperature over the range of mass fluxes 330 – 750 kg/m²-s.
Figure 3.4 Heat rate density performance for Device A and B as a function of A) air-side Re
and B) refrigerant-side mass flux. C) The overall thermal conductance is shown as a function
of refrigerant mass flux for air-side condition where PA=689.5 kPa.
3.4.1 Data Reduction

Heat transfer by convection is a combination of conduction and fluid motion which entails a prescribed temperature difference between a solid surface and a moving fluid. Convection problems generally require analytical solutions to a system of mass, momentum, and energy conservation equations for a body’s geometry and fluid properties, which can be solved for the resulting flow and temperature fields. Analytical solutions for simple geometries under restrictive assumptions are widely available. A simple method based on Newton’s law of cooling, Eq. (3.29), is introduced as the standard for characterizing convection heat transfer. Newton’s law of cooling established an algebraic relation between the heat flow by convection \( Q \), the surface area \( A_s \), and the temperature difference between the solid surface \( T_s \) and the fluid \( T_f \) by the introduction of a convection coefficient \( h \) [117].

\[
Q_{\text{conv}} = hA_s(T_s - T_f)
\]  

(3.29)

Convective heat transfer for compact heat exchangers may involve complex geometries and intricate flows making analytical solutions and traditional measurement techniques difficult if not impossible. The main difficulty in directly applying Newton’s law of cooling lies in the experimental measurement of surface temperature. Experimental heat exchanger data are usually obtained by measuring the fluid temperatures at the inlet and outlet of the device for a given flow and surface geometry. The surface temperature varies from point to point due to flow development, multidimensional thermal conduction, and conjugate effects, and the flow pattern could be altered by the presence of temperature sensors. The problem of measuring surface temperatures becomes more complicated and prohibitive as heat exchangers become more compact. An alternate method for determining convection coefficients in heat exchangers without making a surface measurement is therefore preferred for broad use in practical applications [118].

The Wilson plot method [119] is a popular technique used to estimate convection coefficients for a wide variety of convective heat transfer processes. Considering a heat exchanger system with two unmixed fluids, Wilson theorized that if the mass flow rate of one fluid was modified while the other is held constant, then the change in the overall resistance would be primarily attributed to the change in convective heat transfer of the changing fluid. Wilson reasoned that the thermal resistances outside of the changing fluid could therefore be considered constant. The heat flow is determined from the energy change of the fluid thereby avoiding disturbance to the characterization of fluid flow and heat transfer which would be introduced by a
surface temperature measurement. Modifications to the Wilson plot method have been proposed by numerous researchers in an effort to enhance its accuracy [118, 120-124].

The Wilson plot method relies on the separation of the overall thermal resistance into its constitutive terms. The application of this method is based on experimental measurements of the mass flow rate and temperature of the two fluids. The overall thermal resistance Eq. (3.30) is expressed as the sum of the refrigerant-side convective term, Eq. (3.31), the conductive wall resistance, Eq. (3.32), and the air-side convective term, Eq. (3.33). Substitution of the proper expressions for each type of thermal resistance leads to into the overall thermal resistance given in Eq. (3.34).

\[
R_{tot} = R_{ref} + R_{cnd} + R_{air}
\]  
(3.30)

\[
R_{ref} = \frac{1}{h_{ref}A_{ref}}
\]  
(3.31)

\[
R_{cnd} = \frac{1}{k_{WS}}
\]  
(3.32)

\[
R_{air} = \frac{1}{\eta_f h_{air}A_{air}}
\]  
(3.33)

\[
R_{tot} = \frac{1}{h_{ref}A_{ref}} + \frac{L}{k_{WS}} + \frac{1}{\eta_f h_{air}A_{air}}
\]  
(3.34)

The overall thermal resistance of the device can be obtained experimentally by measuring the inlet temperature and outlet temperature at various mass flow rates operating in the same flow regime. The overall thermal conductance, $UA$, of the heat exchanger is related to overall resistance by Eq. (3.35). The thermal conductance is the ratio of the heat flow transferred between the two fluids to the log-mean temperature difference between the refrigerant and the air-flow $\Delta T_{LM}$, given in Eq (3.37). The first law of thermodynamics is applied to both refrigerant-side flow and the air-side flow in order to calculate the total heat flow leaving the refrigerant flow and entering the air flow through the heat exchanger.

\[
R_{tot} = \frac{1}{UA}
\]  
(3.35)

\[
Q = FUA\Delta T_{LM}
\]  
(3.36)
The log mean temperature difference is calculated for a cross-flow heat exchanger using a correction factor $F$ for cross-flow heat exchanger configuration with both flows unmixed using Eq. (3.37)-(3.41) [125].

$$\Delta T_{LM,CF} = \frac{(T_{H,\text{out}} - T_{C,\text{in}}) - (T_{H,\text{in}} - T_{C,\text{out}})}{\ln \left( \frac{T_{H,\text{out}} - T_{C,\text{in}}}{T_{H,\text{in}} - T_{C,\text{out}}} \right)}$$ (3.37)

$$\Delta T_{LM} = F \Delta T_{LM,CF}$$ (3.38)

$$F = f(P, R)$$ (3.39)

$$P = \frac{T_{C,\text{out}} - T_{C,\text{in}}}{T_{H,\text{out}} - T_{C,\text{in}}}$$ (3.40)

$$R = \frac{T_{H,\text{in}} - T_{H,\text{out}}}{T_{C,\text{out}} - T_{C,\text{in}}}$$ (3.41)

### 3.4.2 Determination of Thermal Resistances

For a given convective heat transfer coefficient, the heat transfer resistance can be reduced by enlarging the transfer area. Since it is desirable to minimize the projected area of the chip module, this extended area is usually best provided by a finned structure. The fin efficiency concept can be used to deal with the conductive resistance of the fin structure. Fin efficiency is defined as the ratio of the average temperature difference between the fin and the fluid to the temperature difference between the fin base and the fluid. Short thick fins with high thermal conductivity provide the highest values of efficiency [2].

The air side of the heat exchanger was modeled as four parallel fins as depicted in Figure 2.3. Fin efficiency is calculated from the fin parameter defined in Eq. (3.42) and the fin length using Eq. (3.43).

$$\beta = \frac{h_{\text{air}} P_{f\text{in}}}{\sqrt{k_{W} A_{c,\text{fin}}}}$$ (3.42)

$$\eta_f = \frac{\tanh(\beta L_f)}{\beta L_f}$$ (3.43)

The non-dimensional Maranzana number, $Mz$, was evaluated to determine the effects of axial conduction, defined as the ratio of the conductive heat flux to the convective heat flux.
Maranzana et al. [126] stated that axial conduction can be neglected if this parameter is less than 0.01.

\[ Mz = \frac{k_w A_c}{k_f A \text{RePr}} \quad (3.44) \]

In the current configuration the Maranzana number remains well below this threshold, therefore axial condition was neglected. This was shown to be in good agreement with the conjugate modeling performed using COMSOL, therefore a 1-D fin approximation is considered appropriate.

Dittus-Boelter [127] recommend the correlation given in Eq. (3.45) for heating in a fully developed turbulent flow in a smooth circular tube. Sieder-Tate recommend [122] a turbulent flow correlation for flows characterized by large property variations where all properties except for viscosity are evaluated at the mean fluid temperature given in Eq. (3.46). Sieder-Tate also recommend a correlation for laminar flow including the effect of entry length given by Eq. (3.47).

\[ Nu_{\text{Dittus-Boelter}} = 0.023 \text{Re}_D^{0.8} \text{Pr}^{0.4} \quad (3.45) \]

\[ Nu_{\text{Sieder-Tate, turb}} = 0.027 \text{Re}_D^{0.8} \text{Pr}^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \quad (3.46) \]

\[ Nu_{\text{Sieder-Tate, lam}} = 1.86 \left( \frac{\text{Re}_D \text{Pr}}{L/D} \right)^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \quad (3.47) \]

The air-side convection coefficient was determined using the modified-Wilson plot method previously described. The liquid only refrigerant-side mass flow rate was adjusted while holding the air-side mass flow rate constant and the heat loads for both fluid streams were measured. In the modified-Wilson plot regression, the air-side Nusselt number was expected to assume the form of the turbulent flow Sieder-Tate correlation for circular tubes given by Eq. (3.48). The refrigerant-side was expected to assume the laminar form of the Sieder-Tate equation given by Eq. (3.49). These forms of the Nusselt number are plugged into Eq. (3.50) to calculate the convection coefficient and then substituted into the total thermal resistance network introduced by Eq. (3.34).

\[ Nu_{\text{air}} = C_A \text{Re}^{0.8} \text{Pr}^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \quad (3.48) \]

\[ Nu_{\text{ref}} = C_{R1} \left( \frac{\text{Re}_D \text{Pr}}{L/D} \right)^{c_{R2}} \left( \frac{\mu}{\mu_w} \right)^{0.14} \quad (3.49) \]
\[ h_i = \frac{N u_i k_i}{c_{p,i}} \quad (3.50) \]

The most general conceptualization of the Wilson plot method involves simultaneously solving for three unknown constants; two correspond to the unknown functional coefficient and the third corresponds to an unknown exponent of a flow parameter. The unknown constants do not behave linearly and need to be found by an iterative procedure. Many procedures for carrying out the iteration for different numbers of unknown constants have been proposed, however most references focus on the modified-Wilson plot method proposed by Briggs and Young [123]. Their modification to the Wilson method allows for calculation of three unknowns, the two coefficients \( C_A, C_{RI} \), and one exponent, \( C_{R2} \). However there are situations where all of the heat transfer correlations and the wall thermal resistance are unknown. Khartabil [128] developed a method capable of determining three resistances with up to five unknown parameters. However in order for this method to be employed three data sets are required, each with a different dominant resistance. In the current analysis the conducive resistance is obtained for the geometry from in COMSOL and the assumption of scaling the Reynolds number to the power of 4/5 on the air-side is regarded a reasonable assumption for circular channels with turbulent flow.

The equation for overall resistance was rearranged into system of equations with three unknowns \( C_A, C_{RI}, \) and \( C_{R2} \), which are constrained by \( Y = mX + B \) as given by Eq. (3.51) - (3.55) An iterative regression is carried out for different values of the constants driven by maximization of the coefficient of determination, \( R^2 \) such that the residuals are minimized. The refrigerant-side constant is proportional to the inverse of the slope and the air-side constant is proportional to the inverse of the y-intercept according to the model. The results of this model were plugged into the COMSOL model to provide a new value which was determined to be \( R_{end} < 0.01 \) for all conditions explored.

\[ Y = \left( \frac{1}{UA} - R_{end} \right) \left[ k \frac{k_{\eta f}}{D_n} A \eta f Re^{0.8} Pr^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \right]_{air} \quad (3.51) \]

\[ X = \left[ k \frac{k_{\eta f}}{D_n} A \eta f Re^{0.8} Pr^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14} \right]_{air} \quad (3.52) \]
\[ m = \frac{1}{C_{R1}} \]  
\[ B = \frac{1}{C_A} \]  
\[ Y = mX + B \]

The air-side constant \( C_A \) was calculated to be approximately 0.026 – 0.028. The air-side constant scarcely varied as a function of the experimental conditions. The measured air-side heat transfer coefficients for Device A and Device B are shown to be in very good agreement with the predictions of the turbulent Sieder-Tate model as shown in Figure 3.5A. The heat transfer coefficients are slightly under-predicted by the Dittus-Boelter equation which fails to account for change in fluid properties of the flow. This result is in agreement with the recent findings of C. Yang et al. [129] who concluded that conventional heat transfer correlations could be well applied for gas flow in circular microtubes with diameters ranging from 86 – 920 μm.

The refrigerant-side heat transfer coefficients were calculated from the modified-Wilson plot method. The refrigerant-side constants were calculated to be 1.19 –1.36 for \( C_{R1} \) and with 0.45 – 0.50 for \( C_{R2} \). On the refrigerant-side the Nusselt number predictions from these constants did not vary significantly, as a higher computation of \( C_{R2} \) was compensated by lower values of \( C_{R1} \). The resulting heat transfer coefficients significantly exceed the values that would be predicted by the laminar Sieder-Tate correlations.

The calculated refrigerant-side convection coefficients are shown in Figure 3.5B compared to Phillips [109] laminar developing flow model for flow in a rectangular channel with an aspect ratio of 0.25 and the turbulent prediction recommended by Petukov [79]. The calculated refrigerant-side convection coefficient are much higher than what is predicted by the theoretical models due to the dominance of entrance effects and the transitional Reynolds numbers incurred. There is limited data for developing flow in rectangular microchannels Phillips [109] predicts the convection coefficients using a mathematical model. The results predicted by Phillips agree with the numerical predictions suggested by Lee et al. [130] in a study on the effect of thermally developing flow on heat transfer in rectangular channels having different aspect ratios.

Phillips predicts a local Nusselt number at the inlet of the channel is equal to 30, corresponding to a convection coefficient of ~2700. Stephan et al. [131] presented correlations to
predict simultaneously developing flow in circular channels for constant wall heat flux and constant wall temperature boundary conditions.

Developing flow entrance region effects for laminar flow in the short channel lengths employed in microchannels are significant [132]. Entrance region effects become more significant at higher Reynolds numbers, in part explaining the increasing Nusselt number trend with Reynolds number [132]. Surface roughness effects which increase exchanger surface area and cause an early transition to turbulence are expected to have an effect although a transition to turbulence was not clearly indicated by the friction factor data for the refrigerant-side Reynolds number range explored, $1000 < \text{Re}_{ref} < 2900$. Kandlikar et al. [81] determined that the effect of relative roughness on pressure drop was minimal but the heat transfer in the entry region showed a distinct dependence on roughness. Kandlikar showed observed Nusselt number enhancements of 80 – 100% for relative roughness ranging between 0.161 – 0.35%. The relative roughness in the refrigerant-side channels were not directly measured however, however it is expected that surface roughness effects account for some of the discrepancy between the measured data and Phillips laminar model. It is not possible to present any equations for heat transfer enhancement with roughness due to a lack of systematic data on the devices.

For the conditions explored in this investigation, Stephan predicts Nusselt numbers ranging 74-177 for constant wall temperature boundary conditions and 17-24 for a constant heat flux boundary condition. The wide spread between these Nusselt number ranges for the constant heat flux and constant wall temperature boundary condition relations may elucidate the disparity observed between Phillips laminar model and the experimental data. It is speculated that for the higher refrigerant-side mass flow rates enhanced fluid mixing occurs. This effect may be responsible for altering the fluid boundary at the front and back refrigerant side faces, increasing the heat transfer. Under this condition it is possible that a constant wall temperature boundary condition may be more suitably applied to the face of exchanger than a constant heat flux boundary condition.

Figure 3.6 shows the thermal resistance of the heat exchanger for Device A and B as a function of refrigerant-side mass flux for the air-side case where the upstream pressure is held at $P_A = 689.5 \text{ kPa}$. The thermal resistance is decomposed into the convective air-side and refrigerant-side terms. The conduction resistance is not shown in the graph. It is clear that the liquid-side refrigerant thermal resistance dominates the total thermal resistance. These results highlight the
directive for two-phase flow heat transfer enhancements in order to further reduce the overall thermal resistance and increase performance.

Figure 3.5 A) Comparison of the measured air-side convection coefficients to the Sieder-Tate and Dittus-Boelter correlations. B) Comparison of the single-phase liquid refrigerant-side convection coefficients compared to Phillips laminar model and Pertukov turbulent flow model.
3.4.3 Thermal Conduction Heat Transfer

The heat flow from one isothermal surface to another isothermal surface can be calculated according to the simple relationship given in Eq. (3.56). The conduction resistance is inversely proportional to the thermal conductivity of the material between the two surfaces. The relationship represented by Eq. (3.57) introduces a shape coefficient, \( S \), commonly referred to as a shape factor in conductive heat flow. The shape factor only depends on the geometrical arrangement of the two isothermal surfaces between which heat is transferred by conduction.

\[
Q_{cnd} = \frac{T_1 - T_2}{R_{cnd}} \quad \text{(3.56)}
\]

\[
R_{cnd} = \frac{1}{kS} \quad \text{(3.57)}
\]

The shape factor can be determined by integration of the local heat flux over the isothermal surfaces \( A_1 \) and \( A_2 \). The heat flow is given by Eq. (3.58) in which the surface normals \( n_1 \) and \( n_2 \) are directed into the conductive medium. The following relationship enables the shape factor to be calculated from the known temperature field [133].
\[ Q_{cn} = -k \int_{A_1} \frac{\partial T}{\partial n_1} dA_1 = k \int_{A_2} \frac{\partial T}{\partial n_2} dA_2 \quad (3.58) \]

COMSOL Multiphysics was used to calculate shape factors for the heat exchanger geometries over the range of air-side and refrigerant-side boundary conditions explored. The steady state temperature profile of a three-dimensional model of each heat exchanger was obtained by application of the average temperature and heat transfer coefficients obtained from the Wilson plot method. The top and bottom of the heat exchanger are treated as adiabatic surfaces. The heat flow was calculated by a surface area integration over the refrigerant-side area. The shape factor was calculated by dividing the heat flow rate, determined from the surface integration, by the temperature difference between the surface of the refrigerant-side channel and the nearest air-side channel. The relationship given in Eq. (3.59) reduces to Eq. (3.60) by carrying out the surface integration over the transfer area.

\[ S = -\frac{1}{T_1 - T_2} \int_{A_1} \frac{\partial T}{\partial n_1} dA_1 = \frac{1}{T_1 - T_2} \int_{A_2} \frac{\partial T}{\partial n_2} dA_2 \quad (3.59) \]

\[ S = \frac{Q}{k(T_1 - T_2)} \quad (3.60) \]

The thermal conductivity of pure copper varies >1%, 396.9 – 393.4, over the temperature range 20 – 80 °C. The tellurium copper alloy has a thermal conductivity of 354.6 at 20 °C, due to a lack of readily available thermal conductivity data on this alloy, this value is assumed constant in the analysis. The temperature dependence of the thermal conductivity could be readily incorporated by implementation of Eq.(3.61). A function to include this effect is available in COMSOL Multiphysics for many common materials, however this data is unavailable for the tellurium-copper alloy used in this research.

\[ k = \frac{1}{T_1 - T_2} \int_{T_2}^{T_1} k(T) dT \quad (3.61) \]

3.5 Performance Comparison

The resistance model predictions for single-phase refrigerant heating performance are compared to the experimental data in Table 3.4. The resistance model heat rate is given using the measured log mean temperature divided by the total thermal resistance. The modeled thermal resistances were determined using the Sieder-Tate on the air-side and Pertukov for the refrigerant-
side where the Re ~ 3000. The modeled air-side friction factors were determined by the Fanno model. The refrigerant-side friction factor was determined from the Phillips model for developing laminar flow.

Table 3.5 compares to the resistance model predictions and the COMSOL simulation results to the experimental results with two-phase heating condition. The simulation results were attained by applying the experimentally obtained convection coefficients and the mean temperature to a solid model of the device geometry COMSOL. The simulation power performance was obtained by integrating steady state temperature profile over the heat exchanger surface area with the given convection coefficients.

<table>
<thead>
<tr>
<th></th>
<th>Device A</th>
<th>Device B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q</td>
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<td>68.8</td>
</tr>
<tr>
<td>$f_{air}$</td>
<td>0.090</td>
<td>0.077</td>
</tr>
<tr>
<td>$f_{ref}$</td>
<td>0.018</td>
<td>0.019</td>
</tr>
<tr>
<td>$R_{tot}$ [K/W]</td>
<td>0.71</td>
<td>0.73</td>
</tr>
<tr>
<td>$R_{air}$ [K/W]</td>
<td>0.18</td>
<td>0.18</td>
</tr>
<tr>
<td>$R_{ref}$ [K/W]</td>
<td>0.52</td>
<td>0.53</td>
</tr>
<tr>
<td>$R_{cnd}$ [K/W]</td>
<td>&lt;0.01</td>
<td>&lt;0.01</td>
</tr>
<tr>
<td>LMTD</td>
<td>-</td>
<td>50</td>
</tr>
</tbody>
</table>

Table 3.4 Table comparing modeling and experimental data for the two heat exchangers with single-phase refrigerant heating.

<table>
<thead>
<tr>
<th></th>
<th></th>
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<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Device A</td>
<td>68.8</td>
<td>70.4</td>
<td>72.3</td>
</tr>
<tr>
<td>Device B</td>
<td>68.8</td>
<td>73.0</td>
<td>69.4</td>
</tr>
</tbody>
</table>

Table 3.5 Single-phase refrigerant heating data for Device A and Device B are compared to the expected values as calculated by the modeled thermal resistances and by the application of the experimental convection coefficients to a 3-D COMSOL model.

The heat exchanger performance for single-phase refrigerant flow is evaluated in Figure 3.7 which shows the coefficient of performance, COP, and air-side pumping power as a function of the power dissipation of the heat exchangers. It is evident from this graph that the relative increase in pumping power required is far larger than the power dissipation in the device. This trend is captured by comparing the friction factor to the Colburn factor of the heat exchangers to the Chilton-Colburn relation.
Figure 3.7 A) COP and Pumping power as a function of power dissipation in the heat exchangers with single-phase heating. B) The experimentally obtained $j$ vs. $f$ curve is compared to the Chilton-Colburn relation.
3.6 Conclusions

Air-side pressure drops were measured in Device A and B. Well known method were employed to reduce the data accounting for inlet, momentum, and acceleration effects in order to produce experimentally determined friction factor data for compressible air-flow in 1 cm long circular microchannels with hydraulic diameters of 335 and 520 μm. The experimentally obtained data agrees with the isothermal compressible flow model to within 1%. The experimentally obtained friction factors agree with the predictions of the adiabatic compressible Fanno flow model within 15%. The measured friction factor for both devices are shown to decrease with increasing Reynolds numbers. The measured friction factors for the device with the smaller hydraulic diameter of 335 μm are less than those measured in the 520 μm channels.

Single-phase liquid refrigerant-side pressure drop for developing flow in eight parallel rectangular copper channels having a hydraulic diameter of 0.8 mm and an aspect ratio of 0.25 were obtained. Flow development effects are shown to be relevant for 1 cm long channel. The experimentally measured apparent friction factor was in agreement with the predictions of Phillips [109]. Developing flow effects are relevant in determination of the pressure drop characteristics in these devices and should be thusly incorporated into design of system when operating with single-phase liquid only fluid flow in short rectangular microchannels.

The heat transfer performance for two 1 cm³ novel heat exchangers were characterized by application of the first law of thermodynamics to the free stream of the fluid flow in order to calculate the total energy rate dissipated in the device. Average convection heat transfer coefficients for air-side and single phase refrigerant-side flow were obtained using the well-known modified-Wilson plot.

The experimentally obtained air-side convection coefficients were well predicted by the Sieder-Tate turbulent flow model. The experimentally obtained single-phase liquid refrigerant-side convection coefficients were under-predicted by the Phillips [109] and Lee et al. [130] models for developing flow in rectangular microchannels with an aspect ratio of 0.25. The enhancement of the experimental results were within the range of expected enhancements resulting surface roughness and boundary condition effects.

For the case of single-phase refrigerant-side heating, the performance enhancement of increasing the refrigerant-side mass flux was more pronounced than adjustments to the air-side due to the dominance in refrigerant-side thermal resistance. The air-side and refrigerant-side
convection coefficients were similar however the ratio of the air side transfer area to the refrigerant-side transfer area 3 and 3.33 for Device A and B respectively.
4.1 Overview

The heat transfer performance of the heat exchangers is evaluated by application of the first law of thermodynamics to the free stream of the fluid flow in order to calculate the total energy between the upstream and downstream positions. The next step in the thermal characterization is to obtain average convection heat transfer coefficients for air-side and refrigerant-side flow in the device. The obtained experimental results are used to characterize deviations from the predictions of conventional theory.

Two-phase refrigerant-side pressure drops were measured for the case of R245fa condensing at 80 °C in rectangular minichannels in Device A and Device B. The experimental results were compared to the conventional correlations recommended by Friedel. Given the short channel lengths and very small pressure drops incurred it was determined that these models were adequate for the purpose of designing heat exchanger systems with short lengths. It is emphasized that the additional pressure drop effects due to changes in momentum of the condensing liquid, contraction at the inlet, and expansion at the exit are of equal relevance and should be accounted for. It is recommended that these effects be accounted for in determination of two-phase condensation pressure drop when designing heat exchangers with short channel lengths.

The overall heat transfer performance for two 1 cm$^3$ heat exchangers were compared for a range of operating conditions. Comparison with the literature for heat transfer in compact heat exchangers show that these results were reasonably predicted through utilization of the conventional theory. The experimentally obtained two-phase refrigerant-side convection coefficient were within the expected range predicted by the Akers model.

4.2 Condensation Refrigerant-Side Pressure Drop

The frictional pressure drop for two-phase flow in microchannels is generally determined using the macroscale correlations recommended by Lockhart Martinelli [134] and Friedel [94]. There have been a number of correlations based on modifications to these models [86, 135]. Friedel developed a correlation based on a database of 25,000 points for adiabatic flow through channels 1 mm in diameter or larger. The correlation uses a liquid-only Reynolds number, Re$_{LO}$, Eq. (4.1),
to find liquid-only friction factors, $f_{LO}$, Eq. (4.2) and a vapor-only Reynolds number, $Re_{LO}$, Eq. (4.3) to find a vapor-only friction factor, $f_{VO}$, given by Eq. (4.4).

$$Re_{LO} = \frac{GD_h}{\mu_l} \quad (4.1)$$

$$f_{LO} = 0.079Re^{-0.25} \quad (4.2)$$

$$Re_{VO} = \frac{GD_h}{\mu_v} \quad (4.3)$$

$$f_{VO} = 0.079Re^{-0.25} \quad (4.4)$$

The two-phase pressure drop given by Eq. (4.5) is expressed in terms of a liquid only two-phase multiplier corresponding to single-phase liquid as given by Eq.(4.6)-(4.8). The two-phase multiplier is a function of the liquid-only and vapor-only density, viscosity, Weber number and Froude number.

$$\left( -\frac{dP}{dz} \right) = \phi_{LO}^2 \frac{2f_{LO}G^2}{\rho_lD_h} \quad (4.5)$$

$$\phi_{LO}^2 = C_1 + \frac{3.24C_2}{Fr^{0.045}We^{0.035}} \quad (4.6)$$

$$C_1 = (1 - x)^2 + x^2 \left( \frac{\rho_l f_{VO}}{\rho_v f_{VO}} \right) \quad (4.7)$$

$$C_2 = x^{0.78}(1 - x)^{0.24}\left\{ \left( \frac{\rho_l}{\rho_v} \right)^{0.91} \left( \frac{\mu_l}{\mu_v} \right)^{0.19} \left( \frac{1 - \mu_v}{\mu_l} \right)^{0.7} \right\} \quad (4.8)$$

The Weber number is the dimensionless ratio of the fluids inertia to its surface tension defined by Eq. (4.9). The Froude number is the dimensionless ratio of the characteristic fluid velocity to gravitational forces as defined by Eq. (4.10). The two-phase mixture density is calculated by Eq. (4.11).

$$We = \frac{G^2D_h}{\rho_{tp}\sigma} \quad (4.9)$$

$$Fr = \frac{G^2}{gD_h\rho_{tp}} \quad (4.10)$$
\[
\rho_{TP} = \left( \frac{x}{\rho_v} + \frac{1 - x}{\rho_l} \right)^{-1}
\]

(4.11)

The measured pressure drop includes additional losses at the channel entrance and exit, as well as changes in the acceleration. These effects must be accounted for in order to compare the experimentally determined pressure drop to the predictions of the models. The measured pressure drop for two-phase flow is given by Eq. (4.12). The equation accounts for the effects of contraction at the inlet, expansion at the exit, frictional pressure loss in the core, and momentum changes caused by the changing vapor fraction as condensation takes place along the channel.

\[
\Delta P_{meas} = \Delta P_{ent} + \Delta P_{fric} + \Delta P_{mom} + \Delta P_{exit}
\]

(4.12)

Hewitt et al. [136] recommend a homogenous flow model to account for contraction effects at the entrance using Eq. (4.13), where \( C_C \) is the coefficient of contraction given by Eq. (4.14) [137].

\[
\Delta P_{ent} = \frac{G^2}{2\rho_f} \left[ \left( \frac{1}{C_c} - 1 \right)^2 + 1 - \sigma^2 \right] \left[ 1 + x \left( \frac{\rho_f}{\rho_g} - 1 \right) \right]
\]

(4.13)

\[
C_c = \frac{1}{0.639(1 - \sigma)^{0.5} + 1}
\]

(4.14)
Hewitt et al. recommend a separated flow model to account for the expansion effects at the exit using Eq. (4.15) where $\psi_S$ is the separated flow multiplier given by Eq. (4.16).

$$\Delta P_{\text{exit}} = \frac{G^2 \sigma (1 - \sigma) \psi_S}{\rho_f} \quad (4.15)$$

$$\psi_S = 1 + \left( \frac{\rho_f}{\rho_g} - 1 \right) \left[ 0.25 x (1 - x) + x^2 \right] \quad (4.16)$$

Carey [138] recommends Eq. (4.17) to account for the pressure drop due to momentum changes as the fluid decelerates due to the change in quality where $\alpha$ is the void fraction given by Eq. (4.18). The void fraction is the ratio of the cross sectional area occupied by the gas relative to the total cross sectional area of the channel. The calculation of this parameter is considered essential to predicting the pressure drop in two-phase flow and various models have been proposed to evaluate this parameter in the literature [43].
\[ \Delta P_{mom} = \left[ \frac{G^2 x^2}{\rho_g \alpha} + \frac{G^2 (1-x)^2}{\rho_f (1-\alpha)} \right]_{x=x_{out}} \]

\[ - \left[ \frac{G^2 x^2}{\rho_g \alpha} + \frac{G^2 (1-x)^2}{\rho_f (1-\alpha)} \right]_{x=x_{in}} \]

\[ \alpha = \frac{A_v}{A} \]  

Figure 4.1B depicts the measured pressure drop for Device A and Device B for the case of R245fa condensing at 80 °C. After entrance, exit, and momentum effects are accounted for, the experimental pressure drop agrees with the predictions of both the Friedel and Lockhart Martinelli. A study accounting for the quality at the inlet and exit would be required to provide a more thorough investigation on this subject. These models are appropriate for the intent of designing heat exchanger systems at this scale where the channel length is very short where ancillary effects due to momentum, contraction, and expansion are on the order of relevance as the frictional pressure drop.

4.3 Heat Exchanger Performance

Energy is transferred in the exchanger from the condensing refrigerant to the air flow. The energy transferred between the two fluids in the heat exchanger is evaluated considering the reduction of the energy equation for the air-side fluid stream. The change in energy between any two points in the air flow can be calculated by Eq. (4.19) assuming a steady, compressible flow, with negligible potential energy change, and negligible external volume forces [34].

\[ Q_{air} = c_{p,air}(T_{0,\text{out}} - T_{0,\text{in}}) \]  

An energy balance on the heat exchanger is satisfied for a control volume including the cross flow heat exchanger assuming the exchanger is well insulated, there is no heat loss to the surroundings by the equality given in Eq. (4.20).

\[ Q_{air} = Q_{ref} \]  

Figure 4.2 shows the measured performance for the 1 cm³ heat exchanger devices for two-phase refrigerant flow as a function of air-side Reynolds number. Device A and B are characterized at saturation temperatures of 80 °C with and a refrigerant-side mass flux of 750 kg/m²-s. The heat exchangers’ power density performance increased as a function of the air-side Reynolds number.
For Device A and Device B heat rates of 114 – 152 W and 127 – 179 W were achieved for the respective Reynolds number ranges 9000 – 20,000 and 8000 – 13,500 and in-tube refrigerant-side condensation at a saturation temperature of 80 °C. The additional performance enhancement of increasing the refrigerant-side mass flux was observed but not reported due to the investigations focus on high performance results. The thermal conductance for Device A and Device B are shown in Figure 4.3 for the same range of parameter operation described. A total thermal conductance of 3.3 K/W and 3.7 K/W are achieved for Device A and B respectively for the highest air-side flow rate tested.

![Graph](image.png)

**Figure 4.2** Heat rate power performance of Device A and Device B with two-phase refrigerant side heating at a mass flux of 750 kg/m²-s as a function of air-side Re.
4.3.1 Two-Phase Condensation

The heat rate was experimentally determined from the thermodynamic energy balance between the upstream and downstream air-side flow which was carried out and validated during single-phase liquid flow of the refrigerant. Evaluation of the experimental heat rate and log-mean temperature difference gave the overall heat transfer coefficient and the overall thermal resistance. Characterization of the single-phase performance yielded air-side convection coefficients in to be agreement with the Sieder-Tate correlation. The air-side convective resistance and conduction resistance were determined using the result from Sieder-Tate and COMSOL models respectively. The refrigerant-side resistance was determined Eq. (4.21) which was used to calculate the refrigerant-side convection coefficient with Eq. (4.22).

\[
R_{ref,TP} = R_{tot} - R_{air} - R_{cnd} \tag{4.21}
\]

\[
h_{ref} = \frac{1}{R_{ref}A_{ref}} \tag{4.22}
\]
Akers defines an equivalent mass flux, Eq. (4.23), to express an equivalent Reynolds number given by Eq. (4.24). The Akers Nusselt number correlation for $\text{Re}_e < 50,000$ is given by Eq. (4.25), where $Pr_l$ is the liquid-only Prandtl number.

\[
G_e = G \left[ (1 - x) + x \left( \frac{\rho_l}{\rho_v} \right)^{0.5} \right] 
\]

(4.23)

\[
\text{Re}_e = \frac{G_e D_h}{\mu_l} 
\]

(4.24)

\[
\text{Nu}_{Akers} = 5.03 \text{Re}_e^{1/3} Pr_l^{1/3} 
\]

(4.25)

Figure 4.4 compares the experimentally derived refrigerant-side convection coefficients those obtained from the Akers model. The Akers model predictions are a strong function of refrigerant vapor quality. The vapor-only maximum and the liquid-only minimum limits of the model are shown as limits of the prediction. Since the vapor refrigerant quality was not accounted for in the experiment there is not an explicit way to compare the model to the experimental data. As evidenced in the figure, the convection coefficients are within the range of what the Akers model predicts for the possible range of refrigerant vapor quality.

Figure 4.5 shows the total thermal resistance is decomposed into constitutive terms accounting for air-side convective resistance, refrigerant-side convective resistance and the conduction resistance of the exchanger. The thermal resistance on the air-side is slightly higher than the thermal resistance on the refrigerant side but the two thermal resistances are well matched as intended by the design for two-phase flow.

There have been few studies on measurement of heat transfer that address $D_h < 1$ mm. Yang and Webb [139] found the Akers correlation better predicted their results than the Shah correlation. The heat transfer coefficient in their work showed a heat flux dependence $h \sim Q^{-0.2}$. Heat flux dependence is typically only observed in stratified flows that only occur at very low mass fluxes. It was reasoned that the observed dependence of the heat transfer coefficient on heat flux was due to the momentum contribution [140]. Wang et al. [141] investigated condensation of R134a in 10 multiport 610-mm long rectangular channels with a hydraulic diameter of 1.46 mm (1.5 mm x 1.4 mm) over the mass flux range 75-750 kg/m²-s. The study reports strong heat transfer coefficient dependence on quality at high mass fluxes, a sign of annual flow conditions. Of the available heat transfer correlations their data agree best with the Akers model. The study also
suggests that the rectangular corners aided in the formation of annular flow, and an earlier transition to annular flow was observed than expected with the Soliman map [43].

Cooling of superheated vapors involves condensation when the wall temperature is below the saturation temperature of the vapor. If the wall temperature is below the saturation temperature, then condensation of the superheated vapor will occur in the thermal boundary layer on the tube wall, even though the bulk vapor is superheated. Since the condensing heat transfer is much more effective than single-phase heat transfer to a vapor it is important to include the effect of the de-superheating zone with respect to the saturated condensing zone. The saturated zone method is usually evaluated at a vapor quality of 0.99. This scenario illustrates the dependence on the process path the system follows to reach its operating state. If the hot refrigerant is applied to the already cooled heat exchanger the de-superheating zone is found using the condensing heat transfer coefficient in the thermal resistance analysis rather than the single-phase heat transfer coefficient of the vapor phase [91].

Figure 4.4 Experimentally reduced refrigerant-side convection coefficients as a function of the heat flux compared to the Akers model for a refrigerant quality of \( x=1 \) and \( x=0 \).
Figure 4.5 The total thermal resistance for A) Device A and B) Device B is decomposed into constitutive terms accounting for air-side convective resistance, refrigerant-side convective resistance and the conduction resistance of the exchanger.
4.3.2 Thermal Conduction Heat Transfer

The heat flow from one isothermal surface to another isothermal surface can be calculated according to the simple relationship given in Eq. (4.26). The conduction resistance is inversely proportional to the thermal conductivity of the material between the two surfaces. The relationship represented by Eq. (4.27) introduces a shape coefficient, $S$, commonly referred to as a shape factor in conductive heat flow. The shape factor only depends on the geometrical arrangement of the two isothermal surfaces between which heat is transferred by conduction.

\[ Q_{\text{cnd}} = \frac{T_1 - T_2}{R_{\text{cnd}}} \]  
\[ R_{\text{cnd}} = \frac{1}{kS} \]  
\(4.26\)
\(4.27\)

The shape factor can be determined by integration of the local heat flux over the isothermal surfaces $A_1$ and $A_2$. The heat flow is given by Eq. (4.28) in which the surface normals $n_1$ and $n_2$ are directed into the conductive medium. The following relationship enables the shape factor to be calculated from the known temperature field [133].

\[ Q_{\text{cnd}} = -k \int_{A_1} \frac{\partial T}{\partial n_1} dA_1 = k \int_{A_2} \frac{\partial T}{\partial n_2} dA_2 \]  
\(4.28\)

COMSOL Multiphysics was used to calculate shape factors for the heat exchanger geometries over the range of air-side and refrigerant-side boundary conditions explored. The steady state temperature profile of a three-dimensional model of each heat exchanger was obtained by application of the average temperature and heat transfer coefficients obtained from the Wilson plot method. The top and bottom of the heat exchanger are treated as adiabatic surfaces. The heat flow was calculated by a surface area integration over the refrigerant-side area. The shape factor was calculated by dividing the heat flow rate, determined from the surface integration, by the temperature difference between the surface of the refrigerant-side channel and the nearest air-side channel. The relationship given in Eq. (4.29) reduces to Eq. (4.30) by carrying out the surface integration over the transfer area.

\[ S = -\frac{1}{T_1 - T_2} \int_{A_1} \frac{\partial T}{\partial n_1} dA_1 = \frac{1}{T_1 - T_2} \int_{A_2} \frac{\partial T}{\partial n_2} dA_2 \]  
\(4.29\)

\[ S = \frac{Q}{k(T_1 - T_2)} \]  
\(4.30\)

87
The thermal conductivity of pure copper varies \(>1\%\), 396.9 – 393.4, over the temperature range 20 – 80 °C. The tellurium copper alloy has a thermal conductivity of 354.6 at 20 °C, due to a lack of readily available thermal conductivity data on this alloy, this value is assumed constant in the analysis. The temperature dependence of the thermal conductivity could be readily incorporated by implementation of Eq. (4.31). A function to include this effect is available in COMSOL Multiphysics for many common materials, however this data is unavailable for the tellurium-copper alloy used in this research.

\[
k = \frac{1}{T_1 - T_2} \int_{T_2}^{T_1} k(T) dT
\]  

(4.31)

4.4 Performance Comparison

The resistance model predictions for two-phase refrigerant-side performance are compared to the experimental data in Table 4.1. The resistance model power dissipation is given using the measured log mean temperature divided by the total thermal resistance. The modeled thermal resistances were determined using the Sieder-Tate on the air-side and range of the Akers model for the refrigerant-side. Table 4.2 compares to the resistance model predictions and the COMSOL simulation results to the experimental results with two-phase heating condition. The simulation results were attained by applying the experimentally obtained convection coefficients and the mean temperature to a solid model of the device geometry COMSOL. The simulation power performance was obtained by integrating steady state temperature profile over the heat exchanger surface area with the given convection coefficients.

<table>
<thead>
<tr>
<th></th>
<th>Device A</th>
<th></th>
<th>Device B</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Model</td>
<td>Experiment</td>
<td>Model</td>
<td>Experiment</td>
</tr>
<tr>
<td><strong>Q [W]</strong></td>
<td>127.8-158.6</td>
<td>151.6</td>
<td>153.1-196.0</td>
<td>178.6</td>
</tr>
<tr>
<td><strong>R_{tot} [K/W]</strong></td>
<td>0.29-0.36</td>
<td>0.3</td>
<td>0.25-0.32</td>
<td>0.27</td>
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<tr>
<td><strong>R_{air} [K/W]</strong></td>
<td>0.18</td>
<td>0.18</td>
<td>0.14</td>
<td>0.14</td>
</tr>
<tr>
<td><strong>R_{ref} [K/W]</strong></td>
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<td>0.11</td>
<td>0.10-0.17</td>
<td>0.12</td>
</tr>
<tr>
<td><strong>R_{cnd} [K/W]</strong></td>
<td>&lt;0.01</td>
<td>&lt;0.01</td>
<td>&lt;0.01</td>
<td>&lt;0.01</td>
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<tr>
<td><strong>LMTD</strong></td>
<td>-</td>
<td>46.0</td>
<td>-</td>
<td>49.0</td>
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</tbody>
</table>

Table 4.1 Table comparing models and experimental data for the two heat exchangers with two-phase refrigerant heating.
Table 4.2 Two-phase refrigerant heating data for Device A and Device B are compared to the expected values as calculated by the modeled thermal resistances and by the application of the experimental convection coefficients to a 3-D COMSOL model.

<table>
<thead>
<tr>
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<tbody>
<tr>
<td>Device A</td>
<td>151.6</td>
<td>127.8-158.6</td>
<td>152.5</td>
</tr>
<tr>
<td>Device B</td>
<td>178.6</td>
<td>153.1-196.0</td>
<td>186.8</td>
</tr>
</tbody>
</table>

4.5 Conclusions

Two-phase refrigerant-side pressure drops were measured for the case of R245fa condensing at 80 °C in rectangular minichannels in Device A and Device B. The experimental results were compared to the conventional correlations recommended by Friedel. Given the short channel lengths and very small pressure drops incurred it was determined that these models were adequate for the purpose of designing heat exchanger systems with short lengths. It is emphasized that the additional pressure drop effects due to changes in momentum of the condensing liquid, contraction at the inlet, and expansion at the exit are on the same order of relevance as the frictional pressure drop effect. It is recommended that these effects be accounted for in determination of two-phase condensation pressure drop when designing heat exchangers employing short channel lengths.

The friction factors on the air-side were the same for the case of single-phase and two-phase refrigerant-side heating. The measured friction factors for the device with the smaller hydraulic diameter of 335 μm are less than those measured in the 520 μm channels.

The heat transfer performance for two 1 cm³ novel heat exchangers were characterized by application of the first law of thermodynamics to the free stream of the air flow in order to calculate the total energy rate dissipated in the device. Average convection heat transfer coefficients for air-side and single phase refrigerant-side flow were obtained using the well-known modified-Wilson plot method. An average two-phase convection coefficient on the refrigerant-side with condensation phase change was obtained and found to be in agreement with the results of Akers condensation model.

The overall heat transfer performance for two 1 cm³ heat exchangers were compared for a range of operating conditions. Comparison with the literature for heat transfer in compact heat exchangers show that these results were reasonably predicted through utilization of the conventional theory. The experimentally obtained two-phase refrigerant-side convection
coefficient were within the expected range predicted by the Akers model for qualities between 0 and 1.

Experimental techniques in the design of compact micro-heat exchangers are validated for designing compact heat exchangers with high power density performance. For the case of two-phase refrigerant-side heating, the adjustments to the air-side were more pronounced than in the single-phase case and the air-side and refrigerant-side thermal resistances were well matched. Total thermal resistances of 0.3 K/W and 0.27 K/W for the 1 cm³ heat exchangers are reported.
CHAPTER 5: SCALING OF METHODOLOGY

Figure 5.1 shows photographs of Device C which comprises 600 circular air-side microchannels with an average microchannel diameter of 482 μm configured into a high-density honeycomb array with an average center-to-center offset pitch spacing of 671 μm. Device C has 14 refrigeration channels configured into two rows of 7 horizontal rectangular minichannels with a cross sectional aspect ratio of 0.5 x 2 mm separated by 0.5 mm center-to-center pitch offset. The air side and refrigerant side characteristics are summarized in Table 5.1 Device C has a surface area to volume ratio of 2111 m²/m³ and an air-side aspect ratio of 41.5.

Figure 5.1 A) Isometric photograph of 10 cm³ heat exchanger, Device C. Microscope images of B) Air-side and C) refrigerant-side are shown.
Table 5.1 Summary of the air-side and refrigerant-side characteristics of 10 cm³ manufactured heat exchangers.

The performance of Device C is simulated in COMSOL as shown for the range of expected boundary conditions based on the experimental investigation of the two 1 cm³ devices. Scaling of these techniques were applied to a 10 cm³ heat exchanger. A power performance of 1 kW and a power density performance of 100 W/cm³ with an overall thermal resistance < 0.05 are achievable. The expected performance of the device with two-phase refrigerant-side flow at a saturation temperature of 80 °C and an average air-side temperature of 37 °C are shown for the for air-side convection coefficients 1000 – 4000 W/m²K and refrigerant-side convection coefficients 10,000 – 16,000 W/m²K.
Figure 5.2 Simulated power density power dissipation and overall thermal resistance for the range of expected air-side and refrigerant-side boundary conditions.
6.1 Significance of Work

The current investigation demonstrates the design, fabrication, and characterization of ultra-compact air-cooled condenser heat exchangers with exceptional power density performance of up to 180 W/cm³, exceeding the program goal of 100 W/cm³. This exceptional heat exchanger performance is attributed to the implementation of high-speed compressible air flow in circular microchannels and two-phase condensation flow of a refrigerant in rectangular minichannels to achieve high heat transfer performance in a compact design. The heat exchangers are air-cooled with air speeds and flow rates that are in a relatively unexplored range for power electronics cooling. High-speed turbulent flow is explored in parallel copper microchannels. The heat exchangers utilize dielectric refrigerant as the heat transfer working fluid. Performance results are obtained with single-phase liquid phase and phase change condensation of refrigerant R245fa. Phase change is induced at desirable temperatures and reasonable pressures. These advancements are enabled by recent developments in micro-electric discharge machining to produce very high surface area compact heat exchanger devices.

Air-cooled heat sinks have been identified as pivotal to the future of thermal management of microelectronics in the 21st century. Air-cooling systems provide clear advantages in overall ease of integration due to its availability and abundance over forced liquid cooling systems. The enhancement of forced air flow is especially important because it generally represents the dominant thermal resistance of the condenser. The implementation of high-speed compressible air flow in microchannels permits investigation of flow conditions with exceptionally high heat transfer coefficients. This investigation explores turbulent flow regime conditions for air-side Reynolds numbers 8,000 < Re < 20,000, in high-density parallel arrangements of 1 cm long circular copper microchannels, 355 and 520 μm in diameter.

Two-phase heat transfer technology is one of the most efficient methods of waste heat removal in high power electronics cooling. It is especially advantageous in applications where size, weight, and efficiency are important factors. Two-phase active cooling systems consist of an evaporator and a condenser. Heat transfer performance in the evaporator is typically much higher than in the condenser; consequently the condenser is the limiting component of the entire cooling system. Improvements in condenser technology enable electronics systems to operate at a higher
power while reducing the overall cooling system size and weight. The condensation phase change process in microchannels with high aspect ratios yields the formation of a thin film condensation layer on the heat transfer wall, resulting in high heat transfer coefficients with little pressure drop penalty. It is therefore of utmost importance that condensation phenomena, especially in high aspect-ratio microchannels be experimentally investigated. This investigation employs condensation in a parallel array of high-aspect ratio, 0.5 mm x 2 mm, rectangular minichannels.

Compact heat exchangers are manufactured using novel micro-electro-discharge machining to produce high-density, high aspect ratio microchannels in a copper alloy. The heat exchanger performance is characterized for single-phase liquid and phase-change condensation of a refrigerant. The heat exchangers are operated using single-phase liquid flow of dielectric refrigerant R245fa at 80 °C and high-speed flow of air at ~25 °C to demonstrate a power density performance of nearly 70 W/cm³. The heat exchangers are operated using two-phase condensation of R245fa at 80 °C and high-speed flow of air at ~25 °C to achieve power density performance > 175 W/cm³ and overall thermal resistance < 0.27 K/W.

Test methodologies were implemented for determination of the thermal-hydraulic performance of these novel devices. Modeling and characterization of this system were implemented using well-known methods and the results are compared with the corresponding literature for microchannel fluid flow and heat transfer. The results of this work are compared to the state of the art of air-cooled heat exchangers. The Sandia cooler reports performance with a normalized volume of ~0.3, and a thermal resistance of ~0.15 K/W [22]. Published experimental results on the PHUMP which employs an impeller motor to cool a single condenser loop heat pipe with a condenser 26 cm³ (10.2 x 10.2 x 0.25 cm) in size, dissipated a heat load of 200 W, (a power density of 7 W/cm³) and an overall thermal resistance of 0.177 K/W [25]. The study of this system demonstrates an advancement in the known state-of-the-art in power density performance of compact air-cooled heat sinks. Power dissipation rates > 1 kW and an overall thermal resistance of < 0.05 K/W are projected with scaling of these methodologies in a 10 cm³ device.

This work provides direct understanding and validation of test methodologies and characterization techniques for applications of these previously unexplored operating conditions for the application of electronics cooling. The results have generality and impart value beyond the immediate application in industries ranging from military, power electronics, power energy, automotive, HVAC, cryogenics, and MEMS. The modeling and characterization techniques
employed were validated for microchannel geometries where the continuum assumption remains valid down to 10 μm channel diameter.

6.2 Conclusions

The design and manufacturing for μEDM manufactured copper compact heat exchangers are presented. Experimental design for air side and refrigerant side test facilities utilized to characterize the compact heat exchangers are described. Experimental procedures for single-phase and two-phase refrigerant heating over a range of operating conditions are detailed. The experimental methodology including measurement strategy and uncertainty analysis is discussed.

Air-side pressure drops were measured in Device A and B. Well known methods were employed to reduce the data accounting for inlet, momentum, and acceleration effects in order to produce experimentally determined friction factor data for compressible air-flow in 1 cm long circular microchannels with hydraulic diameters of 335 and 520 μm. The experimentally obtained data agrees with the isothermal compressible flow calculation to within 1%. The experimentally obtained friction factors agree with the predictions of the adiabatic compressible Fanno flow model within 15%. The measured friction factor for both devices are shown to decrease with increasing Reynolds numbers. The measured friction factors for the device with the smaller hydraulic diameter of 335 μm are less than those measured in the 520 μm channels.

Single-phase liquid refrigerant-side pressure drop for developing flow in eight parallel rectangular copper channels having a hydraulic diameter of 0.8 mm and an aspect ratio of 0.25 were obtained. Flow development effects are shown to be relevant for 1 cm long channel. The experimentally measured apparent friction factor was in agreement with the predictions of Phillips [109]. Developing flow effects are relevant in determination of the pressure drop characteristics in these devices and should be thusly incorporated into design of system when operating with single-phase liquid only fluid flow in short rectangular microchannels. The pressure drop of single-phase liquid flow in rectangular channels are described by developing laminar flow in rectangular channels once entrance and exit effects were accounted for.

The conversion of thermal energy to kinetic energy by the expanding gas is relevant and must be accounted for in the air-side energy balance. Heat transfer performance of high-speed air flow with Re > 20,000 in circular microchannels were accurately predicted by the Sieder-Tate correlation. Developing flow effects dominate single-phase liquid heat transfer in rectangular
microchannels and should be accounted for in the design phase. The surface roughness effects and boundary conditions at the front and rear faces of the heat exchanger are thought to enhance the overall entrance effects for the transitional Reynolds numbers explored.

The experimental condensation heat transfer coefficients were within the bounds predicted by Akers model. The two-phase pressure drop was within the range predicted by both the Friedel model once entrance, exit, and momentum pressure drop effects were accounted for. The losses due to contraction, expansion, and momentum of the condensing fluid are equally relevant to the frictional loss in 1 cm long rectangular passages and should be incorporated at the design stage for predicting the total pressure drop.

6.3 Recommendations for Future Work

Research focused on scaling these methodologies to achieve 1 kW of power dissipation at a power density of 100 W/cm³ is underway and is expected to be achieved through in Device C introduced in Figure 5.1. Further advances beyond this can be realized by creating shorter air-side fins for the heat exchanger by the addition of more rows of refrigerant-side channels.

Given the volatility of the refrigerant-side heat transfer coefficient as a function of quality, methods to regulate the saturation conditions for optimal performance are recommended. Techniques for adjusting the saturation conditions, such as the use of an electrical heater attached to the surface of a liquid reservoir regulated by a PID controller to stabilize the pressure may be employed as a means of stabilizing the saturation conditions to optimize device performance [84]. By stabilizing the saturation conditions and accounting for the quality changes of the refrigerant, a deeper understanding of the phase change characteristics of the refrigerant can be realized.

Modeling cross-flow heat exchangers with additional cross channels show increased heat transfer performance. The validation of the models demonstrated by this research provide for the engineering design of producing similar compact heat exchangers with improved power density performance and a reduced overall thermal resistance. The heat exchangers employed in this work can be optimized by integration of more refrigerant-side channels and by utilization of the top and bottom surfaces of the devices in heat exchange.

The opportunity of placing several heat exchangers in series with new air being injected into the flow stream promise viability for scaling of these techniques to accommodate higher heat loads. While the temperature of the air flow increases due to heat transfer it simultaneously cools
as it expands along the flow length. Demonstrations of these types of optimization techniques where new fluid is injected into the stream have already being employed to improve efficiency of next generation power plants and in the production of hydrogen. Future research towards the integration of employing multiple heat exchangers in parallel and serial configurations is recommended.
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