Thermal Spray Forming of High-Efficiency Metal-Foam Heat Exchangers

Department of Mechanical and Industrial Engineering
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A dissertation submitted in conformity with the requirements for the degree of

Masters of Applied Science

by:

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Abstract

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Thermal spray coating processes have been employed in the current study to deposit well-adhered, dense skins onto the surfaces of open-cell metal foams. The result is a channel that consists of a metal foam core and a thermal sprayed skin wall that can be used as a compact heat-exchanger by directing the coolant flow through the foam. To study the feasibility of the metallic foam heat-exchangers, hydraulic and heat-transfer characteristics were investigated experimentally. The local wall and fluid temperature distribution and the pressure drop along the length of the heat exchanger were measured for different coolant flow velocities. The Dupuit-Forchheimer modification is employed with the experimental results to determine flow characteristics. To measure the heat transfer performance, a length average Nusselt number is derived from a volumetric heat transfer coefficient based on the local wall and fluid temperatures. Heat transfer was shown to have increased nearly 7 times compared to that of a channel without a foam core.
In Memoriam

I dedicate this thesis to those who were unable to witness my accomplishments over the years, their memory has provided me with a source of encouragement and inspiration.

Nikolaos Athanasios Tsolas
(1923-1988)

Julianna Biro
(1935-2008)
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Nomenclature

\( A \) = coefficient for curve fitting (kg/m\(^3\)s)
\( A \) = cross-sectional area
\( A_{sf} \) = surface area of foam
\( B \) = coefficient for curve fitting (kg/m\(^4\))
\( b \) = thickness of insulation
\( c \) = heat capacity ratio = \( C_{min}/C_{max} \)
\( C_F \) = inertial coefficient
\( c_p \) = specific heat (J/kgK)
\( d_f \) = fiber diameter (mm)
\( D_{hyd} \) = hydraulic diameter
\( d_i \) = inner fiber diameter (mm)
\( d_p \) = pore diameter (mm)
\( Da \) = Darcy factor = \( K/D_{hyd}^2 \)
\( e \) = error associated with a given instrument
\( f \) = friction factor
\( H \) = height of the channel
\( h \) = surface area heat transfer coefficient (W/m\(^2\)K)
\( h_{sf} \) = volumetric heat transfer coefficient (W/m\(^3\)K)
\( j \) = Colburn-factor
\( K \) = permeability of porous medium (m\(^2\))
\( k \) = thermal conductivity (W/mK)
\( k_{eff} \) = effective thermal conductivity (W/mK)
\( L \) = length (mm)
\( m \) = mass (g)
\( NTU \) = number of transfer units
**NOMENCLATURE**

\[ Nu = \text{Nusselt number} \]
\[ Nu_{dp} = \text{Nusselt number based on the pore diameter} \]
\[ Nu_{H} = \text{Nusselt number based on the height of the channel} \]
\[ P = \text{perimeter} \]
\[ P = \text{pressure (Pa)} \]
\[ Pr = \text{Prandtl number} \]
\[ Q_{H} = \text{power provided by the heater (W)} \]
\[ r_f = \text{hollowness ratio} = d_i/d_f \]
\[ R_H = \text{resistance of the heater (Ω)} \]
\[ R_{th} = \text{thermal resistance (K/W)} \]
\[ Re = \text{Reynolds number} \]
\[ Re_{dp} = \text{Reynolds number based on the pore diameter} \]
\[ Re_{K} = \text{Reynolds number based on the permeability} \]
\[ s = \text{standard deviation} \]
\[ T_{\text{fluid}} = \text{temperature of the fluid phase (°C)} \]
\[ T_{\text{loss}} = \text{temperature measured at the exterior of the insulation (°C)} \]
\[ T_{\text{solid}} = \text{temperature of the solid phase (°C)} \]
\[ U = \text{overall heat transfer coefficient (W/m}^2\text{K)} \]
\[ u = \text{fluid flow velocity (m/s)} \]
\[ V = \text{voltage (V)} \]
\[ V = \text{volume (mm}^3\text{)} \]
\[ W = \text{pumping power (W)} \]

**Greek Symbols**

\[ \delta = \text{uncertainty in a measurement} \]
\[ \alpha_{sf} = \text{surface area density (mm}^2\text{/mm}^3\text{)} \]
\[ \Delta T_{lm} = \text{log mean temperature difference} \]
\[ \eta_s = \text{strut efficiency} \]
\[ \mu = \text{dynamic viscosity (kg/m s)} \]
\[ \nu = \text{kinematic viscosity (m}^2\text{/s)} \]
\[ \rho = \text{density (kg/m}^3\text{)} \]
\[ \rho_{eff} = \text{effective relative density} \]
\[ \rho_{rel} = \text{relative density} \]
\[ \varepsilon = \text{porosity} \]
\[ \varepsilon_{eff} = \text{effective porosity} \]
\[ \varepsilon_{model} = \text{porosity computed from the ideal model} \]
\[ \xi_s = \text{strut effectiveness} \]
Chapter 1

Introduction

1.1 Introduction

The use of open-cell metal foams in numerous technological applications has increased rapidly in recent years. Through improved manufacturing processes, metallic foams can be produced in a variety of materials and specifications at a relatively low cost. The use of metal foams in heat exchanger applications is fairly recent, with commercially viable technology still under development and mainly a subject of fundamental and applied research [7]. The porous cell structure provides all the ideal qualities for a well designed heat exchanger. The high porosity (often greater then 0.9), provides a large accessible solid-to-fluid surface area, combined with a high thermal conducting metallic phase, such as aluminum ($k = 200$ W/mK) or copper ($k = 400$ W/mK) allows for enhanced heat transfer by conducting heat from the metallic struts to the passing fluid phase [59]. The induced turbulence and dispersion as a result of tortuous paths through the foam further enhances heat transfer [59]. It is through such characteristics that metallic foam heat exchangers experience better performance and efficiency than current conventional designs. In order for metallic foams to be used in heat exchanger applications, skins need to be applied to the structure of the foam to contain the flowing fluid. In the past, processes such as cladding, brazing, and diffusion bonding of metal sheets where used to form skins [6]. However these methods proved problematic, as reduced bonding contact between the skin and foam was apparent, subsequently diminishing their overall performance but also limiting their design to simple shapes. Through a new innovative method developed by the Center for Advanced Coating Technologies at the University of Toronto, metallic skins can be applied through a thermal spray coating process that leaves the foam core structure uninhibited to allow fluid flow while providing enhanced contact between the skin and foam.
CHAPTER 1. INTRODUCTION

The purpose of the present study was to characterize the hydraulic and heat transfer performance of thermal sprayed metal foam heat exchangers.

1.2 Literature Review

Metal foams are recognized as an innovative material with diverse properties, particularly for its ability to enhance heat transfer due to its unique and random structure. Researchers have mainly experimented with metal foams in order to characterize their fluid flow and heat transfer behavior. In this section a summary of the relevant research regarding metal foams is presented.

Bastawros (1998) [1] demonstrated the efficiency of metallic foams in forced convection applications. He was able to show that a cellular aluminum heat sink can remove 2-3 times the usual heat flux compared to a pin-fin array, at a third of the weight and with only a moderate increase in the pressure drop. At low air flow velocities, the heat flux was dominated by the convective heat transfer to the flowing fluid. At higher fluid velocities, it was shown that the heat flux was limited by the heat conduction from the substrate to the foam. Later on Bastawros et al. (1999) [24] conducted experiments on the heat transfer and fluid flow of metal foams subjected to transverse airflow during steady-state. The experimental work was performed on 30PPI aluminum foams with porosities of $\varepsilon > 0.90$. It was found that a power law followed when pressure versus velocity plots were recorded. The thermal measurements were correlated with models on thermal dispersions in porous media. These correlations revealed that the filaments normal to the flow direction transmitted most of the heat flux.

Khayargoli et al. (2004) [2] studied the effect of the metal foam microstructure on the flow parameters. To ensure different microstructure geometries, samples used in this study were obtained from different manufacturers. They used various samples from RECEMAT and IMI made of nickel and nickel-chromium alloys with porosities ranging from 0.83 to 0.90. It was observed that as the pore size decreased the surface area increased creating additional flow resistance. The permeability increased and the inertia coefficient decreased with increasing pore size diameter, but did not show any clear relation with the porosity. They claimed the greater permeability with large pore sizes was due to increases in drag forces on the flowing fluid which also contributed to the pressure drop. They finally concluded that although the flow phenomenon in the medium was complex, the flow parameters could be predicted using an Ergun like correlation.

Boomsma et al. [28] did some research with aluminum foams of 40PPI pore densities compressed to give various porosities, while using water as the working fluid. The flow parameters for these samples
were obtained with a curve fit of the pressure drop versus velocity data. It was found that the structural
differences in the pre-compressed metal foams of 95% and the 92% porosity did not have a noticeable
effect on the permeability. Increasing the compression factor decreased the permeability by regular
incremental amounts and holding the porosity constant while decreasing the pore diameter decreased
the permeability and increased the inertial coefficient. Finally, it was found that changing the velocity
regime resulted in different values for the flow parameters.

Antohe et al. (1997) [48] investigated the use of porous materials in heat exchanger applications, specifi-
cally to develop a heat exchanger composed of mechanically compressed micro-porous matrices for
cooling high frequency microwave systems. He experimented with nine 40PPI samples all at different
compression ratios to study the flow characteristics with air and a poly-alpha olefin fluid. Foam and flow
parameters such as porosity, permeability and inertial-coefficient were calculated. It was observed that
the pressure drop was more sensitive to changes in compression ratio at high flow speeds. The hydraulic
behavior of a compressed matrix seemed to become insensitive to changes in the initial matrix density
beyond a certain threshold value.

An important application of metal foams is its potential to serve as a compact heat exchangers. Boomsma
et al. (2003) [59] showed that compressed open-cell aluminum foams can dissipate large amounts of heat
in electronic cooling applications. Through his experimental investigation the hydraulic characterization
and thermal performance of metal foams were studied, from which he could determine the parameters for
the most efficient heat exchanger in a particular heat transfer scenario. It was seen that the compressed
aluminum foams made a significant improvement in the efficiency over several commercially available
heat exchangers.

Lu et al. (1998) [21] studied the use of open-celled metal foams as heat sinks for high power electronic
devices and multi-layered heat exchangers for aeronautical applications. An analytical model was devel-
oped with simple cubic unit cells consisting of heated slender cylinders, based on existing heat transfer
data on convective cross flow through cylinders banks. A foam-filled channel having constant wall tem-
peratures was analyzed to obtain the temperature distribution inside the channel as a function of foam
density, cell size and other heat transfer parameters. The overall heat transfer coefficient of the heat
exchanging system was calculated, and the pressure drop experienced by the fluid flow was obtained.

Writz (1997) [3] developed a semi-empirical model for the combined conduction and convection heat
transfer in a thin porous wall. The model assumed one-dimensional conduction in the porous matrix
and one-dimensional flow of the coolant through the foam wall. For the same volume of the heat
exchanger, the porous matrix provided approximately 1.5 times more heat transfer surface than the
offset strip fin array.

Tradrist et al. [57] investigated the use of aluminum metal foams for compact heat exchangers. The porosities were $\varepsilon > 0.90$. They experimentally determined the flow parameters permeability and inertia coefficient and used an Ergun type correlation to describe the pressure drop with respect to the fluid velocity.

Kim et al. (2000) [4] studied the impact of porous fins on the pressure drop and heat transfer characteristics in plate-fin heat exchangers. They used six 6101 aluminum alloy foams of 10, 20 and 40PPI with different porosities using water as the fluid. Both the friction factor and heat transfer were significantly affected by the permeability and the porosities of the samples foam fins. They used the Darcy number, the geometry and the Reynolds number to correlate the friction factor and they used the Forchheimer equation[33] for the flow parameters.

Paek et al. (2000) [5] did experimental work with aluminum foams of different porosities (0.89 to 0.96) to determine the materials thermophysical properties. At a fixed porosity, decreasing the cell size increased the surface area in a given volume, which therefore increased the flow resistance by lowering the permeability and increased the pressure drop. So it was inferred that the permeability was influenced appreciably by both the porosity and the cell size. The friction factor was correlated with the permeability based Reynolds number. Also in this work the one-dimensional conductivity of the material was calculated and the results indicated that it decreased as the porosity increased.

1.3 Research Objectives

The objectives of this research are:

- To fabricate compact heat exchangers using metal foams with thermal sprayed skins.
  - To determine the fabrication limitations of thermal spraying metal foams by varying pore density, material and skin material.
  - To design and test different metal foam heat exchangers, including solid-to-gas heat exchangers and water-to-air heat exchangers.

- To measure the pore diameter, fiber diameter and porosity by image analysis for metal foams of different pore densities.
CHAPTER 1. INTRODUCTION

- Using these parameters an ideal geometric model is to be developed to represent the metal foam structure.

- To study the hydraulic characteristics of fluid flow through the metal foam channels.
  - To analyze the effect geometric properties have on the pressure drop, by determining flow parameters such as permeability and inertial coefficient.
  - To develop a friction factor correlation to describe the experimental results using the relevant flow parameters.

- To study the heat transfer characteristics of fluid flow through the metal foam channels.
  - Analyze the effect of foam material and structure on forced convection by determining the convective heat transfer coefficient assuming an ideal foam structure.
  - Develop a Nusselt number correlation based on relevant geometric parameters to describe the experimental results.

- To determine the performance capability of a compact water-to-air metal foam heat exchanger.
  - Develop design correlations to describe the thermal performance of the metal foam heat exchanger.

Parameters Varied:

- Foam material: Nickel ($k_{Ni} = 90.7 \text{ W/mK}$) and Copper ($k_{Cu} = 401.0 \text{ W/mK}$)
- Foam pore size: 10PPI (2.54 mm) and 40PPI (0.64 mm)
- Fluid flow velocities: 0 - 4.13 m/s
- Heat flux: 107 - 6846 W/m$^2$

1.4 Organization of Thesis

The present chapter gives a general background, literature review and objectives of this research thesis. Chapter 2 explains the experimental apparatus and methodology used in the pressure drop and heat transfer experiments.
Chapter 3 introduces metal foams, their industrial applications and their manufacturing process. Relevant geometric parameters used to characterize metals foams is discussed and measured. Several methods are employed to measure parameters such as porosity, relative density and pore and fiber diameters. Measured values of the foam geometry are then used to develop an idealized model to represent the foam structure.

Chapter 4 describes the hydraulic characteristic associated with fluid flow through metal foams. In particular pressure drop is examined, where existing theory on porous mediums is applied to determine relevant flow parameters such as permeability and the inertial-coefficient. These parameter are then used to describe and compare the difference in fluid flow behavior between foams of varying pore densities. The analysis is then extended to develop friction factor correlations such that the pressure drop can be independently characterize based on the experimentally obtained flow parameters.

Chapter 5 describes the heat transfer characteristics associated with metal foams. First, conduction by stagnate flow is examined, where a one-equation model is used to predict the effective thermal conductivity of the porous medium. Second, forced convection is examined where experimental temperature results are applied to a two-equation model to calculate the convective heat transfer coefficient. In this study two definitions of the convective heat transfer coefficient are presented, one based on a volume basis and the other based on surface area. From the computed heat transfer coefficients an overall Nusselt number correlation is developed based on the relevant flow parameters found in Chapter 4. From the analysis, a better understanding of the heat transfer behavior within metal foams is deduced where it can be described that the foams act similar to fins.

Chapter 6 describes the fabrication and experimental testing of a compact water-to-air heat exchanger using metal foam. Using existing correlations the performance of the heat exchanger is evaluated based on the convection heat transfer coefficient on the air side. A trade-off analysis is performed to determine if the gain in heat transfer justifies the increase in pressure losses. At the end of the chapter, design correlations are developed from the experimental results to assist in the selection and design process of future metal foam heat exchangers.

Chapter 7 finally draws conclusions from the present research.

At the end, references, and the appendices containing some schematics and raw data obtained during the experimental runs.
Chapter 2

Experimental Methods

2.1 Introduction

Metal foams sheets with 10 and 40PPI pore densities are used in this study to make single channel tube heat exchangers. Using existing wire arc spray technology, dense metallic skins are applied to the metal foams through a patented method that leaves the foam core structure open to allow fluid flow while providing enhanced contact between the skin and foam. In this study, four different tube channels are made: two nickel foam channels with stainless steel skins and two copper foam channels with copper skins. For all experimental purposes air is used as the working fluid supplied from the laboratory. To measure the hydraulic and thermal performance of the metal foam channels a simple circulation loop is designed and built to measure pressure drop and temperatures across the channel under different conditions. The pressure drop across the channel is measured by a pressure transducer connected to two fixed pressure ports along the length of the channel. A rope heater wrapped around the channel acts a heat source, such that when an applied voltage is varied the heat flux can be varied. Thermocouples are placed at various axial locations along the length of the channel to measure the temperature at the wall and the flowing fluid under a given heat source. The experimental procedure is repeated for different flow rates of air and heat flux intensities.

2.2 Thermal Spray Skin Deposition

In order for metallic foams to be used in heat exchanger applications, skins need to be applied to the structure of the foam to contain the flowing fluid. In the past, processes such as cladding, brazing, and
diffusion bonding of metal sheets where used as skins [6]. However these methods proved problematic, as reduced bonding contact between the skin and foam was apparent, subsequently diminishing their overall performance but also limiting their design to simple shapes. Difficulties in forming the skins on the surfaces of the foam structures have limited industrial applications, especially for complex shaped heat exchangers. The following section describes the process in which thermal sprayed skins are applied to the metal foams to create tubular channels. In total four samples were made, by varying the pore densities and applying different skin materials. The samples are used for all experimental purposes as described in Chapter 4 and 5 to measure both hydraulic and heat transfer characteristics.

For the first samples nickel and copper foam sheets with a thickness of 20 mm and pore densities of 10 and 40 PPI, supplied by Dalian Thrive Mining Co. Ltd, Dalian, China were used. To provide mechanical rigidity to the metallic foam channel, a stainless steel (multipurpose stainless-steel Type 304, McMaster-Carr, Princeton, NJ) bracket was fabricated to fasten 9.525 mm (0.375 in.) compression fittings at the inlet and outlet, but also six 0.813 mm (0.032 in.) compression fittings along the length of the channel for thermocouple probes and pressure taps. The bracket is 304.8 mm (12.0 in.) long with three sides removed to expose the metal foam. Stainless steel was chosen because of its low thermal conductivity, which would reduce any axial conduction through the skin. Appendix A shows a schematic of the metal bracket.

The foams sheets were cut with a square cross-section to the dimensions of 304.8 mm × 20 mm × 20 mm (L × W × H) by an electric-discharge machine (AD325L CNC Wire EDM, Sodick, Japan) in order to minimize any deformation and damage to the pore fibers. The foam samples are placed within the bracket and the inlet and outlet caps are welded on, Figure 2.1 (a). In the present study, a new method developed by the Center of Advanced Coating Technologies at the University of Toronto, is implemented to deposit well-adhered dense skins onto the surface of the metal foams regardless of the pore size [6]. The following is an outline of the procedure:

1. Before a thermal sprayed skin can be applied to the metal foam, the voids along the surface of the foam need to be filled. A paste is made by combining a thermoset resin (Acrodur 950 L, BASF, Germany) with metallic powder particles (Metco 42C stainless steel alloy powder, Sultzer Metco, USA). The composition of the paste should be approximately 70% powder to 30% resin by weight. For this these specific samples 71.51 g of powder to 20.73 g of resin were mixed to cover the surface area of a single channel.

2. The paste is then applied to the surface of the foam, penetrating 1 to 3 mm into the foam, Figure
2.1 (b).

3. The paste filled foam is then left to cure for 24 hours, after which the surfaces of the sample are lightly grit-blasted to expose as many foam struts as possible while roughening the surface for thermal spraying. Roughening the surface enhances the adhesion of the spray coating to the sample. At this point the exposed struts make up approximately 30% of the total surface area, providing a direct contact point between the skin and metal foam.

4. A stainless steel metallic skin (Metcoloy 2 alloy wire, Sultzer Metco, Westbury, NY) is deposited onto the nickel foams and a copper metallic skin (Metco Copper AW, Sultzer Metco, Westbury, NY) onto the copper foams using a ValuArc (Sulzer Metco Inc., Westbury, NY) electric wire arc spray process. The spray process parameters utilized for each material are listed in Table 2.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Gun</th>
<th>ValuArc</th>
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<tr>
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<td>Voltage (V)</td>
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<tr>
<td>Inlet Pressure (psi)</td>
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<tr>
<td>Air Flow Rate (SCFM)</td>
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</tr>
<tr>
<td>Skin Thickness (mm)</td>
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<td></td>
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</table>

(a)

<table>
<thead>
<tr>
<th>Parameter</th>
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<tr>
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<td>Voltage (V)</td>
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<td>Inlet Pressure (psi)</td>
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<tr>
<td>Air Flow Rate (SCFM)</td>
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</tr>
<tr>
<td>Skin Thickness (mm)</td>
<td>3.1</td>
<td></td>
</tr>
</tbody>
</table>

(b)

Table 2.1: Wire Arc thermal spray process parameters to deposit (a) stainless steel onto Ni foams and (b) copper onto Cu foams.

5. Once the skin has been deposited, Figure 2.1 (c), the entire sample is heat treated for approximately 2 hours at 400°C to relieve residual stresses and burn away any remnants of the resin used to fill surface pores. All that remains then is the residual powder portion which can be easily removed from the samples by some simple agitation.

Figure 2.2 shows an enlarged view of a foam strut, detailing how the metallic skin has bonded to the foam substrate by using the resin mixture as a preform.

The microstructures of thermally sprayed coatings are very complex and incorporate process-dependent defects such as globular pores and interlamellar pores, which in effect give the coating an inherent porosity. For the applications specific to this study, a porous skin is not wanted so that the working fluid is contained, \( m_{\text{in}} = m_{\text{out}} \). To test for any leakages through the thermal skin, the samples were submerged in water while passing air through the sample. Leaks are detected by any bubble formation along the surface of the skin, indicating that air is escaping. To correct any defects in the coating the paste was
CHAPTER 2. EXPERIMENTAL METHODS

(a) 40PPI Ni foam sample placed within the stainless steel bracket before surface preparation.

(b) Foam sample filled with resin mixture. Lightly grit-blasted to roughen the surface for thermal spraying.

(c) Completed metal foam channel with a stainless steel skin deposited.

Figure 2.1: Illustrative procedure to prepare metal foam samples for thermal spray coating.
CHAPTER 2. EXPERIMENTAL METHODS

Figure 2.2: SEM micrograph of the metallic skin deposited by wire arc spraying onto the Ni metal foam microstructure.

Figure 2.3: Cross-sectional geometry of a 10PPI and 40PPI Ni metal foam sample cross-section with stainless steel skin deposition used in the present study. The samples are 20mm wide, with approximately 1.2 mm thick skin.
re-applied to specific areas and the sample re-sprayed. One of the main objectives of this study is to examine the feasibility of using metal foams with a thermal sprayed skin for heat exchanger applications. It has been shown that thermal skins can be applied to metal foams with varying geometries and pore densities. The advantage of using such a process, compared to conventional designs are:

- The thermal skins act as thin walls containing the working fluid, thus the thermal resistivity between the outer wall and the metal foam core is reduced.

- The increase in surface area provided by the foam (struts) improves heat transfer, compared to conventional ribbed fin and single tube designs.

- Geometries of heat exchangers can be made to meet most industrial applications without any restrictions on shape and size. The potential for irregular shape heat exchangers introduce new applications for heat exchangers that were previously not possible.

- The increased bond between the thermal skin and the struts of the metal foam are greater then any brazing technique currently used.

2.3 Pressure Drop Apparatus

The experimental setup used to collect the pressure drop data is shown schematically in Figure 2.4 (a). The apparatus consists of the compressed air facility, the thermal sprayed metal foam channel and instrumentation. Compressed air provided by the laboratory is driven through a pressure regulator attached to a globe valve, which when adjusted, can control the amount of air released to the test section. Positioned upstream before the test sample the measurement of mass flow rate is done by an electronic gas mass flowmeter (Model FMA1842, Omega Company, Stamford, CT) for a flowrate range of 0.0 L/min to 100.0 L/min, accurate to within ±1.5% of full scale (FS). The air then enters the test sample via ductwork that is mated to the compression fitting inlet on the metal foam bracket. The direction of air flow is from left to right as in the Figure 2.6 and progresses downstream where it exits to the the surroundings. The small holes on the bottom of the bracket were used to attach two 0.813mm (0.032 inch) compression fittings at 25.4 mm and 279.4 mm from the inlet. These fittings act as pressure ports to attach the differential pressure transducer (Model PX160, Omega Company, Stamford, CT) for a differential pressure range of 0.000 psi to 1.000 psi with an accuracy of ±0.25% FS. A digital process panel meter (Model DP24-E, Omega Company, Stamford, CT) calibrated for a 1 to 6 VDC input, with an accuracy of ±1 LSD (least significant digit) was used to display the pressure reading. All reported
pressures were measured between these two ports, spanning a distance of 254.0 mm. By conservation of mass the incompressible fluid (air) must decelerate as it enters the channel from the fitting due to the larger channel entrance area, but then accelerates as the effective cross-sectional area of the channel is reduced by the physical presence of the foam. The opposite occurs when the fluid leaves the foam, the fluid velocity must decrease in order to compensate for the increase in the effective cross-sectional area of the channel and then accelerate as it exits the fitting. Locating the pressure ports 25.4 mm from the entrance and exit of the sample reduces the static pressure-altering effects of the fluid's acceleration and deceleration. The remaining holes on the bottom of the bracket were sealed with closed-end fittings.

During a typical experimental run, the metal foam channel was setup as described above. The setup was first tested for leaks to ensure that the ductwork and fittings were in place. This was done by closing the outlet on the right in Figure 2.6 and running air. Air leaks were detected by applying a thin soap film around the fittings and ductwork and observing if any bubble formation occurred. The leaks around the fittings were re-tightened with Teflon tape around the threads. Once the leaks were sealed and the experimental system was deemed suitable the pressure drop data was taken. Air flow rate through the channel was set at a desired value by adjusting the globe valve and recording the given pressure drop. The pressure drop was measured for a flowrate range of 5.0 L/min to 90.0 L/min, and repeated several times to obtain sufficient data to perform the intended analysis. The pressure drop experiments were conducted from the high-end to the low-end of the flowrate range. To check for hysteresis, selected experiments were performed from the low-end of the flowrate range to the high-end. No hysteresis was observed in these experiments when they were compared to experiments taken by varying the flowrate from high to low. Therefore all the data used during calculations were taken by varying the flow rate from high to low. The temperature of the air during the pressure drop experiments did not vary more than $23.0 \pm 2.0^\circ C$ to consider a change in physical properties within the calculations. Experiments were performed for both the nickel and copper channels of varying pore densities.

2.4 Heat Transfer Apparatus

The experimental setup consists of the air facility, the thermal sprayed metal foam channel and instrumentation. A schematic representation of the experimental setup is shown in Figure 2.4 (a). Compressed air provided by the laboratory is driven through a pressure regulator attached to a globe valve, which when adjusted, can control the amount of air released to the test section. Positioned upstream before the test sample the measurement of mass flow rate is done by an electronic gas mass flow-meter (Model
FMA1842, Omega Company, Stamford, CT) for a flowrate range of 0.0 L/min to 100.0 L/min, accurate to within ±1.5% of full scale (FS). The air then enters the test sample via ductwork to a mated inlet using a compression fitting. The length of the test sample in the flow direction is held fixed at 304.8 mm (12.0 in.). The downstream end of the sample is then mated to further ductwork, which is open to the surroundings. The direction of air flow is from left to right as in Figure 2.6 and progresses downstream where it exits to the the surroundings.

The small holes on the bottom of the bracket were used to attach six 0.813 mm (0.032 in.) compression fittings axially distributed at 25.4, 76.2, 127.0, 177.8, 228.6 and 279.2 mm from the entrance along the flow direction. At each of the holes the foam was bored-out to create a void such that the probe can be inserted into the sample and measure the bulk temperature of the flowing air at each axial position. In each fitting a 152.4 mm (6 in.) Type-K (chromel-alumel) thermocouple probe with a 304 SS sheath of 0.813-mm-diameter (Model T36-CASS-032-G-6, Omega Company, Stamford, CT) was inserted until the probe tip was approximately at the mid-way height of the sample. In addition six more Type-K thermocouples, with a 0.25-mm-diameter junction were fixed to the top surface of the test sample to measure the local wall temperature of the deposited skin at the same axial positions as the probes. The thermocouple junctions were pressed tightly against the outer wall and fixed in place using high-temperature cement (CC High Temperature, Omega Company, Stamford, CT).

Heating of the test sample on all of its surfaces was achieved by tightly wrapping a resistance wire heating tape, encased in fiberglass insulation (Model FGH051-060, Omega Company, Stamford, CT). This heater has dimensions of 12.7 mm × 1828.8 mm (0.5 in. × 72 in.) approximately the same size as the test sample. It has an electrical resistance of 46.0 Ω and produces a maximum power output of 313 W at 120 V. Power to the heater is provided by a variable transformer (power supply) that can be used to vary the constant heat flux condition from 107 W/m² to 6846 W/m² (10VAC to 80VAC) at the outer walls. To prevent any extraneous heat loss to the surroundings the entire test sample is covered using 38.1 mm (1.5 in.) thick fiberglass insulation with an aluminized outer surface, having an average thermal conductivity of \( k_i = 0.038 \) W/mK (Micro-Flex, Johns Manville Corporation, Denver, CO).

To validate the heat loss from the experiment, two more Type-K, with a 0.25-mm-diameter junction thermocouples were fixed to the surfaces of the insulation. The heat losses through the the surfaces of the insulation where calculated using the temperature readings from these thermocouples, \( T_{loss} \). This procedure has been shown to accurately represent the lateral heat losses in this type of experimental arrangement [16, 79]. To account for heat losses through the insulation, the heat entering the test section
Figure 2.4: Schematic of the constant heat-flux experimental apparatus (a) showing the instrumentation and configuration. (b) Detailed view of the test section.
Figure 2.5: Experimental apparatus with instrumentation.

Figure 2.6: Detailed view of test section, showing the placement of thermocouples and tape heater (the insulation has been removed for clarity). The flow direction of air is indicated from inlet (left) to outlet (right).
is calculated by

\[ Q_H = \frac{V^2}{R_H} - k_i A_i \left( \frac{\Delta T_{\text{loss}}}{b} \right) \]  \tag{2.1}

For all further calculations presented in this study, the amount of heat provided to the test section is based on Equation (2.1), where heat losses are accounted for.

The thermocouple voltages were recorded by a National Instruments data acquisition (DAQ) unit and transmitted directly to a personal computer. The DAQ system consists of the following modules:

- \( \times 1 \) SCXI 1000 - 4-slot 120V AC-powered chassis (Model 776570-01).
- \( \times 1 \) SCXI 1600 - control module with 16-bit resolution, capable of 200 kS/s sampling rate and connects to a PC via USB 2.0 connector (Model 776572-1600).
- \( \times 2 \) SCXI 1112 - 8 channel thermocouple input module, with an accuracy of \( \pm 1.2^\circ \text{C} \) and a repeatability of \( \pm 0.4^\circ \text{C} \) (Model 776572-12).

In total 14 thermocouples were used simultaneously, the temperature readings of the thermocouples are recorded in real-time using a personal computer equipped with a graphical programming environment to record the temperature readings (LabVIEW Signal Express v.3.0, National Instruments Corporation, Austin, TX).

The method to compute the measured uncertainty associated with digital data acquisition systems can be readily described in Wheeler and Ganji [49]. In the case of uncertainty in the temperature measurements, the procedure followed is the root-sum-squares (RSS) method. In this method, the bias and precision elemental errors within the instrumentation are combined to determine a realistic estimate for uncertainty. The uncertainty associated with the data acquisition system is

\[ \delta e_{DAQ} = \sqrt{\epsilon_{\text{bias}}^2 + \epsilon_{\text{precision}}^2} \]

\[ \delta e_{DAQ} = \sqrt{(1.2^\circ \text{C})^2 + (0.4^\circ \text{C})^2} \]

\[ \delta e_{DAQ} = \pm 1.3^\circ \text{C} \]

Based on the manufacturing specifications, the thermocouples have a standard limit of error of \( \pm 2.2^\circ \text{C} \). The resolution of LabView was set to equal \( \pm 0.01^\circ \text{C} \), thus the zero-order uncertainty is equal to
±0.005°C. Hence the uncertainty associated to the thermocouples is

\[ \delta e_{\text{thermocouple}} = \sqrt{e_{\text{bias}}^2 + e_{\text{precision}}^2} \]

\[ \delta e_{\text{thermocouple}} = \sqrt{(2.2^\circ C)^2 + (0.005^\circ C)^2} \]

\[ \delta e_{\text{thermocouple}} = \pm 2.2^\circ C \]

The overall uncertainty associated with the temperature measurement is the combined elemental error of the data acquisition system and thermocouple

\[ \delta T = \sqrt{e_{\text{DAQ}}^2 + e_{\text{thermocouple}}^2} \]

\[ \delta T = \sqrt{(2.2^\circ C)^2 + (1.3^\circ C)^2} \]

\[ \delta T = \pm 2.6^\circ C \]
Chapter 3

Metal Foams

3.1 Introduction to Metal Foams

Metallic foam structures are a new class of materials that have received much attention recently due to their attractive physical, mechanical, thermal, electrical and acoustic properties. At present metal foams are incompletely characterized and the processes used to make them are imperfectly controlled, resulting in some variability in properties. But even the present generation of metal foams have substantial potential and processing control is improving rapidly. Their low weight and high specific strength and stiffness make them well suited for structural components, energy absorption and vibration dampening systems. Their low density, interconnected porosity, high permeability and the ability to make them out of high thermal conductive metals, make them attractive for a wide range of industrial applications involving fluid flow and heat transfer phenomena, such as heat exchangers, evaporators, burners and separators [50]. The use of open-cell metal foams in numerous technological applications has increased rapidly in recent years. Through improved manufacturing processes, metallic foams can be produced in a variety of materials and specifications at a relatively low cost. The use of metal foams in heat exchanger applications is fairly recent, with commercially viable technology still under development and mainly a subject of fundamental and applied research [7].

3.1.1 Manufacturing and Processing

The first reported method of producing a metal foam occurred in 1948, when Benjamin Sosnick filed a patent on the “Process for making a foam-like mass of metal” [8]. His method was limited to a few
metal alloys, that contained different phases with widely different melting and boiling points. This process took advantage of the fact that when heating a multiphase alloy, one phase can melt while the other boils independently. By heating that alloy under pressure, he could ensure that the gaseous phase would not escape the molten metal phase. Upon releasing the pressure, the more volatile phase would subsequently expand and vaporize within the molten metal substrate, leaving a solid mass full of closed pores [9]. The resulting foam structures were unfortunately very unstable and had a tendency to collapse during cooling. But because only a very small irregular amount could be produced at a relatively expensive cost, Sosnick's method never met true commercial success.

Some time later, in the late 1950s, the first open-cell metal foams were produced. This involved pouring molten aluminum into a preform of compacted rock salt grains, and then dissolving out the salt to leave open pores. This produced a more reliable foam structure than Sosnick's method, but at the time it was still seen as a curiosity rather than a serious engineering material [9]. Suggested applications included shock-proof material to house the fragile electric circuitry in missiles and rockets (which became unnecessary as microchips became more robust), high temperature metal filters, and large-area metal cathodes [9].

Over the last 60 years, numerous developments have been made to manufacture porous metallic structures. Some methods are similar to techniques used for foaming aqueous or polymer liquids, whereas others are specially designed by taking advantage of the characteristic properties of metals such as their sintering activity or the fact that they can be electrically deposited. The general manufacturing concept of a foam involves the controlled combination of two distinct phases, which usually requires a gas to be dispersed within a liquid without dissolving the gas. Currently, there are nine distinct processes that can be used to manufacture metallic foams, with five of them being implemented at a commercial scale for the production of metal foams. The manufacturing of cellular metal foams can be characterized into four main classes based on their initial metal state [10].

A comprehensive overview of all the metal foam manufacturing processes can be found within [10, 11]. The nickel and copper foam sheets utilized in this study have been manufactured through a deposition process, given that they have distinctive hollow ligaments. The process begins by depositing a metal onto a disposable open-cell polymer preform (usually polyurethane). Once the polymer has been fully coated with the required metal, the structure can be sintered to remove the polymer. Depending on foam sheet thickness and cell size the deposition of the metal onto the preforms can be done by one of two techniques.
CHAPTER 3. METAL FOAMS

Chemical Vapour Deposition (CVD)

The open-cell polymer foam serves as a template upon which metal is deposited in a vapour state, the final geometry of the foam is highly depended on the quality of this initial preform. For the manufacturing of nickel foam, a decomposition reaction of nickel carbonyl gas $\text{Ni(CO)}_4$ is used. Figure 3.2 schematically illustrates one approach in which an open-cell polymer is placed in a CVD vacuum chamber and exposing it to a stream of nickel carbonyl gas.

1. The open-cell polymer preform is coated with a strong infrared-absorbing material, usually with carbon black or other absorbing pigments. In order to avoid spontaneous decomposition of the carbonyl gas away from the surface of the preform, the coating is chosen to strongly absorb infrared radiation at a frequency which is only weakly absorbed by the carbonyl gas.

2. The polymer preform is placed in a CVD reactor and nickel carbonyl is introduced. Using infrared radiation heating, the surface of the polymer preform is heated to about 120°C. As the gas comes in contact with the heated surface it decomposes to carbon monoxide and nickel, covering the preform in a layer of nickel several tens of micrometers thick [11].

3. The metal-coated polymer is then removed from the CVD reactor and sintered at a high enough temperature in air that it will decompose the polymer preform.

4. A subsequent sintering step can also be used to thicken the ligaments.

Nickel carbonyl is a highly toxic gas and requires costly environmental controls before commercial manufacturing of nickel foams can begin. Due to regulations restricting the use of nickel carbonyl and
Figure 3.2: Schematic illustration of the CVD process used to create open-cell nickel foams [11]
prohibitively expensive costs to implement it within an industrial process, the production of nickel foams using a CVD process has become very limited.

**Electro-Deposition**

This technique requires that the polymer preform be electrically conductive. This can be achieved by either dipping the polymer foam into a graphite or carbon based slurry, immersing the polymer foam into an electroless plating solution, or by coating the polymer foam with a thin conductive layer by cathode sputtering \[10\]. Once the preform is electrically charged, the preform is electroplated within a solution of metals ions in an electrolyte. The preferred metals are nickel, nickel-chromium alloys and copper. The metal-polymer composite is then heat treated to remove the polymer. Metal foams produced through this method have shown to experience greater mechanical strength and higher electrical conductivity, due to the smoother surfaces that are obtainable.

After the process is complete the result is open cell metal foam which consists of small filaments (struts) that are continuously connected in an open-celled structure. Both deposition techniques result in a open-cell metal network with distinctive hollow struts. Current industrial processes are capable of fabricating various grades of metal foams, ranging from 5 to 60PPI pore sizes, with very low relative densities (0.02-0.05) and porosities as high as 99%. Due to the relative difficulty of depositing metal alloys through a CVD or electrodeposition process, the methods are restricted to foaming pure elements, such as nickel, copper and titanium \[10\].

### 3.2 Characterization of Metal Foams

Porous mediums in general can be broadly divided into two categories, natural and engineered. In a natural porous medium the distribution of pores with respect to shape and size is irregular and non-uniform, such examples include sand, soil, the pulmonary alveolus of the human lung and trabecular bone. Engineered porous media are further sub-divided into two categories (1) fibrous media, such as insulation, metallic foam and ceramic materials, and (2) packed beds such as packed spheres and sintered particles \[9\]. The properties of open-cell metal foams depends primarily on the material from which they are made of and their geometric structure. Although this influence is not perfectly understood at present, it is a topic of intense current study. Better understanding of such properties can lead to greater process control and to a greater understanding of their fluid flow and heat transfer characteristics.
An open-cell metal foam consists of a solid matrix with interconnected voids (pores), typical metal foam structures are shown in Figure 3.3 with varying pore densities. The interconnection of the voids allows for one or more fluids to flow through the material by means of a tortuous path. In the present study the porous medium is saturated with a single-phase fluid. The individual pores of the foam are often represented as a unit cell, having an approximate shape of a dodecahedron with 12-14 pentagonal or hexagonal faces \([3]\). The edges of the dodecahedron form the fibers (struts) that interconnect one unit cell to the other. It is quite common due to the deposition process that an accumulation of material occurs at the fiber intersections. There are several key geometric and physical parameters used to characterize the physical structure of a porous medium. The \textit{porosity} \(\varepsilon\) is defined as

\[
\varepsilon = \frac{\text{void volume contained in porous medium sample}}{\text{total volume of porous medium sample}} \tag{3.1}
\]

In engineered porous media, fibrous materials usually have a constant porosity. The \textit{pore density} is defined by a standard industrial specification as pores per inch (PPI), which measures the number of pores per linear inch. The \textit{pore size} \(d_p\) is defined by the equivalent diameter of one of the faces of the dodecahedron unit cell. At the microscale level, the pore is nonuniform in its size and distribution throughout the bulk of the matrix, thus they are usually presented as a statistical average of the local values.

The \textit{fiber diameter} \(d_f\) is the equivalent diameter of the fiber cross-section. In the case of hollow fibers, a hollowness ratio \(r_f\) is defined as

\[
r_f = \frac{d_i}{d_f} \tag{3.2}
\]

where \(d_i\) and \(d_f\) are the inner and outer fiber diameters. The porosity and the pore size are two parameters independent of each other \([16]\). However by reducing the porosity (i.e. the amount of solid material in the metal foam increases) the fiber diameter becomes larger and hence stronger, eventually improving the strength of the overall foam structure. Subsequently, it has been shown that the fiber
CHAPTER 3. METAL FOAMS

Figure 3.4: Comparison of a dodecahedral unit cell representation with an actual pore unit of an open-cell metal foam.

Figure 3.5: Fiber cross-sectional geometry as a function of porosity.

Figure 3.6: SEM image of a fiber cross-section from a metal foam used in the present study. The image captures the distinctive hollow feature and inner concave shape, $\varepsilon = 0.94$. 
cross-section morphs from an inner concave triangle to a circular shape as a function of porosity \[14, 16\]. The relative density $\rho_{rel}$ is defined as the ratio of the foam density to the density of the solid material as in Equation (3.3). The ratio of solid mass to the foam mass is unity, since the mass of the foam is entirely comprised of the solid. In other words the relative density represents the amount of solid present in a given volume of foam.

$$\rho_{rel} = \frac{\rho_{foam}}{\rho_{solid}} = \frac{m_{foam}}{m_{solid}} \frac{V_{solid}}{V_{foam}}$$  \hspace{1cm} (3.3)

From this, the conventional definition of porosity can be redefined as

$$\varepsilon = \frac{V_{void}}{V_{foam}} = \frac{V_{foam} - V_{solid}}{V_{foam}} = 1 - \rho_{rel}$$  \hspace{1cm} (3.4)

In the case of metal foams with hollow fibers, the definition of porosity in Equation (3.4) can be misleading, since it neglects the mass absent from the void of the fibers. Hence, it is necessary to consider the porosity that the fluid 'sees' and derive a method that accounts for the void mass of the hollow fibers. It should be noted that not all of the above parameters are independent of each other.

### 3.2.1 Foam Geometric Measurements

Accurate values of the cell parameters are crucial to the numerical calculations to be presented later. In this section, pore diameter, fiber diameter, the hollowness ratio along with the porosity and relative density will be carefully measured. The aim here is to use the measured values to validate the applicability of an idealized model of the foam structure and determine if it is an adequate model to capture geometric parameters needed to characterize fluid flow and heat transfer. Measurements were performed on 8 epoxy prepared and 8 non-epoxy prepared samples, refer to Appendix B for description of samples.

#### Sample Preparation and Image Capturing

In order to ascertain the structural characteristics of the metal foams, a procedure for determining the different geometrical parameters by image analysis was implemented. The following is a description of the procedure used to prepare the metal foam specimens, followed by a description of the image capturing process.

1. **Sampling/Sectioning**

   To avoid cell damage the nickel and copper foams are sectioned using an abrasive cut-off machine (ME-B221-1201 Brilliant 221, Clemex Technologies Inc., Longueuil, QC) with a metal bonded
diamond wavering blade. Samples are cut at a rate of 1.5 mm/min with a blade speed of 3400 rpm to an approximate size of 20 mm × 25 mm × 20 mm.

2. Mounting

Mounting protects the sample from destructive attack and provides edge protection during the grinding and polishing procedures. Placed in a castable mounting cup the metal foam specimens are encapsulated using a low viscosity opaque castable epoxy (cold mounting) with a mix ratio of 5:2 by weight (EpoThin, Buehler, Whitby, ON). The specimen is then placed in a Vacuum Impregnation System (No. 20-1382-160, MetLab Corporation, Niagara Falls, NY) for 5-10 minutes to infiltrate the open pores of the foam and left to cure for 48 hours.

3. Grinding/Polishing

The subsequent stages are intended to remove the initial depth of deformation from cutting but also to create a clean and undistorted surface appropriate for image analysis. The damage produced by each preparation step must be removed by the subsequent step, leaving considerably less surface damage after each step. Table 3.1 lists the abrasive method used to prepare the specimens.

Grinding and polishing is achieved using an automated grinding wheel (DPS 2000, Imptech, Seoul, KR).

<table>
<thead>
<tr>
<th>Stage</th>
<th>Surface</th>
<th>Lubricant</th>
<th>Abrasive Type/Size</th>
<th>Time (min)</th>
<th>Load (psi)</th>
<th>Base Speed (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Planar Grinding</td>
<td>SiC Paper</td>
<td>Water</td>
<td>180 grit</td>
<td>3</td>
<td>6</td>
<td>150</td>
</tr>
<tr>
<td>Fine Grinding</td>
<td>SiC Paper</td>
<td>Water</td>
<td>240 grit</td>
<td>3</td>
<td>6</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>SiC Paper</td>
<td>Water</td>
<td>320 grit</td>
<td>3</td>
<td>6</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>SiC Paper</td>
<td>Water</td>
<td>400 grit</td>
<td>3</td>
<td>6</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>SiC Paper</td>
<td>Water</td>
<td>600 grit</td>
<td>3</td>
<td>6</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>SiC Paper</td>
<td>Water</td>
<td>800 grit</td>
<td>3</td>
<td>6</td>
<td>150</td>
</tr>
<tr>
<td>Rough Polishing</td>
<td>Polyester</td>
<td>-</td>
<td>1 μm PD</td>
<td>6</td>
<td>1</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>Cloth</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**PD** - polycrystalline diamond, *SiC - silicone carbide

Table 3.2: Description of grinding and polishing parameters used to prepare the epoxy mounted samples.
4. Image Capturing

Depending on the magnification and the intended analysis, images were captured by two different means. For high magnifications (1000×) requiring detailed views of the microstructure, especially for the small pore size specimens, a scanning electron microscope (SEM) (S-570, Hitachi High-Techologies Canada Incorporated, Toronto, ON) was used on the non-epoxy covered samples. For lower magnifications (5×) an optical microscope (MM-100 Measurescope, Nikon Corporation, Tokyo, JPN) was used with a 3.43 mega-pixel digital camera (Coolpix 990, Nikon Corporation, Tokyo, JPN). A scale micrometer was used to calibrate the microscope with an accuracy of ±1µm at any given magnification. In order to gain a greater field of view than what is possible with the lowest magnification, several images of a given sample were stitched together using an image editing software (CorelDRAW Graphics Suite X5, Corel Corporation, Ottawa, ON) to create a panoramic image of the overall sample. Using the microscope’s stage manipulator, images of the sample were taken at increments of 0.01 mm in a given horizontal direction, and then repeated for vertical increments of 0.5 mm. Once all the required images are captured, measurements of the cell and foam parameters are taken using a calibrated image analysis software (ImageJ v.1.43, National Institutes of Health, Bethesda, MD).

3.2.2 Results of Measured Microstructure

Fiber Diameter and Hollowness Ratio

Two dimensional images were extracted from the optical microscope using the digital camera at magnifications of 10× and 40× to measure cross-sectional geometries of the hollow fibers, Figure 3.8 (a). The images correspond to the plane shown in Figure 3.7, perpendicular to the plane cut by the diamond saw. For each sample five fiber and hollow hole diameters are measured, from which an average value and
standard deviation can be found. The images are first filtered using a threshold operation to convert it into a binary image, Figure 3.8 (b), where unwanted features are removed leaving only the geometry of the fiber. The area and perimeter are then calculated by Image J based on the edges of the desired feature, Figure 3.8 (c). The fiber and hollow hole diameters are then determined using the hydraulic diameter definition for each object, \( j \).

\[
d_j = \frac{4A_j}{P_j}
\]

(3.5)

From the analysis, an average fiber diameter \( \bar{d}_f \) and hollow hole diameter \( \bar{d}_o \) can be found with their respective uncertainties in Table 3.3. The hollowness ratio of these samples ranges from 30 to 62 per cent, indicating that the conventional definition of porosity may lead to over prediction. A summary of the measurements taken for fiber and hollow hole diameter for each sample is given in Appendix B.

<table>
<thead>
<tr>
<th>Sample</th>
<th>PPI</th>
<th>( \bar{d}_f ) (( \mu \text{m} ))</th>
<th>( \sigma_{\bar{d}_f} ) (( \pm \mu \text{m} ))</th>
<th>( \sigma_{\bar{d}_f} ) (%)</th>
<th>( \bar{d}_o ) (( \mu \text{m} ))</th>
<th>( \sigma_{\bar{d}_o} ) (( \pm \mu \text{m} ))</th>
<th>( \sigma_{\bar{d}_o} ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni</td>
<td>10</td>
<td>332</td>
<td>30</td>
<td>9.1</td>
<td>151</td>
<td>16</td>
<td>10.8</td>
</tr>
<tr>
<td>Cu</td>
<td>10</td>
<td>319</td>
<td>43</td>
<td>13.4</td>
<td>159</td>
<td>21</td>
<td>13.0</td>
</tr>
<tr>
<td>Ni</td>
<td>40</td>
<td>95</td>
<td>16</td>
<td>17.3</td>
<td>42</td>
<td>7</td>
<td>15.4</td>
</tr>
<tr>
<td>Cu</td>
<td>40</td>
<td>102</td>
<td>21</td>
<td>21.0</td>
<td>46</td>
<td>4</td>
<td>7.9</td>
</tr>
</tbody>
</table>

Table 3.3: Average fiber and hollow hole diameters based on image analysis of 10PPI and 40PPI metal foam samples

**Pore Diameter**

The nominal pore size of the metal foam can be calculated directly from the product specification based on PPI. The foams examined here are 10PPI and 40PPI corresponding to nominal pore sizes of 2.54 mm and 0.64 mm respectively. To verify the manufacturer’s claim the pores are measured by an image analysis method, Figure 3.9 (a), where the foam pores are captured by an SEM at a beam voltage of 20 kV. For each sample the area and perimeter of five pores are measured. The equivalent diameter is again computed based on the hydraulic diameter definition given in Equation (3.5) and presented in Table 3.4 with its uncertainty. A summary of the measurements taken for pore diameter from each sample is given in Appendix B.
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Figure 3.8: Image progression to extract cross-sectional geometries of a Ni foam sample at $10 \times$ magnification. (a) Cropped optical microscope image. (b) Binary image exposing the geometry of the fiber. (c) Image J analysis of the area and perimeter for the specified features.

Figure 3.9: SEM image progression to extract cross-sectional geometries of a Ni foam sample at $55 \times$ magnification. (a) SEM image with defined hydraulic pore diameter. (b) Image J analysis of the area and perimeter obtained from a binary image.
Table 3.4: Average pore diameters based on image analysis of 10PPI and 40PPI metal foam samples

<table>
<thead>
<tr>
<th>Sample</th>
<th>PPI</th>
<th>( d_p ) (mm)</th>
<th>( \sigma_{d_p} ) (( \pm ) mm)</th>
<th>( \sigma_{d_p} ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni</td>
<td>10</td>
<td>2.771</td>
<td>0.443</td>
<td>16.0</td>
</tr>
<tr>
<td>Cu</td>
<td>10</td>
<td>2.868</td>
<td>0.442</td>
<td>15.4</td>
</tr>
<tr>
<td>Ni</td>
<td>40</td>
<td>0.742</td>
<td>0.090</td>
<td>12.1</td>
</tr>
<tr>
<td>Cu</td>
<td>40</td>
<td>0.631</td>
<td>0.074</td>
<td>11.8</td>
</tr>
</tbody>
</table>

It should be noted that comparing the nominal values claimed by the manufacturer and the values measured by image analysis are in relatively good agreement.

**Porosity and Relative Density**

The porosities of the foam samples were calculated based on Equation (3.4). The dimensions of the sample were first measured using a digital (Model 500-421, Mitutoyo Corporation, Mississauga, ON) caliper with an accuracy of \( \pm 0.005 \) mm, from which an approximation of the samples volume \( V_{sample} \) can be determined. The samples are then weighted with a digital scale (ExplorerPro EP214, Ohaus, Toronto, ON) with an accuracy of \( \pm 0.00005 \) g and its mass \( m_{sample} \) is obtained. Dividing the mass of the sample with the density of the foam material, the approximate volume of solid \( V_{solid} \) within the sample is determined. Porosity can then be calculated with use of Equation (3.6).

\[
\varepsilon = \frac{V_{sample} - V_{solid}}{V_{sample}}
\]  

(3.6)

The porosities have been averaged over 8 samples and presented with their uncertainties in Table 3.5. A summary of the measurements taken for porosity from each sample is given in Appendix B. Because of the presence of the hollow fibers the porosity measured with the use of Equation (3.6) leads to overpredicted values. An attempt to confirm the actual porosity is made by considering the area fractions of each phase present in a given plane. To determine these values an effective porosity and relative density are computed by image analysis, where the area fraction of the void phase and solid phase, assuming unit length, are defined as

\[
\varepsilon_{eff} = \frac{A_{void}}{A_{total}}
\]  

(3.7)

\[
\rho_{eff} = \frac{A_{solid}}{A_{total}} = 1 - \varepsilon_{eff}
\]  

(3.8)

Using Image J, a given plane of the the foams cross-section, Figure 3.10 (a), is converted to a binary
Figure 3.10: Image progression to extract the porosity of a Cu foam sample at a given plane. (a) Stitched microscopic images. (b) Binary image exposing the area fraction of fibers to the void.
image, Figure 3.10 (b), where the interior boundaries of the fibers are considered homogeneous in order to neglect the presence of the hollow fibers. Correlating the the number of black pixels to white pixels, Image J is able to compute the area fraction of each phase. The planar images are obtained from the stitched images captured from the epoxy prepared samples as described in Section 3.2.1, where the measured porosities are averaged over two samples. Table 3.5 compares the porosity results by both the aforementioned methods.

<table>
<thead>
<tr>
<th>Sample</th>
<th>PPI</th>
<th>$\varepsilon$</th>
<th>$\sigma_{\varepsilon}$ (%)</th>
<th>$\rho_{rel}$</th>
<th>$\sigma_{\rho_{rel}}$ (%)</th>
<th>$\varepsilon_{eff}$</th>
<th>$\rho_{eff}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni</td>
<td>10</td>
<td>0.985</td>
<td>0.241</td>
<td>0.015</td>
<td>0.241</td>
<td>0.9404</td>
<td>0.0596</td>
</tr>
<tr>
<td>Cu</td>
<td>10</td>
<td>0.982</td>
<td>0.378</td>
<td>0.018</td>
<td>0.378</td>
<td>0.9437</td>
<td>0.0563</td>
</tr>
<tr>
<td>Ni</td>
<td>40</td>
<td>0.972</td>
<td>0.651</td>
<td>0.028</td>
<td>0.651</td>
<td>0.9416</td>
<td>0.0583</td>
</tr>
<tr>
<td>Cu</td>
<td>40</td>
<td>0.963</td>
<td>0.401</td>
<td>0.037</td>
<td>0.401</td>
<td>0.9377</td>
<td>0.0623</td>
</tr>
</tbody>
</table>

Table 3.5: Comparison of porosity for the metal foam samples by means of weighing and by image analysis.

It is evident from the results that the existence of the hollow fibers effects the measurement of porosity.

### 3.2.3 Ideal Structure

Since the metal foam structure is very complex, several investigations have been attempted to idealize the structure in order to determine analytical relations for hydraulic and heat transfer characteristics. Boomsma and Poulilakos [17, 18] proposed a 14-sided tetralaidecahedron unit cell claiming it to be the optimal packing cell structure to represent metal foams. A similar structure was modified by Schmiere and Razani [14] referred to as the TetraK model where the cubic nodes were replaced by spherical nodes. Both models were implemented into comprehensive computational investigations, including a finite-element analysis and a computational fluid dynamic analysis, where a non-destructive method is employed using X-ray microtopography to render a three-dimensional computer tomography of the foam morphology [19, 20]. For the forthcoming analysis such an elaborate model is not required, but rather a simplified structure will be used that has been widely implemented by a variety of investigators including Calmidi [16], Lu et al. [21] and Gani et al. [22]. The open-cell metal foam is approximated as a cubical array of cross-cylinders oriented in three perpendicular directions, as shown in Figure 3.11. The pore size $d_p$ and fiber diameter $d_f$ are shown in Figure 3.11.
Although, both structures are open-cell, there are two important differences between the original structure in Figure 3.4 and the representation in Figure 3.11. First, the shape of the cell is dodecahedral in the former and is cubical in the latter. Second, the shape of the cross-section of the fibers varies with porosity in the former and is considered circular and constant in the latter. In order to derive an equivalence between the two structures, $d_p$ and $d_f$ must be related to their respective counterparts.

Considering the model proposed in Figure 3.11, the overall cell volume $V_{cell}$ can be calculated as

$$V_{cell} = d_p^3$$  \hspace{1cm} (3.9)

The total volume of the solid cylinders $V_{solid}$ can be expressed as a function of the foam void fraction

$$V_{solid} = (1 - \varepsilon) V_{cell} = (1 - \varepsilon) d_p^3$$  \hspace{1cm} (3.10)

Alternatively, $V_{solid}$ can also be calculated as the overall volume of the cylinders included in the unit cell, considering that each strut is shared among four unit cells

$$V_{solid} = \frac{3\pi}{4} d_f^2 d_p$$  \hspace{1cm} (3.11)
By combining Equation (3.11) and (3.12) one is able to obtain an expression that relates $d_p$ and $d_f$ to $\varepsilon$

$$\frac{d_f}{d_p} = 2\sqrt{\frac{(1-\varepsilon)}{3\pi}}$$

(3.12)

Furthermore, if Equations (3.10) and (3.12) are combined the porosity of the model $\varepsilon_{\text{model}}$ can be expressed as

$$\varepsilon_{\text{model}} = 1 - \frac{V_{\text{solid}}}{V_{\text{cell}}} = 1 - \frac{3\pi}{4} \left(\frac{d_f}{d_p}\right)^2$$

(3.13)

The surface area density $\alpha_{sf}$, is the interfacial surface area that is in contact between the fluid and solid phase. Its derivation is based on the ratio of the total surface area of the 12 cylinders to the volume of the unit cell

$$\alpha_{sf} = 3\pi \frac{d_f}{d_p}^2$$

(3.14)

The surface area density is inversely proportional to the square of the pore diameter and proportional to the cell fiber diameter. Using the measured geometrical parameters found in Section 3.2.2, the next step is to determine if the proposed cubic model can give an acceptable approximation for the structure of the metal foam. In Table 3.5, a summary of the geometrical parameters of the foam are given and compared to their calculated counterparts based on the cubic model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Ni</th>
<th>Cu</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal Pore Size, (PPI)</td>
<td>10</td>
<td>40</td>
</tr>
<tr>
<td>Nominal Pore Diameter, (mm)</td>
<td>2.54</td>
<td>0.64</td>
</tr>
<tr>
<td>Measured Pore Diameter, $d_p$ (mm)</td>
<td>2.771</td>
<td>0.742</td>
</tr>
<tr>
<td>Measured Fiber Diameter, $d_f$ (mm)</td>
<td>0.332</td>
<td>0.095</td>
</tr>
<tr>
<td>Measured $d_f / d_p$</td>
<td>0.119</td>
<td>0.128</td>
</tr>
<tr>
<td>Modeled $d_f / d_p$</td>
<td>0.159</td>
<td>0.157</td>
</tr>
<tr>
<td>Porosity, $\varepsilon_{\text{eff}}$</td>
<td>0.940</td>
<td>0.942</td>
</tr>
<tr>
<td>Modeled Porosity, $\varepsilon_{\text{model}}$</td>
<td>0.966</td>
<td>0.961</td>
</tr>
<tr>
<td>Uncertainty in porosity, $\delta \varepsilon$ (%)</td>
<td>1.28</td>
<td>1.69</td>
</tr>
<tr>
<td>Surface area density, $\alpha_{sf}$ (mm$^2$/mm$^3$)</td>
<td>0.407</td>
<td>1.626</td>
</tr>
<tr>
<td>Uncertainty in surface area, $\delta \alpha_{sf}$ (%)</td>
<td>33.39</td>
<td>29.75</td>
</tr>
</tbody>
</table>

Table 3.6: Summary of measured and modeled geometric parameters for metal foams used in the present study.

Using Equation (3.12) the ratio of fiber diameter to pore diameter is calculated with the use of the effective porosity. Comparing the measured and modeled fiber and pore diameter ratios it can be seen that the derived model over predicts the measured ratio by a maximum deviation of 39% (for Cu 10PPI). This deviation can be attributed to the models assumption of a constant circular cross-section. As discussed before, when considering metal foams with porosities greater then 0.85 it has been established
that the shape of the fiber alters as a function of porosity. Thus to account for this discrepancy a shape-factor should be introduced to better predict the ratio. However, comparing the actual porosity to that of the model, relatively good results have been achieved, with only a 3% maximum deviation. Based on this analysis and the assumption that the cubic model can predict the necessary parameters with acceptable accuracy, the computed surface area will be used in the study of heat convection in metal foams. From a design perspective the derivation of the cubic model equations is a means of predicting geometric parameters of metal foams with reliable certainty without the need for a comprehensive image analysis, which can be costly and time consuming.

### 3.2.4 Uncertainty Analysis

The uncertainty in the geometric parameters are given by the general formula for error propagation described by Taylor [49], while assuming that the bias error limit is entirely based on the accuracy of the experimental instrumentation (i.e microscope and SEM) used to measure the respective quantities. Since the pore and fiber diameters are calculated based on the hydraulic diameter definition they can be treated as functions of the cross-sectional area $A_i$ and perimeter $P_i$. Similarly, the measured porosity from image analysis can be treated as a function of the void cross-sectional area $A_{\text{void}}$ and the total cross-sectional area $A_{\text{total}}$, the uncertainties are determined as

$$
\delta d_i = \left( \sum_n \left( \frac{\partial d_f}{\partial A_i} \delta A_i \right)^2 + \sum_n \left( \frac{\partial d_f}{\partial P_i} \delta P_i \right)^2 \right)^{1/2} \tag{3.15}
$$

$$
\delta \varepsilon_{\text{eff}} = \left( \sum_n \left( \frac{\partial \varepsilon_{\text{eff}}}{\partial A_{\text{void}}} \delta A_{\text{void}} \right)^2 + \sum_n \left( \frac{\partial \varepsilon_{\text{eff}}}{\partial A_{\text{total}}} \delta A_{\text{total}} \right)^2 \right)^{1/2} \tag{3.16}
$$

Each $i$ term represents a single experimental data pair, and it is assumed that the uncertainty is uniformly distributed. For the measured porosity by the weighing method, the uncertainty is dependent on the standard deviation, $s$. The error contributed by the accuracy of the instrumentation to measure mass and dimensions are neglected due to their negligible length scales. Hence, using the root mean square method the combined uncertainty in measuring porosity is

$$
\delta \varepsilon = \sqrt{(\delta s)^2 + (\delta m_{\text{sample}})^2 + (\delta l_{\text{sample}})^2} \tag{3.17}
$$

where the uncertainty in the estimated standard deviation from the mean is is taken over the sample
size, \( n \)

\[
\delta s = \frac{s}{\sqrt{n}} \quad (3.18)
\]

Since the standard deviation on each sample set is less than 7\%, the average computed values are acceptable. Knowing the individual uncertainty in each measured parameter, the uncertainty associated with the porosity and surface area density of the model can be found, which will prove important in the analysis later to come. Again treating \( \varepsilon_{\text{model}} \) and \( \alpha_{sf} \) as functions of \( d_f \) and \( d_p \) the uncertainties are described as

\[
\delta \varepsilon_{\text{model}} = \sqrt{\sum_n \left( \frac{\partial \varepsilon_{\text{model}}}{\partial d_p} \delta d_p \right)^2 + \sum_n \left( \frac{\partial \varepsilon_{\text{model}}}{\partial d_f} \delta d_f \right)^2} \quad (3.19)
\]

\[
\delta \alpha_{sf} = \sqrt{\sum_n \left( \frac{\partial \alpha_{sf}}{\partial d_p} \delta d_p \right)^2 + \sum_n \left( \frac{\partial \alpha_{sf}}{\partial d_f} \delta d_f \right)^2} \quad (3.20)
\]

The uncertainties of the porosity and surface area density based on the idealized model and are tabulated in Table 4.5. All uncertainties are expressed as percentages of the parameter value \( x \).

\[
\sigma_x = \frac{\delta x}{x} \times 100\%
\]
Chapter 4

Hydraulic Characteristics in Metal Foams

4.1 Introduction

The objective of the following chapter is to investigate the hydraulic characteristics of a single-phase fluid, flowing through a metal foam with a thermal sprayed skin. The use of open-cell metal foams in heat exchanger applications requires a thorough understanding of the pressure drop behavior through the porous structure in order to assess the trade-off between the increased heat transfer and the associated increase in pressure drop [23]. Extensive work has been done to characterize the pressure-drop through porous media, but has been limited to packed granular beds or beds of packed spheres. Due to the structural and geometric differences between open-cell metal foams and packed beds of spheres, a renewed effort must be made to characterize fluid flow and pressure-drop within metal foams and confirm the validity of existing theory. As a result of tortuosity and irregular-shaped flow passages that exist in the porous matrix, there is a continuous disruption of any hydrodynamic boundary layers. The fluid flow recirculates at the back of the solid fibers, where turbulence and unsteady flows usually occur [24]. The geometric complexity of the porous medium prevents exact solutions to the transport equations. This has led researchers to predominately rely on experimentation and empirical models to determine flow characteristics [24, 25].
4.1.1 Flow Laws for Metal Foams

Over the past 150 years, several different models have been developed to characterize fluid flow in porous media on a macroscopic scale. An extensive history of the development of flow laws in porous media can be found within [25]. The various flow models used for porous media were introduced in Chapter 2 and are discussed here in more detail.

- **Darcy Law**

  The first attempt can be traced back to Henry Darcy’s (1856) experimental observations of water flowing through columns of packed earth and sand. In applying a volumetric force balance, he developed an equation which related the the volumetric flow rate of water through a column of packed earth to the height of the water in the column [26]. However, Krüger (1918) is credited as the first to incorporate directly the idea of viscosity into Darcy’s equation [27, 28]. This equation has now been rewritten as the well-known *Darcy law* which states that the bulk resistance to the flow or the pressure-drop per unit length for a fluid flow through a porous medium is proportional to the product of the fluid velocity and the dynamic viscosity, and inversely proportional to the permeability.

\[
\nabla P = -\frac{\mu}{K} u
\]

The *permeability* $K$ is a measure of the flow conductance of the porous matrix, which has been shown to be independent of the nature of the fluid flow but emerges as a function of the flow geometry (i.e pore structure). In flow characterization, the Reynolds number is a non-dimensional parameter used to describe a flow based on its characteristic length. Various definitions of the characteristic length have been proposed in the past [29], but the most widely accepted definition is defined at the macroscopic level where the square root of the permeability is used to describe porous media. Using $\sqrt{K}$ as the appropriate length the permeability based Reynolds number is given as Equation (4.2). Through extensive experimentation and theoretical studies it has been found that Darcy’s law is applicable to fluid flows at relatively low velocities, where $Re_K$ is of order unity or smaller [12].

\[
Re_K = \frac{\rho u \sqrt{K}}{\mu} < O(1)
\]

- **Dupuit-Forchheimer Equation**

  As the flow velocity increases, inertial drag becomes more prevalent as a result of solid obstacles inhibiting the fluid flow [30, 31]. This effect subsequently results in a deviation from Darcy’s linear
law and is attributed to the dominance of inertial forces over viscous forces. To account for this effect a quadratic velocity term is added as suggested by Dupuit (1863) [34, 35], which yields the following relation for pressure drop

$$\nabla P = -\frac{\mu}{K} u - \frac{C_F}{\sqrt{K}} \rho u^2$$  \hspace{1cm} (4.3)

As a note, the dimensionless inertial coefficient $C_F$ is commonly referred to as the Forchheimer (1901) coefficient after the extensive work that he published, in which he states his agreement with the use of the quadratic extension in the pressure drop equation [35]. The inertial coefficient varies with the nature of the porous medium [35, 39] and can be as small as 0.01 in the case of metal foams [36]. However, the inertial-drag effects of metal foams do not resemble those generated in packed beds of spheres, due to the complex structure of the metal foams. For this reason previous empirical correlations are invalid and instead the inertial coefficient is evaluated empirically for specific metal foam samples [28, 40]. The addition of the quadratic term in Equation (4.3) has been proved to be applicable to packed beds of spheres in the range of $5 \leq \text{Re}_K < 80$ by Dybbs and Edwards [37] and Fand et al. [38] for randomly packed spheres with various diameters.

**Brinkman’s Equation**

The boundary effects result from shear stresses initiated at the wall of the surfaces containing the solid matrix. This boundary frictional resistance is in addition to the bulk frictional drag induced by the solid matrix (Darcy term). To include viscous effects at the boundary, Brinkman (1947) developed the following model known as the Brinkman’s extension of the Darcy’s law [41].

$$\nabla P = -\frac{\mu}{K} u + \mu' \nabla^2 u$$  \hspace{1cm} (4.4)

In this equation $\mu'$ is the effective viscosity. Shown experimentally by Lundgren and Neale et al. [42, 43] $\mu'$ and $\mu$ are set to equal each other at high porosity cases. However, employing the Brinkman equation as the general momentum equation within a given situation complicates the analysis and can be further complicated if the porous medium is not isotropic [42].

The velocity terms in Equations (4.1), (4.3), (4.4) and the Reynolds number can either be the Darcian velocity of the fluid flow, which is based on the cross-sectional dimension of the channel

$$u_D = \frac{Q}{A_c}$$  \hspace{1cm} (4.5)
or the pore (filter) velocity as given by the Dupuit-Forchheimer relation, which account for the solid phase in the channel by dividing the Darcian velocity by the volumetric void fraction (porosity) of the porous medium [44].

\[
  u_p = \frac{u_D}{\varepsilon}
\]  

(4.6)

Either velocity can be used to characterize fluid flow using the above mentioned relations, but one must state which velocity is used in the calculations [31, 44].

4.2 Pressure Drop

All data was calculated and reported on a Darcian flow velocity basis, as given in Equation (4.5). This velocity accounts for only the channel dimensions and is independent of the porosity of the test sample. For practical purposes this definition is chosen so that the data sets obtained in the present study can be compared to other data sets on porous media. Using the apparatus described in Chapter 2, the pressure gradient data was collected (refer to Appendix C for pressure drop data) for the prepared thermal spray channels. Reviewing the pressure drop data as in Figure 4.1 it becomes apparent that the flow through the open-cell metal foam deviates from Darcy law flow behavior and mimics a quadratic function with respect to flow velocity.

Thus the relation between the one-dimensional pressure gradient and flow velocity will take on the form of Equation (4.3), yielding the the following quadratic equation for the length-normalized pressure drop.

\[
\frac{\Delta P}{L} = Au + Bu^2
\]  

(4.7)

where

\[
A = \frac{\mu}{K}
\]  

(4.8)

and

\[
B = \frac{\rho C_F}{\sqrt{K}}
\]  

(4.9)

There are several ways in which the permeability \(K\) and inertial coefficients \(C_F\) can be calculated through experimentation. The first approach has been widely implemented, more notably by Beavers et al. [39] and by Givler and Altobelli [40, 46, 47] by modifying Equation (4.7) to bring it into a linear function of velocity.

\[
\frac{\Delta P}{Lu} = A + Bu
\]  

(4.10)
Figure 4.1: Experimentally obtained pressure drop data plotted as a function of the flowrate of air during a typical experimental run for the nickel foam channels. The experimental uncertainty values are ±0.003 psi in the pressure drop measurement and ±1.5 L/min in the fluid flow velocity measurements.
By applying a best least-square fit to the RHS of Equation (4.10), $K$ and $C_F$ are determined by using the y-intercept and the slope of this linear function. However, this method has been shown to lack accuracy resulting in higher errors associated to the final values of $K$ and $C_F$ since it relies on extrapolation \[48\]. Therefore, a more direct and accurate way has been introduced by Antohe et al. \[48\] by using a least-squares quadratic regression through Equation (4.7), which gives the following results for the coefficients $A$ and $B$ \[49\]. However, in this study, it is noted that the difference in the values of $K$ and $C_F$ obtained from each method was negligible and the parabolic method was used throughout since it is more convenient.

$$\begin{align*}
A &= \frac{\sum x_i y_i \sum x_i^4 - \sum x_i^4 \sum x_i y_i}{\sum x_i^2 \sum x_i^4 - (\sum x_i^3)^2} & (4.11) \\
B &= \frac{\sum x_i^2 y_i \sum x_i^4 - \sum x_i y_i \sum x_i^3}{\sum x_i^2 \sum x_i^4 - (\sum x_i^3)^2} & (4.12)
\end{align*}$$

In these equations the $x_i$'s refer to the various fluid velocity values at which the data was taken, $y_i$'s refer to the corresponding pressure-drop per unit length values and $n$ to the number of experimental points. Knowing $A$ and $B$, the permeability and the inertia coefficient are obtained by backsolving Equations (4.8) and (4.9). The advantage of the latter method is that it provides the means for an accurate uncertainty analysis that is critical when analyzing and reporting quantities derived from experimental results. In all cases, the regression analysis yielded fits with $R^2 > 94.0\%$ indicating that the quadratic relationship was is indeed valid. Figures 4.2 and 4.3 graphically represent the procedure used to obtain the values of $A$ and $B$ by fitting a second-order curve to the experimental obtained data points averaged over seven measurements at a given fluid velocity, the standard error did not exceed 5%. The lines passing through the points are the second-order curves fitted to the pressure drop data points using the curve fitting techniques described by Equations (4.7) to (4.12). The results of the experimentation show the effects of foam geometry on the permeability and inertial coefficient for each test sample, which are tabulated in Table 4.1.
CHAPTER 4. HYDRAULIC CHARACTERISTICS IN METAL FOAMS

Figure 4.2: The experimentally obtained pressure drop data are plotted along with the fitted curves for the Ni foam samples. The experimental uncertainty values are ±81.43 Pa/m in the length-normalized pressure drop measurement and ±0.06 m/s in the fluid flow velocity measurements.

Figure 4.3: The experimentally obtained pressure drop data are plotted along with the fitted curves for the Cu foam samples. The experimental uncertainty values are ±81.43 Pa/m in the length-normalized pressure drop measurement and ±0.06 m/s in the fluid flow velocity measurements.
Equation (4.7) provides the global balance of forces acting on a fully saturated permeable volume with uni-directional steady flow under the incompressible condition. Hence the effect of flow compressibility was neglected and the fluid was assumed to be an ideal gas. It can be seen that the pressure drop in all the metal foam channels increases with increases in the flow velocity and with the foam pore density. The four metal foam channels which were tested were of nearly the same porosity (within 0.6%), and the only difference between the samples was the average pore diameter. Referring to Table 3.5, the porosities of the foams ranges from 0.937 to 0.944 and the pore diameters varied from an average of 2.87 mm to 0.63 mm. The difference in pore diameters appears to dramatically affect the permeability and inertia coefficients of the foams, reaffirming their dependence on the geometry of the porous medium as previously stated. Decreasing the pore diameter decreased the permeability and increased the inertia coefficient. The 10PPI foams which had a pore size in the order of 2 mm, generated the least amount of flow resistance with permeabilities of $8.83 \times 10^{-8} \, \text{m}^2$ and $9.24 \times 10^{-8} \, \text{m}^2$, with inertial coefficients of 0.0326 and 0.0335 for the nickel and copper foams respectively. In contrast the 40PPI channels with significantly smaller pore sizes in the range of 0.63 mm, had greater flow resistances with permeabilities of $2.34 \times 10^{-8} \, \text{m}^2$ and $2.10 \times 10^{-8} \, \text{m}^2$, and with inertial coefficients of 0.0640 and 0.0695 for the nickel and copper foams respectively. The increase in the flow resistance of the 40PPI samples directly relates to the “effective surface length” as explained by Lage [25], which relates an increase in drag to the increase in the specific surface area. Also, the inertial coefficient essentially represents the magnitude of deviation from Darcy flow. Thus the higher $C_F$ values obtained from the 40PPI foams indicates that the flow experiences greater degrees of obstruction as a result of smaller pore sizes. From Chapter 3, it was determined that the nickel and copper foams of the same pore density have similar geometric structures as a result of the manufacturing process. Due to these similarities in structures, flow characteristics with similar proportions were obtained from the foam samples of the same pore density, indicating again that the fluid flow through a porous medium is highly dependent on geometric structure and independent of foam material.

An effort has been made to compare the hydraulic characteristics obtained in the present study to those

<table>
<thead>
<tr>
<th>Foam</th>
<th>$A$</th>
<th>$B$</th>
<th>$K$</th>
<th>$C_F$</th>
<th>$\sigma_K$ (%)</th>
<th>$\sigma_{C_F}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni-10PPI</td>
<td>209.49</td>
<td>129.99</td>
<td>8.83</td>
<td>0.0326</td>
<td>14.6</td>
<td>4.4</td>
</tr>
<tr>
<td>Ni-40PPI</td>
<td>788.71</td>
<td>494.53</td>
<td>2.34</td>
<td>0.0640</td>
<td>13.1</td>
<td>6.2</td>
</tr>
<tr>
<td>Cu-10PPI</td>
<td>200.02</td>
<td>130.28</td>
<td>9.24</td>
<td>0.0335</td>
<td>15.9</td>
<td>3.9</td>
</tr>
<tr>
<td>Cu-40PPI</td>
<td>879.78</td>
<td>367.89</td>
<td>2.10</td>
<td>0.0695</td>
<td>12.8</td>
<td>9.5</td>
</tr>
</tbody>
</table>

Table 4.1: Calculated fluid flow characteristics for Ni and Cu metal foam channels of varying pore densities.
CHAPTER 4. HYDRAULIC CHARACTERISTICS IN METAL FOAMS

of other investigations in porous media. Firstly, to validate the present experimental results and secondly, to deduce any effect that may exist on the fluid flow through the metal foam as a result of applying the thermal sprayed skin. Since the hydraulic characteristics are intrinsically related to the density of the working fluid as per Equation (4.3), experimental results have been obtained where air was used as the working fluid such that valid comparisons can be made. There have been many investigations where other working fluids have been used such as water by Boomsma et al. [28], Bonnet et al. [50], and Du Plessis et al. [51], sodium hydroxide (NaOH) by Langlois et al. [52] and pentane by Madani et al. [53].

Table 4.2 summarizes some results found within the literature and compares them to the present study. Upon reviewing the permeability and the inertial coefficient from past studies, it can be seen that the results obtained in this study are in relatively good agreement. The values of $K$ and $C_F$ for the nickel and copper foam samples are of the same order of magnitude, validating that the analysis performed was correct and that the flow behavior was correctly characterized by the quadratic relation presented by Equation (4.3).

Since the samples that have been investigated here were of similar porosity, the effect of changing porosity on flow characteristic could not be examined. The data presented in Table 4.2 contains samples ranging in porosity from 0.90 to 0.98, which when combined with the values of the current study give the variation of flow characteristics with respect to porosity. It can be seen that there is no definitive correlation between porosity, permeability and the inertial coefficient, a claim that is confirmed by the analysis of Boomsma et al. [28]. Subsequently, in a separate investigation carried out by Boomsma et al. [59], metal sheets were brazed to the surfaces of the metal foams to create heat exchangers. They claimed that an increase of nearly 22% in flow resistance occurred as a result of adding the brazing material.

From the comparison of results in Table 4.2 it can be seen that the samples with the thermal sprayed skin experience a greater amount of flow resistance than other investigations. This can be reaffirmed by an investigation performed by Salimi Jazi et al. [6], where a 10PPI nickel foam channel with a thermal sprayed nickel-chromium (alloy) skin was studied. Results from that study show even greater pressure drop magnitudes, where their 10PPI channel experiences similar pressure losses to that of the 40PPI channel presented here. The difference in the pressure drop results can be attributed to the difference in channel geometry. The channel cross-section used in the Salimi Jazi et al. study had dimensions of 10mm by height and 100mm by width. It also believed that the left over paste that remains from the application of the skin process may be contributing to the flow resistance. However this cannot be confirmed without studying the flow characteristics of the foams without the skin.
### Table 4.2: Comparison of hydraulic characteristics of metal foam samples based on literature of previous investigations for (a) 40PPI and (b) 10PPI pore densities.

<table>
<thead>
<tr>
<th>Foam</th>
<th>PPI</th>
<th>Porosity</th>
<th>$d_f$ (mm)</th>
<th>$d_p$ (mm)</th>
<th>$K$ ($\times 10^{-8} m^2$)</th>
<th>$C_F$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tsonias</td>
<td>Cu</td>
<td>40</td>
<td>0.938</td>
<td>0.102</td>
<td>0.631</td>
<td>2.10</td>
</tr>
<tr>
<td>Tsonias</td>
<td>Ni</td>
<td>40</td>
<td>0.942</td>
<td>0.095</td>
<td>0.742</td>
<td>2.34</td>
</tr>
<tr>
<td>Bhattacharya et al.</td>
<td>Al</td>
<td>40</td>
<td>0.972</td>
<td>0.230</td>
<td>1.800</td>
<td>5.20</td>
</tr>
<tr>
<td>Calamdi</td>
<td>Al</td>
<td>40</td>
<td>0.966</td>
<td>0.200</td>
<td>1.900</td>
<td>5.50</td>
</tr>
<tr>
<td>Phanikumar et al.</td>
<td>Al</td>
<td>40</td>
<td>0.958</td>
<td>0.209</td>
<td>1.727</td>
<td>5.99</td>
</tr>
<tr>
<td>Liu et al.</td>
<td>Al</td>
<td>40</td>
<td>0.935</td>
<td>-</td>
<td>0.685</td>
<td>13.30</td>
</tr>
<tr>
<td>Dukhan</td>
<td>Al</td>
<td>40</td>
<td>0.923</td>
<td>-</td>
<td>-</td>
<td>4.70</td>
</tr>
<tr>
<td>Hernandez</td>
<td>Al</td>
<td>40</td>
<td>0.918</td>
<td>0.102</td>
<td>1.700</td>
<td>4.54</td>
</tr>
<tr>
<td>Jin et al.</td>
<td>Al</td>
<td>40</td>
<td>0.900</td>
<td>0.113</td>
<td>-</td>
<td>2.86</td>
</tr>
<tr>
<td>Tardist et al.</td>
<td>Al</td>
<td>40</td>
<td>0.905</td>
<td>-</td>
<td>0.344</td>
<td>6.00</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Foam</th>
<th>PPI</th>
<th>Porosity</th>
<th>$d_f$ (mm)</th>
<th>$d_p$ (mm)</th>
<th>$K$ ($\times 10^{-8} m^2$)</th>
<th>$C_F$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tsonias</td>
<td>Cu</td>
<td>10</td>
<td>0.944</td>
<td>0.319</td>
<td>2.868</td>
<td>9.24</td>
</tr>
<tr>
<td>Tsonias</td>
<td>Ni</td>
<td>10</td>
<td>0.940</td>
<td>0.332</td>
<td>2.771</td>
<td>8.83</td>
</tr>
<tr>
<td>Bhattacharya et al.</td>
<td>Al</td>
<td>10</td>
<td>0.9138</td>
<td>0.450</td>
<td>3.28</td>
<td>11.00</td>
</tr>
<tr>
<td>Calamdi</td>
<td>Al</td>
<td>10</td>
<td>0.949</td>
<td>0.370</td>
<td>3.10</td>
<td>14.90</td>
</tr>
<tr>
<td>Phanikumar et al.</td>
<td>Al</td>
<td>10</td>
<td>0.939</td>
<td>0.411</td>
<td>3.41</td>
<td>11.71</td>
</tr>
<tr>
<td>Liu et al.</td>
<td>Al</td>
<td>10</td>
<td>0.918</td>
<td>-</td>
<td>1.900</td>
<td>6.23</td>
</tr>
<tr>
<td>Dukhan</td>
<td>Al</td>
<td>10</td>
<td>0.919</td>
<td>-</td>
<td>-</td>
<td>10.00</td>
</tr>
<tr>
<td>Hernandez</td>
<td>Al</td>
<td>10</td>
<td>0.914</td>
<td>0.406</td>
<td>5.080</td>
<td>10.08</td>
</tr>
<tr>
<td>Jin et al.</td>
<td>Al</td>
<td>10</td>
<td>0.910</td>
<td>0.427</td>
<td>-</td>
<td>4.21</td>
</tr>
<tr>
<td>Tardist et al.</td>
<td>Al</td>
<td>10</td>
<td>0.917</td>
<td>-</td>
<td>0.397</td>
<td>13.00</td>
</tr>
<tr>
<td>Salimi Jazi et al.</td>
<td>Ni</td>
<td>10</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.49</td>
</tr>
</tbody>
</table>
4.2.1 Transition From Linear Darcy Regime

An important deduction from the pressure drop analysis is the ability to determine when the pressure drop across the metal foam leaves the linear Darcy regime and enters the inertial-dominated regime, characterized by the quadratic velocity term in Equation (4.3). Rearranging Equation (4.7) into its linear form as in Equation (4.10), a graphical means of separating the linear and non-linear regimes is attainable. The data from the pressure drop experiments are plotted in the following figures according to Equation (4.10) against the fluid flow velocity.

Figures 4.4 and 4.5 are separated depending on pore diameter, labeled as 10PPI and 40PPI, where the discrete data points are the experimentally derived pressure drop results averaged over seven runs. Taking a closer look at Equation (4.10) it can be seen that when the coefficients \( A \) and \( B \) are constant, the plotted line has a slope of \( B \) and a y-intercept of \( A \). When \( B \) is equal to zero the line is horizontal. This horizontal region can be described as the pressure drop region where the inertial-coefficient, \( C_F \), is zero and the pressure drop is entirely governed by Darcy’s law. The transition between the two regimes is seen by the discrete experimental points indicated in Figures 4.4 and 4.5. The experimental points form a horizontal line in the Darcian flow regime where fluid velocities are less than 1.148 m/s and 0.689 m/s for the 10PPI and 40PPI foams respectively. At flow velocities greater then these transitional values the experimental data points turn onto the curve-fitted line with a non-zero slope of \( B \), which by Equation (4.10) indicates that the pressure drop is dictated by the inertial forces. Boomsma et al. \cite{28} were able to deduce that the best metric to determine the the transition between flow regimes is the permeability based Reynolds Number, \( Re_K \).

<table>
<thead>
<tr>
<th>Foam</th>
<th>( d_p ) (mm)</th>
<th>( u_{\text{transition}} ) (m/s)</th>
<th>( Re_K )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni-10PPI</td>
<td>0.742</td>
<td>1.148</td>
<td>21.83</td>
</tr>
<tr>
<td>Cu-10PPI</td>
<td>0.631</td>
<td>1.148</td>
<td>22.33</td>
</tr>
<tr>
<td>Ni-40PPI</td>
<td>2.771</td>
<td>0.689</td>
<td>6.74</td>
</tr>
<tr>
<td>Cu-40PPI</td>
<td>2.868</td>
<td>0.689</td>
<td>6.39</td>
</tr>
</tbody>
</table>

Table 4.3: Transitional Reynolds numbers for 10PPI and 40PPI metal foams.

The transition velocity of the 40PPI foams happens at a lower velocity then the 10PPI foams as a result of increased hindrance of fluid flow. This can also be attributed to the increase in tortuosity that the flow experiences in the highly obstructed 40PPI foams.
Figure 4.4: The quantity $\left( \Delta P/\Delta u \right)$ for the 10PPI Ni (2.771 mm pore diameter) and Cu (2.868 mm pore diameter) foams are plotted to show the pressure drop deviation from linear Darcy flow at fluid velocities greater than 1.148 m/s.

Figure 4.5: The quantity $\left( \Delta P/\Delta u \right)$ for the 40PPI Ni (0.742 mm pore diameter) and Cu (0.631 mm pore diameters) foams are plotted to show the pressure drop deviation from linear Darcy flow at fluid velocities greater than 0.638 m/s.
CHAPTER 4. HYDRAULIC CHARACTERISTICS IN METAL FOAMS

4.2.2 Uncertainty Analysis

The uncertainties generated by the least-squares curve fit are given by the general formula for error propagation as described by Taylor [49]. If $A$ and $B$ can be treated as functions of $x_i$ and $y_i$, the uncertainties in coefficients of Equation (4.7) are determined as

$$
\delta A = \sqrt{\sum_n \left( \frac{\partial A}{\partial x_i} \delta x_i \right)^2 + \sum_n \left( \frac{\partial A}{\partial y_i} \delta y_i \right)^2}
$$

(4.13)

$$
\delta B = \sqrt{\sum_n \left( \frac{\partial B}{\partial x_i} \delta x_i \right)^2 + \sum_n \left( \frac{\partial B}{\partial y_i} \delta y_i \right)^2}
$$

(4.14)

Each $i$ term represents a single experimental data pair, Darcy flow velocity and length-normalized pressure drop. $\delta x_i$ and $\delta y_i$ are the uncertainties in the velocity and the pressure drop which are determined from the accuracy of the experimental instrumentation used to measure the respective quantities. In evaluating Equations (4.13) and (4.14) it is assumed that the confidence level in the uncertainty of $A$ and $B$, will be the same as the confidence level of uncertainty in the $x_i$'s and $y_i$'s. The corresponding partial derivatives of Equations (4.11) and (4.12) were calculated and inserted in Equations (4.13) and (4.14) in order to yield $\delta A$ and $\delta B$. The uncertainty in the value of $K$ is equal to the uncertainty in the value of $A$ assuming that there is zero uncertainty in the value of dynamic viscosity, thus the following relationship is given

$$
\sigma_K = \frac{\delta A}{A} \times 100\%
$$

(4.15)

The uncertainty in the inertial-coefficient is derived in a similar fashion, assuming that there is zero uncertainty in the density term, the uncertainty in the inertia-coefficient is given as

$$
\sigma_{C_F} = \frac{\delta B}{B} \times 100\%
$$

(4.16)

The uncertainties of the permeabilities and the inertial-coefficients are tabulated in Table 4.1.

4.3 Friction Factor

The hydraulic characteristics of heat exchangers can be compared to one another by examining their non-dimensional flow factors. Such factors include the Reynolds number as defined by porous media [31], where the characteristic length is replaced by the square root of permeability, Equation (4.12).
other commonly used non-dimensional flow describing factor is the friction factor \( f \), which defines the pressure loss due to friction and is given in Equation (4.17)

\[
f = \left( \frac{\Delta P}{L} \right) \frac{D_{hyd}}{\rho u^2}
\]

where the hydraulic diameter, \( D_{hyd} \), simplifies to the height \( H \) of a square cross-sectional channel.

\[
D_{hyd} = \frac{4A}{P} = H
\]  

(4.18)

From Equation (4.13) a general correlation for the friction factor applicable to metal foams can be derived. This factor provides information concerning the amount of pressure drop developed across the heat exchanger and can be used accordingly to evaluate the gain in heat transfer performance to the required pumping power \([59, 58]\). Substituting and rearranging (4.3) with the following definitions the friction factor is obtained, Equation (4.19).

\[
f = \left( \frac{\Delta P}{L} \right) \frac{D_{hyd}}{\rho u^2} \quad Da = \frac{K}{D_{hyd}^2} \quad Re = \frac{\rho V \sqrt{K}}{\mu}
\]

\[
f = \frac{1}{Re_K Da^{0.5}} + \frac{C_F}{Da^{0.5}}
\]  

(4.19)

From the hydraulic parameters of the metal foams listed in Table 4.1, a high pore density foam corresponds to a low permeability and a high inertial coefficient. The data in Figures 4.6 and 4.7 show that the experimental results are in good agreement with the proposed correlation defined in Equation (4.19), which suggests that a small value of \( K \) and a large value of \( C_F \) will result in a large friction factor at a constant Reynolds number for different porous mediums. It can also be shown that significant departures of the experimental results from Darcy’s law first occur at Reynolds numbers on the order of one, confirming the applicability of Darcy’s law and the existence of inertial effects at higher flow rates. It is also suspected that the irregularity of the skin contour at the boundary between the skin and foam may be contributing to the disruption of coolant flow.

One of the main objectives of defining a friction factor correlation, is to provide a means of determining the pressure drop across a heat exchanger independent of its dimensions without the need to perform rigorous experiments. Figure 4.8 compares several correlations defined by other investigators with the experimental results of the current study and it seems that they all fit with relatively good agreement.
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Figure 4.6: Friction factors for steady-state flow through nickel foams of varying pore densities.

Figure 4.7: Friction factors for steady-state flow through copper foams of varying pore densities.
Figure 4.8: Comparison of friction factor correlations.
Due to the geometrical similarities between the copper and nickel samples, a global correlation for the 10PPI and 40 PPI foams is proposed. From the analysis the following correlations are proposed to describe the pressure drop

$$\begin{align*}
40\text{PPI} & \quad fDa^{1/2} = \frac{1}{Re_K} + 0.067 \\
10\text{PPI} & \quad fDa^{1/2} = \frac{1}{Re_K} + 0.033
\end{align*}$$

(4.20)

(4.21)

The 10PPI foam correlation predicts the experimental results with a maximum deviation of 8% at flow rates $Re_K > 4$. For the 40PPI foams a maximum deviation of 11% occurs at $Re_K > 4$. 
Chapter 5

Heat Transfer Characteristics in Metal Foams

5.1 Introduction

The problem under investigation is forced convection of incompressible fluid flow through open-celled metal foams with a thermal sprayed skin. As previously discussed, the complexity of the the cellular structure usually precludes a detailed microscopic investigation of the transport phenomena at the pore level in porous media. Therefore, the general heat transfer equations are commonly integrated over a representative elementary volume, which accommodates the fluid and solid phases within the porous structure. In the present study, two volume-averaging techniques will be applied to the heat transfer investigations. One is averaging over the representative model containing both the solid and the fluid phases, and the other is averaging separately over each of the phases, thus resulting in a separate energy equation for each phase. These two models are referred to as the one-equation model (conduction) and the two-equation model (forced convection). The one-equation model is valid when the the local temperature difference between the the fluid and solid phases are negligible small. However, in forced convection scenarios where air is used as the coolant, the temperature differences between the two phases cannot be neglected and the two-equation model is applied. Based on the literature heat transfer enhancement of metals foams is caused by two different mechanisms.

1. Metal foams cause the flow to be more turbulent, thus increasing the heat transfer from the bounding substrate walls. This is typically referred to as “thermal dispersion”
2. In the case of metal foams with a high thermal conductivity, the foams act to provide an extended surface for heat transfer.

However, the heat transfer behavior within metal foam is far more complex than the aforementioned mechanisms. Several analytical and experimental investigation have been performed which have reported several modes of heat transfer occurring simultaneously. Local heat transfer characteristics are highly dependent on the foam structure, where the complexity associated with the porous structure makes it difficult to predict the effective thermal conductivity as well as the velocity profiles \[60, 61\]. Furthermore, it has been reported that at certain flow regimes, conduction through the fluid phases is the dominant heat transfer mode \[62, 63\].

### 5.2 Conduction

Conduction was achieved by applying a symmetrical constant heat flux boundary condition to the walls of the metal foam channel, using the apparatus described in Chapter 3. By closing the compressed air supply stagnate air remains in the channel. In order to avoid burning the tape heater, heat fluxes at constant voltages of 10V, 20V and 30V are selected producing heat fluxes in the ranges of 107 W/m², 428 W/m², and 963 W/m² respectively. The axial wall and bulk fluid temperatures for each heat flux input are recorded, and the experiment is repeated for each metal foam channel type. A typical set of experimentally obtained temperature profiles are shown in Figure 5.1. Refer to Appendix D for raw data.

As expected, the curves follow the characteristic parabolic shape of a one-dimensional heat conduction through a planar wall. What should also be noticed is that the temperature difference between the wall temperature and the bulk fluid temperature does not vary more than 24.5°C across the length of the channel, hence validating the use of the one-dimensional model.

#### 5.2.1 Simplified Heat Conduction Model

Estimating the heat conduction through the fluid-saturated metal foam in a stagnant flow situation presents a variety of difficulties due to a number of factors which influence the effective thermal conductivity of the fluid-saturated medium. The system consists typically of two different substances which are also in two different physical phases, in this case gas-solid.
Figure 5.1: Experimentally obtained temperature results for the wall and bulk fluid temperatures, plotted as a function of the axial distance from the inlet of a 10PPI nickel foam channel. The experimental uncertainty values are ±2.6°C in the temperature measurement and ±0.5 mm in the length measurement.
This combination of various substances and phases including an intricate solid phase geometry has several consequences. The structure of the solid phase influences the effective thermal conductivity to a large extent because the thermal conductivity of the solid phase is typically greater than that of the fluid phase. Adding to this complexity is the consideration of the interconnecting nature of the medium resulting in a different proportion of each phase at any given cross-section along the length of the medium. The analysis of the thermal conduction through a fluid-saturated porous medium which calculates the effective thermal conductivity is dependent on the effective area of both phases of the medium. By understanding the proportion of each phase within the medium, the effects of conduction through the solid phases can be accounted for in the analysis of forced convection.

Heat conduction is governed by the well-known Fourier’s law of conduction which is a macroscopic equation that describes the macroscopic effects of particle motion on an atomic scale. The simplest form of the conduction equation considers only one dimension and requires the following assumptions:

- The properties of the material remain constant with respect to time, space and temperature.
- There is no heat generation within the material.

Under these two assumption the one-dimensional heat conduction equation becomes

$$Q = -kA \frac{dT}{dx}$$  \hspace{1cm} (5.1)

The model proposed to predict the thermal conduction in the metal foams uses Equation (5.1) as a base. The most simplified method of predicting the thermal conductivity of the fluid saturated porous medium is to consider one-dimensional heat transfer. The method is well described by Bejan \[73\] by considering a representative elementary volume, also known as an REV, with a prismatic cross-section, Figure 5.2, in which the porosity can be defined by the following ratio

$$\varepsilon = \frac{A_f dx}{A_{total} dx}$$  \hspace{1cm} (5.2)

In order to obtain the proposed relation in Equation (5.2) the one-dimensional steady-state heat conduction through a planar channel is taken on a per volume basis. The conduction equations are derived separately for each phase and are then coupled together using the local thermal equilibrium assumption, \(T_s = T_f = T\). After performing the derivation (refer to Appendix F), the overall axial temperature
Figure 5.2: One-dimensional prismatic element which serves as a basis for the development of heat conduction in a porous medium.

Figure 5.3: Application of the one-equation model to the control volume solve for axial heat conduction.
distribution for conduction through the metal foam channel is expressed as

\[ k_{\text{eff}} \frac{d^2 T}{dx^2} = q'''' \]  \hspace{1cm} (5.3)

where the one-dimensional representation for the effective thermal conductivity of a fluid-saturated porous medium becomes

\[ k_{\text{eff}} = \varepsilon k_{\text{fluid}} + (1 - \varepsilon) k_{\text{solid}} \]  \hspace{1cm} (5.4)

This derivation comes about by simply accounting for the area fraction of each substance, giving the resulting relation based on the porosity and the thermal conductivity of each substance. Equation (5.4) does not account for the convection between the solid and the fluid phases, contact resistance between the unit cells of the foam or the foam and the thermal sprayed skin, radiation heat transfer, nor structural features. The stagnate flow conduction model requires additional assumptions concerning the behavior of heat flow through the medium. They include:

- The cross-section of the porous medium is uniform across the length of the channel.
- Natural convection and radiation heat transfer effects inside the the porous medium are neglected.
- The physical properties of the fluid and solid phases remain constant through out the temperature range.
- The temperature distribution of the solid phase, is constant in the vertical direction, thus the wall temperature is taken as the bulk temperature of the solid phase, \( T_{\text{wall}} = T_{\text{solid}} \).
- The fluid and solid phases are in local thermal equilibrium, \( T_{\text{fluid}}(x) = T_{\text{solid}}(x) \).

The non-dimensionalization of Equation (5.3) results in the following groups to appear:

\[ \theta (X) = \frac{T(x) - T_1}{T_2 - T_1} \]  \hspace{1cm} (5.5)

\[ X = \frac{x}{L} \]  \hspace{1cm} (5.6)

\[ \phi = \frac{q''''L^2}{k_{\text{eff}}(T_2 - T_1)} \]  \hspace{1cm} (5.7)

Where the second-order differential equation, Equation (5.3), is solved using specified temperature values at the inlet and outlet as boundary conditions. Applying the non-dimensional terms with its respective
boundary conditions, Equation (5.3) results in the following

\[
\theta = \frac{\phi}{2} X^2 + \left(1 - \frac{\phi}{2}\right) X
\]

where \(\theta = 0\) at the inlet \((X = 0)\) and \(\theta = 1\) at the outlet \((X = 1)\).

### 5.2.2 Predicting Porosity from \(k_{\text{eff}}\)

Accounting for the heat losses from the heater, it can be seen in Figures 5.4 and 5.5 that the one-dimensional conduction model fits the experimental results with relatively good agreement. As previously mentioned the effective thermal conductivity is dependent on the relative portions of the solid and fluid phases, namely the porosity. Using an iterative method the model can be fitted to the experimental results by adjusting the porosity so that the maximum deviation between the results and model are minimized. From this an effective area upon which conduction acts through the metal foam channel can be calculated and compared to the measured porosity obtained in Chapter 3. Tables 5.3 and 5.4 summarize the effective conductivities of each metal foam channel along with their respective porosity.

<table>
<thead>
<tr>
<th>Foam</th>
<th>(q') (\text{W/m}(^2)}</th>
<th>(k_{\text{eff}}) (\text{W/mK})</th>
<th>(\varepsilon)</th>
<th>(\sigma_{\text{max}}) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni 10PPI</td>
<td>107</td>
<td>16.35</td>
<td>0.82</td>
<td>16.3</td>
</tr>
<tr>
<td></td>
<td>428</td>
<td>19.97</td>
<td>0.78</td>
<td>12.3</td>
</tr>
<tr>
<td></td>
<td>963</td>
<td>23.61</td>
<td>0.74</td>
<td>7.9</td>
</tr>
<tr>
<td>Ni 40PPI</td>
<td>107</td>
<td>20.88</td>
<td>0.77</td>
<td>12.2</td>
</tr>
<tr>
<td></td>
<td>428</td>
<td>25.42</td>
<td>0.72</td>
<td>11.9</td>
</tr>
<tr>
<td></td>
<td>963</td>
<td>33.57</td>
<td>0.63</td>
<td>22.2</td>
</tr>
</tbody>
</table>

Table 5.1: Computed effective thermal conductivities for Ni metal foam channels with respective porosity and uncertainties, based on one-dimensional conduction model.

<table>
<thead>
<tr>
<th>Foam</th>
<th>(q') (\text{W/m}(^2)}</th>
<th>(k_{\text{eff}}) (\text{W/mK})</th>
<th>(\varepsilon)</th>
<th>(\sigma_{\text{max}}) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cu 10PPI</td>
<td>107</td>
<td>80.22</td>
<td>0.80</td>
<td>23.2</td>
</tr>
<tr>
<td></td>
<td>428</td>
<td>32.10</td>
<td>0.92</td>
<td>11.4</td>
</tr>
<tr>
<td></td>
<td>963</td>
<td>36.11</td>
<td>0.91</td>
<td>10.1</td>
</tr>
<tr>
<td>Cu 40PPI</td>
<td>107</td>
<td>68.19</td>
<td>0.83</td>
<td>20.7</td>
</tr>
<tr>
<td></td>
<td>428</td>
<td>32.10</td>
<td>0.92</td>
<td>13.5</td>
</tr>
<tr>
<td></td>
<td>963</td>
<td>40.12</td>
<td>0.90</td>
<td>6.7</td>
</tr>
</tbody>
</table>

Table 5.2: Computed effective thermal conductivities for Cu metal foam channels with respective porosity and uncertainties, based on one-dimensional conduction model.

Upon examining the derived results from the numerical model two major discrepancies arise. Shown in Table 5.1 the thermal conductivities obtained from the nickel foam seem to increase while porosity
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Figure 5.4: Comparison of one-dimensional conduction model with experimental results for axial temperature distribution for 10 PPI Ni metal foam with varying heat flux values.

Figure 5.5: Comparison of one-dimensional conduction model with experimental results for axial temperature distribution for 40 PPI Cu metal foam with varying heat flux values.
CHAPTER 5. HEAT TRANSFER CHARACTERISTICS IN METAL FOAMS

decreases. Intuitively, this is not plausible, since porosity is a constant geometric feature, but it seems to change with a given heat flux. With respect to the copper foams, this is less of an issue as for the majority of the results the porosity values seems to be quite consistent. The second flaw lies in the deviation between the magnitudes of porosity values obtained here and those measured in Chapter 3. For both the copper and nickel foams it was established that the porosity was constant around 0.94, but based on the models the nickel foams are predicting a maximum 33% deviation ($\varepsilon = 0.74$) and the copper at 11% deviation ($\varepsilon = 0.8$). From these issues it is evident that there are limitations to the interpretation of the one-dimensional conduction model, despite its ability to predict the temperature profiles. At this point a qualitative discussion will be attempted to explain the reason for these faults, but it is suggested that a more comprehensive numerical model be developed along with better controlled experimentation to understand the full behavior of conduction through these porous mediums.

Firstly, the one-dimensional model neglects the effects of secondary heat transfer mechanisms through the medium, namely natural convection and radiation effects. At higher heat fluxes, these mechanisms can contribute significantly to the overall heat transfer, resulting in an increase in effective thermal conductivity of both phases. To maintain a given energy balance, if thermal conductivity increases a proportional decrease in effective area (i.e. porosity) must occur, which is the trend displayed in the results. For the second issue, the porosities of the foam measured in Chapter 3, are based on samples where the solid phase is entirely comprised of metal foam, and is reflected in the derivation of the numerical model. In the experiment, the solid phase is made up of several constituents that can influence conduction, which includes the metal foam, the thermal skin, and the solid metal bracket which encapsulates the channel. Considering these additions, at any given cross-section the porosity would effectively be less than the porosity of the metal foam by itself.

5.3 Forced Convection

Forced convection was achieved by running air at different velocities through the metal foam using the experimental apparatus described in Chapter 3. A typical experimental run started with the application of a symmetrical constant heat flux boundary condition to the walls of the metal foam channel, by maintaining a constant voltage. For a given heat flux the flow rate of air was varied from 5 L/min to 80 L/min. This experimental procedure is repeated for voltages of 20 V, 40 V, 60 V and 80 V, corresponding to heat fluxes of 428 W/m², 1711 W/m², 3851 W/m² and 6846 W/m². All measurements are recorded under steady-state conditions where the temperatures of the solid and fluid are assumed to have changed
less than 5°C in 10 minutes. The inlet pressure of the air during the experiments did not vary more than 1 ± 0.5 psi to consider a change in physical properties within the calculations. A typical set of experimental results are shown in Figure 5.6.

As expected for a constant surface heat flux the temperature curves for both the wall and the fluid increases linearly in the flow direction. It can also be noticed that as the flow rate of air decreases, end effects become more prevalent, perhaps indicating conduction loss at the ends of the sample due to the stainless steel bracket and end fittings to the apparatus.

### 5.3.1 Convective Heat Transfer Coefficient

Convection is defined in the most basic terms as the transport of heat by a fluid. This mode of heat transfer is actually a combination of two processes, heat conduction to and through the fluid, and the bulk motion of fluid. A macroscopic method to provide a measure of the transmission of heat between a fluid and a surface is defined by Newton’s law of cooling which states that the heat flux is directly proportional to the product of the temperature difference and the heat convection coefficient.

\[
Q = hA(T_s - T_f)
\]  
(5.9)

The concept of similitude allows one to define dimensionless numbers which provide a means to compare systems. In a closed channel the characteristic length is taken to be the hydraulic diameter. The factors controlling forced convection are defined as:

**Nusselt number**

\[
Nu = \frac{hD_{hyd}}{k_f}
\]  
(5.10)

**Reynolds Number**

\[
Re = \frac{\rho u D_{hyd}}{\mu}
\]  
(5.11)

**Prandtl Number**

\[
Pr = \frac{\mu c_p}{k_f}
\]  
(5.12)

The one-dimensional model used to study heat conduction, will be reapplied to predict heat convection of forced flow through a porous medium. In order to derive an appropriate definition of the heat transfer coefficient, the following assumptions regarding the forced convection heat transfer through the medium are considered:
Figure 5.6: Experimentally obtained temperature results for (a) the channel wall and (b) the bulk fluid, plotted as a function of the axial distance from the inlet of the metal foam channel. Experimental results correspond to a heat flux of 3851 W/m² for a 10PPI nickel foam channel. The experimental uncertainty values are ±2.6°C in the temperature measurement and ±0.5 mm in the length measurement.
• The porous medium is uniform, based on the ideal model proposed in Chapter 3, with a surface area density of $\alpha_{sf}$.

• Natural convection and radiation heat transfer effects inside the the porous medium are neglected.

• The physical properties of the fluid are evaluated at the bulk mean temperature and at 1 atm pressure.

$$T_f = \frac{T_{\text{inlet}} + T_{\text{outlet}}}{2} \quad (5.13)$$

• The temperature distribution of the solid phase, is constant in the vertical direction, thus the wall temperature is taken as the bulk temperature of the solid phase, $T_{\text{wall}} = T_{\text{solid}}$.

• The only heat transfer mechanism to the fluid is by convection from the solid, and conduction through the fluid is neglected.

As a result of the above assumptions, it can be shown through the elemental volume derivation that the heat transfer to the fluid flowing through the porous medium is equal to the increase in the energy of the fluid

$$Q = \dot{m}c_p(T_{f,\text{outlet}} - T_{f,\text{inlet}}) = hA_{sf}(T_s - T_f) \quad (5.14)$$

In an attempt to appropriately characterize the convective heat transfer performance of the metal foam channels, a heat transfer coefficient per unit volume of foam (volumetric heat transfer coefficient, $h_{sf}$) will be the principle characteristic used. In most heat transfer applications, the available space for heat exchangers is limited and usually predetermined. In this situation a heat transfer coefficient per unit volume of a specific configuration at a given coolant mass flow rate will be measured, which may later facilitate the design process. This explains the attractiveness of using the volumetric heat transfer coefficient as a categorizing parameter. From the experimental results it is only possible to determine a bulk volumetric heat transfer coefficient based on the local wall temperature and height-wise average fluid temperature, hence the reason for assuming a bulk solid temperature equal to the local wall temperature.

In actuality, it should be made clear that the bulk volumetric heat transfer coefficient will be dependent on the thickness of the foam normal to the flow direction, as the local solid temperature is dependent on the the distance from the heated wall. As the foam conductivity increases or the foam thickness decreases, the local heat transfer coefficient will contribute differently towards the bulk volumetric coefficient.

Considering Equation \((5.14)\), as a function of length the energy increase of fluid is expressed as a portion
of the overall energy increase.

\[
\dot{m}c_p (T_{f,o} - T_{f,i}) \frac{\Delta x}{L} = h(x)A_{sf} (T_s(x) - T_f(x)) \tag{5.15}
\]

This is then divided by the total volume of the channel to derive a volumetric expression, where the porosity definition defined in Equation (5.2) is reapplied.

\[
\frac{\rho c_p \varepsilon u (T_{f,o} - T_{f,i}) \Delta x}{L^2 (T_s(x) - T_f(x))} = \frac{h(x)A_{sf}}{A_t L} = h(x)\alpha_{sf} = h_{sf}(x) \left[ \frac{W}{m^3 K} \right] \tag{5.16}
\]

The convection heat transfer coefficient in general, varies along the the flow direction. Thus an average heat transfer coefficient for a surface is determined by properly averaging the local convection heat transfer coefficient over the entire length, where an average volumetric heat transfer coefficient is expressed as

\[
\bar{h}_{sf} = \frac{\Delta x}{L} \sum_n h_{sf}(x) \tag{5.17}
\]

The average volumetric heat transfer coefficients based on the experimental results are shown in Figures 5.7 and 5.8 for the nickel and copper foams with similar pore sizes. It can be seen from the figures that, overall, the copper foams have higher volumetric heat transfer coefficients. This can be attributed to the higher conductivity of the copper. Both types of foams will act to increase mixing and enhance the heat transfer from the substrate walls. In the copper foams, there will also be heat conduction through the foam, so the foam acts as an extended surface for heat transfer. This effect will be considerably smaller in the nickel foams owing to its lower thermal conductivity.

For all the 10 PPI metal foams it can be shown that the heat transfer coefficient increases with fluid flow velocity, suggesting that at higher fluid flows, mixing increases and effectively increases heat transfer. Within each group of foams, there is a spread of volumetric heat transfer coefficients. Younis and Viskanta [75], were able to claim that the variation amongst the volumetric heat transfer coefficients of metal foams is dependent on two main physical parameters, porosity and pore size. The general trend in porous media regarding heat transfer is that an increase in either porosity or pore size should reduce the volumetric heat transfer coefficient [76, 77]. Qualitatively speaking this means that the relative density \((1 - \varepsilon)\), is proportional to the solid effective thermal conductivity and the tortuosity of the flow path and hence the volumetric heat transfer coefficient. Subsequently, the volumetric heat transfer coefficient is proportional to the surface area density and inversely proportional to the pore size. Since the porosity amongst the foams in the present study are essentially constant the relationship between porosity and
Figure 5.7: Experimentally computed volumetric heat transfer coefficient with respect to fluid flow velocity for Ni metal foams with varying heat flux. The experimental uncertainty is ±0.06 m/s in the fluid flow velocity measurements, whereas the maximum uncertainty in the volumetric heat transfer coefficient is 8.73%.

Figure 5.8: Experimentally computed volumetric heat transfer coefficient with respect to fluid flow velocity for Cu metal foams with varying heat flux. The experimental uncertainty is ±0.06 m/s in the fluid flow velocity measurements, whereas the maximum uncertainty in the volumetric heat transfer coefficient is 4.12%.
the heat transfer coefficient cannot be fully determined, and instead an explanation regarding the heat transfer with respect to the pore size will be attempted.

In Figure 5.9 it can be seen that the the pore size has an insignificant effect on the overall heat transfer performance in the nickel foams, suggesting perhaps that porosity is the principle parameter in determining the heat transfer performance. In the copper foams, the pore size has a greater effect on the volumetric heat transfer coefficient. In both foams the porosity determines the tortuosity of the flow path and consequently the degree of mixing, and in both foams the porosity will also determine the effective conductivity, (refer to Section 5.1). However, it is only in the copper foams that the thermal conductivity will play any significant role in the heat transfer. This in turn means that the pore size affects the surface area density of the foam and hence the area available for heat transfer. This pore size is only important in metal foams with relatively high thermal conductivities where conduction through the foam struts allows heat transfer to take place from internal portions of the foam as well as the substrate wall. Referring back to Figures 5.7 and 5.8, the copper foams are able to take advantage of the increased surface area since they have 4 times the thermal conductivity of the nickel foams. Essentially, the lower pore size nickel foams behave like fins, where their low thermal conductivity diminishes their capability to transfer heat across its entire surface area and hence reduce the volumetric heat transfer coefficient.

Alternatively, the volumetric heat transfer definition expressed in Equation (5.16), can be derived to express a heat transfer coefficient based on a per unit surface area (heat transfer coefficient, $h$). Assuming an ideal foam structure and using the surface area density to derive the total surface area of the metal foam channel

$$A_{sf} = \alpha_{sf} H^2 L$$  \hspace{1cm} (5.18)

the surface area heat transfer coefficient based on the energy balance becomes.

$$\frac{\rho C_p \varepsilon A_d V (T_{f,o} - T_{f,i}) \Delta x}{A_{sf} L (T_s(x) - T_f(x))} = h(x) \left[ \frac{W}{m^2 K} \right]$$  \hspace{1cm} (5.19)

Again the heat transfer coefficients are taken as a length averaged value.

$$\bar{h} = \frac{\Delta x}{L} \sum h(x)$$  \hspace{1cm} (5.20)
Figure 5.9: Relationship between the volumetric heat transfer coefficient and pore size from Ni and Cu metal foams for a fluid flow velocity of 0.918 m/s.
Figure 5.10: Experimentally computed heat transfer coefficient with respect to fluid flow velocity for Ni metal foams with varying heat flux. The experimental uncertainty is ±0.06 m/s in the fluid flow velocity measurements, whereas the maximum uncertainty in the heat transfer coefficient is 12.98%.

Figure 5.11: Experimentally computed heat transfer coefficient with respect to fluid flow velocity for Cu metal foams with varying heat flux. The experimental uncertainty is ±0.06 m/s in the fluid flow velocity measurements, whereas the maximum uncertainty in the heat transfer coefficient is 14.41%.
Due to the significantly greater surface area in the smaller pore size foams, the surface area heat transfer coefficients are significantly smaller in magnitude compared to their counterparts for larger pore sizes. Referring to Figures 5.10 and 5.11 the copper foams still outperform the nickel foams by producing twice the amount of heat transfer. What is critical to notice is that at the lower pore sizes, the copper foams are able to produce greater amounts of heat transfer, indicating enhanced mixing mechanisms with less surface area available.

To gain an understanding of the contributions of the relative size of the heat transfer enhancement due to mixing and due to conductivity (the extended surface areas mechanism), the results of the nickel and copper foams are normalized by the results of an empty channel with the same dimensions. To make a valid comparison the resulting heat transfer coefficients of the metal foams are converted to a Nusselt number based on the hydraulic diameter of the channel.

\[
\text{Nu}_H = \frac{h_i H}{k_f} \quad (5.21)
\]

The \(h_i\) term represents the heat transfer coefficient based on the surface area for either the nickel foam, copper foam or the empty channel. It can be seen in Figure 5.12 that the heat transfer in the nickel foam, predominately caused by flow mixing, is about 2 times higher than in an empty channel, whereas with the copper foam the heat transfer is about 7 times greater than an empty channel.

The conventional definition of the Nusselt number for the foam is based on the fiber diameter as this corresponds to the standard correlation for flow past a cylinder or cylinder arrays. The Nusselt number can be rewritten in terms of the the volumetric heat transfer coefficient as follows

\[
\text{Nu}_{d_p} = \frac{h d_f}{k_f} = \frac{h_s f d_f}{k_f} \frac{1}{\alpha_{sf}} = \frac{h_s f d_p^2}{k_f} \quad (5.22)
\]

The above derivation has shown that there is a physical basis for the use of the pore diameter as a length scale in the definition of the Nusselt numbers for foams. The Reynolds number based on the pore diameter is then

\[
\text{Re}_{d_p} = \frac{ud_p}{\nu} \quad (5.23)
\]

The experimental data for heat transfer is often represented with reasonable accuracy by a simple power-law relation. In an attempt to develop a global Nusselt number correlation for the foams the non-dimensional parameter \(Da^{1/2}\) is considered, which is a standard parameter used to characterize the fluid flow in porous mediums. Also, in order to take into consideration the effect thermal conductivity...
Figure 5.12: Heat transfer enhancement with foams of different conductivities. 10PPI Ni and Cu metal foams with a porosity of 0.94

Figure 5.13: Average Nusselt number as a function of the Reynolds-Prandtl-Darcy number for both the Ni and Cu metal foams of varying pore density.
has on the volumetric heat transfer coefficient, a conductivity ratio between the fluid and solid material is also considered. In Figure 5.12, the the volumetric heat transfer coefficient is expressed as an empirical Nusselt number correlation with an $R^2 > 99.0\%$

$$Nu = 19,149Re^{0.227}Pr^{1.794}Da_H^{1.715} \left( \frac{k_s}{k_f} \right)^{0.603}$$

(5.24)

Such a correlation can be used to facilitate the design of future metal foam heat exchangers, where the designer must consider the trade-off between heat transfer by mixing and diminishing effects by low thermal conductive materials.

### 5.3.2 Uncertainty

An example to find the absolute experimental uncertainty in the heat transfer coefficient, is once again determined by following the standard procedure outlined in Taylor [49]. The uncertainty in $\bar{h}$, labeled as $\delta \bar{h}$, is based on the accuracy of the experimental instrumentation, to measure velocity and temperature, but also the uncertainty in porosity determined from Equation (3.19). Assuming negligible uncertainty in the fluid properties, $\bar{h}$ is treated a function of the respective variables

$$\delta \bar{h} = \sqrt{\left( \frac{\partial \bar{h}}{\partial \varepsilon} \delta \varepsilon \right)^2 + \left( \frac{\partial \bar{h}}{\partial u} \delta u \right)^2 + \left( \frac{\partial \bar{h}}{\partial (\Delta T_f)} \delta T \right)^2 + \left( \frac{\partial \bar{h}}{\partial (\Delta T_{sf})} \delta T \right)^2}$$

(5.25)

### 5.4 Limitations of One-Dimensional Model

For both the conduction and forced convection analysis it was assumed that the effects of heat transfer in the vertical direction where negligible. However, based on the analysis and what is known about the heat transfer mechanisms in metal foams, this is not the case. Assuming that the foam can be described by the idealized period structure mentioned in Chapter 3, the vertical dependence of heat transfer can be characterized by a single vertical cylinder modeled as a fin. The following analysis will attempt to show numerically the concept of the extended surface area mechanisms of high thermal conductive foams, but also how the channel height can affect the heat transfer.

In order to apply the standard correlations regarding heat transfer from finned surfaces the following assumptions are considered:

- The cross-section of the fin is constant, where the dimensions of the cylinder correspond to the
dimensions of a single foam strut. Thus, the fin diameter equals the fiber diameter, $d_{fin} = d_{fiber}$.

- Negligible heat loss occurs at the fin tip, hence an adiabatic boundary condition is taken at the mid-height of the channel $L = H/2$.
- The temperature at the fin base is the wall temperature, $T_{base} = T_{wall}$
- The temperature of the surroundings is taken as the bulk fluid temperature, $T_{\infty} = T_f$
- The heat transfer coefficients governing convection from the fin surface are taken from Figures 5.10 and 5.11 based on a per unit surface area.

Applying the above assumptions, the temperature distribution of the fin along the vertical axis is taken from Cengal [75] as

$$T(y) - T_f \over T_{wall} - T_f = \cosh m (H/2 - y) \over \cosh m (H/2)$$

(5.26)

where the dimensions of the cylinder result in a fin parameter as

$$m = \sqrt{4h \over k_s d_f}$$

(5.27)

To compare the various metal foams simultaneously, the non-dimensional terms are expressed for temperature and height

$$\theta(Y) = T(y) - T_f \over T_{wall} - T_f$$

(5.28)

$$Y = y \over H$$

(5.29)

Figure 5.14 shows the non-dimensional temperature distribution along the height of the channel based on the experimental temperature results. Based on the curvature of the profiles, it can immediately be argued that the isothermal temperature assumption is not entirely valid as the foam temperature changes as it approaches the center of the channel height. Comparing the plots, it can be noticed that the nickel foams experience a greater temperature gradient, varying as much as 16% from the temperature at the substrate wall, whereas the copper foams vary by a maximum of 7%. This reduction in temperature amongst the copper foams can be attributed to their higher thermal conductivity, allowing them to dissipate heat at a greater rate. These gradients become more pronounced at higher fluid flow velocities where convection effects increase and cool the strut. As a result of this temperature drop along the length of the strut, heat transfer becomes less because of the decreasing temperature difference $T(y) - T_f$, from the strut to the fluid.
\[ \theta(Y) = \frac{T(y) - T_f}{T_{wall} - T_f} \]

\[ Y = \frac{y}{H} \]

Ni @ \( u = 0.919 \) m/s
Ni @ \( u = 3.674 \) m/s
Cu @ \( u = 0.919 \) m/s
Cu @ \( u = 3.674 \) m/s

Figure 5.14: Temperature profiles along the height of channel for Ni and Cu metal foams with (a) 10PPI and (b) 40PPI pore densities. The profiles correspond to experimental results at a heat flux of 1711 W/m² for varying fluid flow velocities.
Referring back to the extend surface area mechanism defined in the previous section, the copper foams are able to achieve greater heat transfer because of the higher temperature differences between the strut surface and the surrounding fluid. Effectively this means that copper foams are able to utilize their surface area more efficiently than the nickel foams. Apart from the influence of thermal conductivity, these temperature variations can also be effected by the the height of the channel. In order to determine the effect of temperature drop along the length of the strut with respect to its capability to transfer heat, the strut efficiency $\eta_s$, is shown in Figure 5.15 and computed from Equation (5.30)\cite{78}.

$$
\eta_s = \frac{\tanh m(y)}{m(y)}
$$

(5.30)

It is evident that the overall efficiency of the copper foam is greater than the nickel foams, indicating that in fact that surface area mechanism performs better in the copper foams. It can also be shown that the efficiency of the strut decreases as the height of the channel increases, meaning that regardless of how high the thermal conductivity of the foam material is the extended surface area mechanism will gradually diminish as the temperature gradient along the strut decreases. To explain this concept even further the performance of the metal foams with respect to height can be characterized by the effectiveness. The following figure compares the strut effectiveness $\xi_s$\cite{78} of the nickel and copper foams with change in channel height.

$$
\xi_s = \frac{A_{fin}}{A_{base}} \eta_s = \frac{A(y)}{d_f} \eta_s
$$

(5.31)

Effectiveness values greater than 1 indicate that the metal foams are enhancing heat transfer from their surfaces, justifying their use in heat exchanger applications. Several comments can be made in regards to the the effectiveness of the foams. Despite a decreasing efficiency with length, the effectiveness of the fin actually increases, as the surface area of the strut increases with length. However, at some critical length the effectiveness plateaus and the performance of the foam no longer increases. This critical length is more crucial in the smaller pore size foam, where length changes result in substantial increases in surface area. What should be kept in mind is that the theoretical effectiveness of the nickel foams indicate that the 40PPI foam should out perform the 10PPI foam significantly. However, in actuality from the experimentation it was shown that the 40PPI foam provided little or no increase in heat transfer performance. This analysis considers a single strut, and even through this may be a valid assumption of the local heat transfer characteristics the global effects of multiple struts can significantly alter the geometric parameters that characterize the fin equations and subsequently predict different performance potentials.
Figure 5.15: The efficiency of a metal foam strut with respect to the strut length. The curves correspond to experimental results at a heat flux of 1711.42 W/m² for strut diameters corresponding to 0.095 mm for Ni foams and 0.102 mm for Cu foams.

Figure 5.16: Comparison of the Ni and Cu strut effectiveness for varying pore density foams. The curves correspond to experimental results at a heat flux of 1711 W/m².
By treating the heat transfer characteristics of metal foams as extended fin surfaces, several conclusions can be drawn when considering the design of metal foam heat exchangers.

- The thermal conductivity $k_s$ of the foam should be as high as possible such that the efficiency of the foam is maximized. This will ensure the extended surface area heat transfer enhancement will be used to its full potential.

- The ratio between the struts perimeter and cross-sectional area should be as high as possible. This parameter is satisfied with large pore density foams, where the strut diameter decreases with a decrease in pore size.

- The geometry of the heat exchanger plays a significant role, specifically the length across which heat transfer occurs. Depending on the application of the heat exchanger, and in conjunction with the conductivity of the foam material, desired heat transfer capabilities can be achieved by altering the combination of geometry to foam material.
CHAPTER 6. COMPACT WATER-AIR METAL FOAM HEAT EXCHANGER

Chapter 6

Compact Water-Air Metal Foam Heat Exchanger

6.1 Introduction

The purpose of a heat exchanger is to transfer heat from one fluid (either a gas or a liquid) to another. Compact heat exchangers have a significantly greater surface area per unit volume than conventional heat exchangers types. In convectional designs, different tubular and plate configurations are considered in order attain large heat transfer surfaces areas. One of the major draw backs to current designs are their high costs, due to the effort and complexity involved in manufacturing them. One way of increasing the exchange surface area is to replace the fins of a conventional exchanger by a porous structure, where exchange surface areas can reach much higher values. However, this increase amplifies the pressure drop of the fluid circulating in the porous matrix. It is therefore necessary to determine the optimal parameters of the porous medium in order to maximize the heat transfer with regards to the pressure drop. The aim of this last section is to demonstrate the use of metal foam materials in heat exchanger applications by fabricating a two-fluid compact metal foam heat exchanger.
6.2 Experimental Apparatus and Procedure

6.2.1 Fabrication of Heat Exchanger

One of the many objectives in this research is to demonstrate the applicability of thermal spray coating technology in manufacturing compact heat exchangers. Here, the multiple fin and tube conception used in current designs will be replaced by metal foams with thermal sprayed skins. The idea is to try and manufacture a heat exchanger of equal performance at a more cost-effective means. For demonstrative and simplicity purposes, a double-pipe heat exchanger design will be fabricated where air and water are used as the two working fluids. Copper foam with a thickness of 20 mm and a pore density of 40 PPI (Dalian Thrive Mining Co. Ltd, Dalian, China) was cut using an electric-discharge machine (AD325L CNC Wire EDM, Sodnick, Japan) to the dimensions of 76.2 mm \( \times \) 20 mm \( \times \) 20 mm \((L \times W \times H)\). At the center of the square face a 9.525 mm (0.375 in.) diameter hole was bored through the length of the foam in order to insert a 101.6 mm (4 in.) long, 9.525 mm (0.375 in.) diameter copper tube. To ensure maximum attainable heat transfer between the struts of the foam and the outer surface of the copper tube, a Sn-alloy soldering paste (Loctite RP15, Henkel AG and Company, Düsseldorf, GER) was applied to bond the two components together. Two metal sleeves made out of 0.25 mm (0.01 in.) thick stainless steel sheets (multipurpose stainless-steel Type 304, McMaster-Carr, Princeton, NJ) were then fabricated to slip on to the outer surface of the foam, where 9.525 mm (0.375 in.) pipe-to-compression fittings were welded on such that duct work for the air supply line can be attached. On the copper tubes two more 9.525 mm (0.375 in.) compression fitting adapters were attached, in order to connect duct work for the water supply lines, Figure 6.1 (a). The entire surface of the heat exchanger is then thermal sprayed with a copper skin using the same foam preparation procedure and thermal spray parameters outlined in Chapter 2, Figure 6.1 (b). To measure the performance improvement of the metal foam heat exchanger, a similar double-pipe heat exchanger was made with out foam The plain heat exchanger had similar dimensions where a stainless steel, square cross-sectional tube (multipurpose stainless-steel Type 304, McMaster-Carr, Princeton, NJ) was used for the outside shell. Refer to Appendix A for schematics of the heat exchanger design.

6.2.2 Experimental Apparatus

The experimental apparatus fabricated to test the metal foam heat exchanger comprises of two circulatory loops, one for each working fluid, air and water. For this experimental set-up a parallel flow arrangement will be tested and analyzed where air passes through the foam annulus and water through
Figure 6.1: Double-pipe compact metal foam heat exchanger (a) before thermal spray and (b) completed with thermal sprayed copper.
the copper tubing. The water reservoir is a stainless steel drum filled with tap water. Three 9.525 mm (0.375 in.) compression-to-pipe adapters are fitted into the drum to connect plastic tubing for the water supply lines. Referring to Figure 6.2, the bottom line (1) feeds the water into the transfer pump (Model 360, Pony Pump Corporation, Los Angeles, CA), from which it pumps the water downstream at approximately 19 L/min to an intersection (2). Attached to the opposite ends of the intersection are two 9.525 mm (0.375 in.) bonnet needle valves, one leading further downstream to the heat exchanger (3) and the other back into the stainless steel drum (4). By adjusting the two valves accordingly, the flow rate of water exerted by the pump can be controlled. The second valve acts as a by-pass loop, in which the accumulated water gathered in the pipe lines can be drawn back into the water reservoir when the first valve is adjusted. The entire water reservoir is placed on top of a portable hot plate (Model HP-10C, Salton Appliances Corporation, Dollard-des-Ormeaux, QC) to heat the water. The power to the hot plate is adjusted using a variable transformer, which in turn can control the water temperature. The flow rate of water is measured by a rotameter (Model 7520, King Instrument Corporation, Garden Cove, CA) for a flow rate range of 0.0 GPM to 2.0 GPM, accurate to within ±4% of FS. The water then continues onto the inlet of the heat exchanger, where it is met by a 9.525 mm (0.375 in.) tee-junction fitting attached to the copper pipe of the heat exchanger. The water exits the heat exchanger via the same copper pipe, which in turn is connected to another 9.525 mm (0.375 in.) tee-junction which carries the water back into the water reservoir (5).

Air provided by the laboratory is driven through a pressure regulator attached to a globe valve, thus when adjusted, can control the amount of air released to the heat exchanger. The mass flow rate is measured by an electronic gas mass flowmeter (Model FMA1842, Omega Company, Stamford, CT) for a flow rate range of 0.0 L/min to 100.0 L/min, accurate to within ±1.5% of full scale (FS). The air then enters the heat exchanger via ductwork that is mated to a 9.525 mm (0.375 in.) tee-junction fitting. The direction of air flow is from top to bottom as in Figure 6.4 and progresses downstream where it exits to the surroundings. At all inlets and outlets tee-junctions, 13 mm (0.5 in.) Type K pipe plug thermocouple probes with a 304 SS sheath of 6.4-mm- diameter were fixed into the fittings, such that when the cross-flow passes through the fitting the probes come into direct contact with the fluid. The standard error associated with the thermocouples are ±0.1°C. The thermocouples are connected to a 2-channel handheld digital thermometers (Model HH501AJ, Omega Company, Stamford, CT) with an accuracy of ±5°C of the reading and a resolution of ±0.01°C. To prevent any extraneous heat loss to the surroundings the entire test sample is covered using 38.1mm (1.5 in.) thick fiberglass insulation with an aluminized outer surface, having an average thermal conductivity of \( k_i = 0.038 \text{ W/m K} \) (Micro-Flex,
Figure 6.2 Schematic of the compact metal foam heat exchanger apparatus showing the instrumentation and the air-water fluid loop configurations.
The experimental uncertainty associated with the temperature measurements can be determined by using the procedure outlined in Chapter 2 for data acquisition systems. Combining the uncertainty associated with the thermometers and thermocouples the uncertainty in the temperature measurements is found to be $\pm 5.5^\circ C$. A typical experimental run started off by adjusting the the variable transformer to obtain a constant water temperature. Then, by adjusting the globe valves the water is driven through the heat exchanger at a constant flow rate. For a given constant flow rate of water at a constant inlet temperature the flow rate of air was varied from 5 L/min to 90 L/min. The experimental procedure is repeated for flow rates of water of 0.3 GPM, 0.6 GPM, 0.8 GPM and 0.9 GPM all maintained at the same inlet temperature. All measurements are recorded under steady-state conditions where the temperatures at all the inlets and outlets for both fluids have changed less then $5^\circ C$ in 10 minutes. Experimental temperature results can be found in the Appendix E.

To account for the pressure drop across the foam section, pressure ports were attached at the inlet and outlet. The pressure was measured using a differential pressure transducer (Model PX160, Omega Company, Stamford, CT) for a differential pressure range of 0.000 psi to 1.000 psi with an accuracy of $\pm 0.25\%$ FS. A digital process panel meter (Model DP24-E, Omega Company, Stamford, CT) calibrated for a 1 to 6 VAC input, with an accuracy of $\pm 1$ LSD (least significant digit) was used to display the pressure reading. All the pressure measurements were performed independently of the heat transfer experiments.

### 6.3 Thermal and Hydraulic Performance

The theory underlining the heat transfer characteristics for the metal foam heat exchanger will follow the standard correlations used in heat exchanger analysis described by Incropera and DeWitt [31]. In order to derive the appropriate performance indexes presented in the subsequent chapter, an understanding of the convective heat transfer occurring in the metal foam is required. Several assumptions regarding the heat transfer must be presented in order to engage in the appropriate energy balance.

- The heat transfer area acting between the air and the water is taken to be the same, the outside surface area of the tube.

- The conduction resistance from the soldering paste and the wall of the tube are neglected.
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Figure 6.3: Heat exchanger apparatus with instrumentation.

Figure 6.4: Detailed view of heat exchanger test section, showing the placement of thermocouples (the insulation has been removed for clarity). The flow direction of air is indicated from inlet (top) to outlet (down), while the water flow is from inlet (left) to outlet (right).
- Heat losses from the exterior wall (through the insulation) to the surrounding are considered negligible, hence all the heat transferred is from the water to the air.

- The fluid properties are evaluated at the bulk mean temperature for the respective fluids.

- The heat transfer coefficient is considered constant along the length.

- In the presence of the foam, the overall surface efficiency is assumed to be $\eta_s = 1$.

Based on these assumptions the overall heat transfer between the air and water in a parallel-flow configuration is given by Incopera and Dewitt [81] as

$$Q = UA_s \Delta T_{lm} \quad (6.1)$$

where heat transfer occurs along the same surface area, such that the overall heat transfer coefficient is found to be

$$\frac{1}{U} = \frac{1}{h_{air}} + \frac{1}{h_{water}} \quad (6.2)$$

The average temperature difference between the fluids is given by the log mean temperature relation.

$$\Delta T_{lm} = \frac{\Delta T_{outlet} - \Delta T_{inlet}}{ln(\Delta T_{outlet}/\Delta T_{inlet})} \quad (6.3)$$

Because of the very high specific heat of water compared to the air, the temperature drop of water across the length of the tube is quite small and hence considered isothermal. Combined with a very small tube length the heat transfer coefficient for the water side is taken from the correlation for the thermal entry region of circular tubes for an isothermal boundary condition recommended by Hausen [82].

$$Nu = \frac{h_{water}D}{k_{water}} = 3.66 + \frac{0.0688 \left(\frac{D}{L}\right) Re_D Pr}{1 + 0.04 \left(\frac{D}{L}\right) Re_D Pr}^{2/3} \quad (6.4)$$

Evaluating Equation (6.1) and (6.4) using the experimental temperature results the heat transfer coefficient for the air passing through the foam can be found. Figure 6.5 compares the heat transfer to the air by means of the foam and without the foam. The velocity of air entering the foam channel is evaluated at the pore scale, where the cross-sectional area is taken as the area occupied by void. This velocity definition was expressed by Equation (4.6), and is presented here again as

$$u_p = \frac{Q}{A_c \varepsilon} \quad (6.5)$$
However, it should be mentioned that the absolute magnitude of the values computed in Figure 6.5 for the foam are somewhat exaggerated. In actuality, the surface area upon which heat transfer occurs in the foam is substantially greater than the outer surface area of the tube, which would inherently produce lower values at any given flow rate of air. With this in mind, the intention of the analysis is to show the enhancement of heat transfer to the air due to the presence of the foam compared to just a hollow shell. Referring back to the Figure 6.5 it becomes evident that the foam enhances the heat transfer to the air from the water by almost 3 times what is capable from the hollow shell. This heat transfer enhancement can again be attributed to both the turbulent mixing and the extended surface area mechanism involved in metal foam heat transfer.

Heat transfer enhancements in heat exchangers are usually accompanied by increased pressure drops, and metal foams heat exchangers are no exception. Thus the gain in convective performance must be weighed against the energy required to operate the system to determine if any trade-off is worth the cost. The required energy is measured by the *pumping power* $W$, and was computed for both the metal foam and plain tube heat exchanger using Equation (6.6) for various air flow velocities.

$$W = \Delta P Q$$  \hspace{1cm} (6.6)

A common means to measure the heat convection effectiveness is the *thermal resistance* $R_{th}$, where a lower thermal resistance facilitates the heat flow through the heat exchanger. Thermal resistance for a convective scenario is defined as

$$R_{th} = \frac{1}{h_{air} A_s}$$  \hspace{1cm} (6.7)

Using the derived convective heat transfer coefficients in Figure 6.5 and the experimental pressure drop results, Figure 6.6 compares the pumping power relation to the thermal resistance of both the metal foam and plain tube heat exchanger. The optimal design is that which minimizes the magnitude of both thermal resistance and pumping power. It is clear from the plot that both heat exchanger types have their advantages and disadvantages. For flow velocities greater than 1 m/s, the metal foam heat exchanger achieves lower thermal resistances than the plain tube heat exchanger. However, overall the plain tube heat exchanger requires less power to operate. For comparative purposes the maximum obtainable velocity in both heat exchanger configurations was 4.75 m/s, but the metal foam heat exchanger requires 14.17 W of power compared to 0.77 W of the plain tube. This translates into a 173% increase in power for only a 133% improvement in thermal performance. A challenge arises whether the gain in thermal performance justifies the substantial increase in power input.
Figure 6.5: Comparison of the convective heat transfer coefficient on the air side with the foam and without the foam in a compact heat exchanger. The experimental uncertainty in the fluid flow velocity measurements is $\pm0.06 \text{ m/s}$, where the maximum uncertainty in the heat transfer coefficient is 9.58% and 7.24% for the foam and tube heat exchanger respectively.

Figure 6.6: Comparison of required pumping power against the corresponding thermal resistance for both the metal foam and plain tube heat exchangers.
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6.4 Design Correlations

In reality, heat exchangers are selected to accomplish or meet certain heat transfer tasks. Usually, the goal is to either heat or cool the fluid at a known mass flow rate and temperature, to a desired temperature. But in order to determine the performance of a particular heat exchanger, and hence assist in the selection process, certain intrinsic parameters about the heat exchanger are needed. The rate of heat transfer in a prospective heat exchanger is simply defined by Equation (6.1). In the previous section the entire analysis was performed using experimental results to derive the unknown parameters, however experimentation is a luxury not often afforded due to time and cost. Thus, non-dimensional design correlations are often relied on to determine such unknown parameters. The objective of this section is to present commonly used methods to express the experimental results such that they can be use to evaluate the performance of metal foam heat exchangers for future design considerations.

Firstly, the heat transfer coefficient across the foam side can be expressed by the Colburn-factor, given by Incropera and Dewitt [81] as

\[ j = \frac{h}{\rho u p c p} Pr^{2/3} \]  
(6.8)

The characteristic length used to evaluate the Reynolds number is based on the hydraulic diameter, where the cross-sectional area is taken as the square foam cross-section minus the annulus of the tube center. In Figure 6.7 the experimental results are non-dimensionalized to represent the \( j \) factor as a function of the Reynolds number.

Performing a linear regression analysis, an empirical power law expression can be derived to express the experimental results for the convective heat transfer coefficient acting on the foam side with an \( R^2 > 96.5\% \), for fluid flow velocity with \( Re \geq 10^3 \).

\[ j = 1.851 Re^{-0.329} D_{hyd} \]  
(6.9)

By using Equation (6.9), the overall heat transfer coefficient needed to solve Equation (6.1) is possible, with the exception of the log mean temperature difference. A common problem encountered in heat exchanger analysis is the determination of the rate of heat transfer and the outlet temperatures of the hot and cold fluids, when a given mass flow rate and inlet temperature is prescribed. However, if only the inlet temperatures are known, an iterative method is required to solve for the unknown parameters. This is where the effectiveness-NTU method presents itself as a useful tool to avoid the cumbersome calculations. The effectiveness is defined by Incropera and DeWitt [81] as the ratio of the actual heat
CHAPTER 6. COMPACT WATER-AIR METAL FOAM HEAT EXCHANGER

Figure 6.7: Colburn-factor correlation for metal foam heat exchanger.

Figure 6.8: Effectiveness of a parallel-flow metal foam heat exchanger.
transfer for a heat exchanger to the maximum possible heat transfer rate.

\[ \varepsilon = \frac{Q}{Q_{\text{max}}} = \frac{C_h (T_{h,\text{inlet}} - T_{h,\text{outlet}})}{C_{\text{min}} (T_{h,\text{inlet}} - T_{c,\text{inlet}})} \]  \hspace{1cm} (6.10)

The number of transfer units, NTU, is defined as

\[ NTU = \frac{UA_s}{C_{\text{min}}} \]  \hspace{1cm} (6.11)

In Figure 6.8, the experimental results are plotted accordingly to derive an expression to relate the effectiveness as a function of the NTU, where \( c = C_{\text{min}}/C_{\text{max}} \), is the heat capacity ratio of the hot and cold fluids. Due to the configuration of the current experimental set-up, the heat capacity ratios attainable were in a range close to zero. Hence, an average heat capacity ratio of 0.1 was taken to represent all the experimental data. It was found that the standard effectiveness correlation for a parallel-flow configuration [83] represents the data sufficiently with a maximum deviation of 5.7%.

\[ \varepsilon = 1 - \exp \left( -NTU (1 + c) \right) \]  \hspace{1cm} (6.12)

It is expected that by using the empirical correlations defined by Equations (6.9) and Equation (6.12), all the necessary parameters needed to predict the performance of the metal foam heat exchanger are possible.

### 6.5 Uncertainty

To find the absolute experimental uncertainty in the heat transfer coefficient, the procedure outlined by Taylor [49] is used once again. The uncertainty associated with the heat transfer coefficient for the air side is based on the accuracy of the experimental instrumentation used to measure temperature and the flow rate of the passing fluids. First, the uncertainty in the heat transfer from the water to the air is expressed as a function of the air fluid velocity and the temperature increase of the air, such that

\[ \delta Q = \sqrt{\left( \frac{\partial Q}{\partial u} \delta u \right)^2 + \left( \frac{\partial Q}{\partial (\Delta T)} \delta T \right)^2} \]

where the uncertainty in the fluid properties and cross-sectional area are negligible. Thus, expressing the overall heat transfer coefficient as a function of the variables in Equation (6.1), the uncertainty is
found as

\[ \delta U = \sqrt{\left( \frac{\partial U}{\partial Q} \delta Q \right)^2 + \left( \frac{\partial U}{\partial (\Delta T_{lm})} \delta T \right)^2} \]

Again is assumed that the uncertainty associated with the fluid properties and the surface area are negligible. Also, the uncertainty in the heat transfer coefficient derived for the water side is considered negligible since it is obtained from a correlation.
Chapter 7

Conclusions

This research investigation was concerned with the fabrication and experimental testing of thermal sprayed metal foam heat exchangers. The hydraulic and thermal characteristics were studied by measuring the pressure drop and temperature distribution along the length of the metal foam channels. The following is a list of the results achieved in this investigation and several qualitative comments that can be deduced from the analysis:

1. The method used in this study to apply thermal skins onto metal foams is capable of being applied to nickel and copper foams with different pore sizes.

2. The pressure drop along the length of the foam can be accurately described by the Dupuit-Forchheimer equation. Fluid flow resistance is measured by the permeability where the inertial-coefficient measures the flows deviation from the linear Darcy regime. Fluid flow characteristics are highly depend on the geometry of the porous medium. Decreasing the pore diameter decreased the permeability and increased the inertia coefficient which led to higher pressure drops.

3. Based on the experimentally obtained flow parameters, friction factor correlations have been derived to predict the pressure drop in 10PPI (Equation [4.20]) and 40PPI (Equation [4.21]) metal foams. The correlations have a maximum deviation of 11% with respect to the experimental results.

4. It was determined that both the nickel and copper foams enhanced heat transfer by turbulent mixing of the fluid flow due to tortuous paths. It was also determined that heat transfer enhancement occurs as a result of the increased surface area provided by the struts of the foam, acting as extended surfaces for heat transfer. However, the copper foams were able to out perform the nickel foams by nearly 5 times as a result of their higher thermal conductivity.
5. It was concluded that the heat transfer behavior within the metal foams can be compared to fins, where increases in heat transfer occur when the foam conductivity increases or the foam thickness decreases. From a design perspective, there is basically a trade-off between the available surface area and the amount of capable conduction offered by a specific foam material. In the case of low thermal conductive materials, it may be beneficial to use a larger pore size foam, and achieve lower friction losses than use a smaller pore size foam and achieve the same or even less heat transfer with greater friction losses.

6. Using a one-dimensional model volumetric heat transfer coefficients were computed as a function of the fluid flow velocity. Based on these computed values and the geometric parameters characterizing the metal foam structure, the convective performance of the both the nickel and copper metal foams can be predicted with the developed Nusselt number correlation (Equation (5.24)).

7. The performance of a compact water-to-air metal foam heat exchanger was tested against a plain double pipe heat exchanger. It was determined that significant increases in heat transfer are obtainable but at the expense of substantial increases in pressure drop. Hence a critical evaluation must be made to determine if the added increases in heat transfer out-weight the added cost in pumping power.

8. From the water-to-air experimentation relevant correlations were developed to predict the convective heat transfer performance (Equation (6.9)) and the mean temperatures (Equation (6.12)) at the coolant outlets of the metal foam heat exchanger. Such correlations can be used for future design considerations and selection of metal foam heat exchangers.
As this work approaches its conclusion, I reflect upon my efforts, and the following words come to mind:

“Most of the fundamental ideas of science are essentially simple, and may, as a rule, be expressed in a language comprehensible to everyone.”

“It’s not that I’m so smart, it’s just that I stay with problems longer.

~Albert Einstein

Ερωτικά προϊόντα βιοτών φύσης
Φως για τη ζωή είναι κυρίως η εργασία

~Πιθανός, Ολυμπιονικός

Nicholas Tsolas
M.A.Sc Candidate
Department of Mechanical and Industrial Engineering
University of Toronto
References


Appendix A

Schematics

The following schematics are detailed drawings of the different types of metal foam heat exchangers fabricated for use in this study. The first drawing corresponds to the stainless steel metal bracket used to attach compression fittings for thermocouple probes and ductwork, as described in Chapter 2. The second drawing corresponds to the double-pipe metal foam heat exchanger used in Chapter 6. All units are in inches except wherever specified.
Figure A.1: Stainless steel bracket drawing.
Figure A.2: Water-air compact metal foam heat exchanger drawing.
Appendix B

Image Analysis Data

The measurements collected during the image analysis process described in Chapter 3 is given in this appendix in tabular form. The end result of this data can be found summarized in Table 3.5. The data is organized according to the geometric feature measured and separated based on a particular foam material and pore density. The samples used for image analysis are coded to assist in distinguishing between foam samples that were impregnated with epoxy and those that were not. Fiber and porosity measurements conducted by image analysis were performed on the same samples, whereas the pore sizes were performed on samples that were not impregnated with epoxy. Table B.1 outlines the description of each sample.

<table>
<thead>
<tr>
<th>Sample</th>
<th>Material</th>
<th>PPI</th>
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<tbody>
<tr>
<td>Ae</td>
<td>Ni</td>
<td>40</td>
</tr>
<tr>
<td>Be</td>
<td>Ni</td>
<td>40</td>
</tr>
<tr>
<td>Ce</td>
<td>Cu</td>
<td>40</td>
</tr>
<tr>
<td>De</td>
<td>Cu</td>
<td>40</td>
</tr>
<tr>
<td>Ee</td>
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<td>10</td>
</tr>
<tr>
<td>Fe</td>
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<td>10</td>
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<tr>
<td>Ge</td>
<td>Cu</td>
<td>10</td>
</tr>
<tr>
<td>He</td>
<td>Cu</td>
<td>10</td>
</tr>
</tbody>
</table>

Table B.1: Description of foam samples used to measure geometric features. Samples that have an ‘e’ in their designation signify it was an epoxy covered sample.

Measurements taken to determine the porosity by the weighing method were conducted on separate samples. The Tables B.18- B.21 indicate the dimensions and the mass of the particular foam sample used to calculate porosity.
### Table B.2: Fiber measurements for sample Ae

<table>
<thead>
<tr>
<th></th>
<th>Ae-1</th>
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<th>Ae-3</th>
<th>Ae-4</th>
<th>Ae-5</th>
</tr>
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<tbody>
<tr>
<td>$A_j$ ($mm^2$)</td>
<td>0.004</td>
<td>0.011</td>
<td>0.005</td>
<td>0.012</td>
<td>0.005</td>
</tr>
<tr>
<td>$P_j$ ($mm$)</td>
<td>0.434</td>
<td>0.525</td>
<td>0.498</td>
<td>0.585</td>
<td>0.482</td>
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<tr>
<td>$d_j$ ($mm$)</td>
<td>0.041</td>
<td>0.080</td>
<td>0.042</td>
<td>0.082</td>
<td>0.041</td>
</tr>
<tr>
<td>$d_o/d_f$</td>
<td>50.7%</td>
<td>51.1%</td>
<td>51.2%</td>
<td>45.4%</td>
<td>39.0%</td>
</tr>
</tbody>
</table>

### Table B.3: Fiber measurements for sample Be

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<th>Be-3</th>
<th>Be-4</th>
<th>Be-5</th>
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</thead>
<tbody>
<tr>
<td>$A_j$ ($mm^2$)</td>
<td>0.005</td>
<td>0.012</td>
<td>0.004</td>
<td>0.013</td>
<td>0.005</td>
</tr>
<tr>
<td>$P_j$ ($mm$)</td>
<td>0.422</td>
<td>0.550</td>
<td>0.472</td>
<td>0.613</td>
<td>0.479</td>
</tr>
<tr>
<td>$d_j$ ($mm$)</td>
<td>0.044</td>
<td>0.099</td>
<td>0.034</td>
<td>0.082</td>
<td>0.039</td>
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<td>$d_o/d_f$</td>
<td>44.3%</td>
<td>41.1%</td>
<td>29.9%</td>
<td>40.0%</td>
<td>43.7%</td>
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</tbody>
</table>

### Table B.4: Fiber measurements for sample Ce

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<tbody>
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<td>$A_j$ ($mm^2$)</td>
<td>0.004</td>
<td>0.016</td>
<td>0.006</td>
<td>0.012</td>
<td>0.005</td>
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<tr>
<td>$P_j$ ($mm$)</td>
<td>0.424</td>
<td>0.522</td>
<td>0.421</td>
<td>0.461</td>
<td>0.439</td>
</tr>
<tr>
<td>$d_j$ ($mm$)</td>
<td>0.041</td>
<td>0.124</td>
<td>0.053</td>
<td>0.106</td>
<td>0.042</td>
</tr>
<tr>
<td>$d_o/d_f$</td>
<td>33.4%</td>
<td>50.3%</td>
<td>46.9%</td>
<td>54.1%</td>
<td>43.6%</td>
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### Table B.5: Fiber measurements for sample De

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<th>De-4</th>
<th>De-5</th>
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</thead>
<tbody>
<tr>
<td>$A_j$ ($mm^2$)</td>
<td>0.004</td>
<td>0.014</td>
<td>0.005</td>
<td>0.010</td>
<td>0.005</td>
</tr>
<tr>
<td>$P_j$ ($mm$)</td>
<td>0.442</td>
<td>0.588</td>
<td>0.398</td>
<td>0.548</td>
<td>0.444</td>
</tr>
<tr>
<td>$d_j$ ($mm$)</td>
<td>0.043</td>
<td>0.033</td>
<td>0.046</td>
<td>0.075</td>
<td>0.046</td>
</tr>
<tr>
<td>$d_o/d_f$</td>
<td>46.1%</td>
<td>61.7%</td>
<td>31.7%</td>
<td>50.2%</td>
<td>44.3%</td>
</tr>
<tr>
<td></td>
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<td>Ee-3</td>
<td>Ee-4</td>
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</tr>
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<td>$d_o$</td>
<td>$d_f$</td>
</tr>
<tr>
<td>Area, $A_j$ (\text{mm}^2)</td>
<td>0.053</td>
<td>0.143</td>
<td>0.050</td>
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<td>0.076</td>
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<tr>
<td>Perimeter, $P_j$ (\text{mm})</td>
<td>1.547</td>
<td>1.856</td>
<td>1.533</td>
<td>2.032</td>
<td>1.737</td>
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<td>Hydraulic diameter, $d_j$ (\text{mm})</td>
<td>0.151</td>
<td>0.309</td>
<td>0.130</td>
<td>0.365</td>
<td>0.175</td>
</tr>
<tr>
<td>Hallowness ratio, $d_o/d_f$</td>
<td>49.0%</td>
<td>35.8%</td>
<td>47.3%</td>
<td>48.9%</td>
<td>39.5%</td>
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Table B.6: Fiber measurements for sample Ee

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<tr>
<td></td>
<td>$d_f$</td>
<td>$d_o$</td>
<td>$d_f$</td>
<td>$d_o$</td>
<td>$d_f$</td>
</tr>
<tr>
<td>Area, $A_j$ (\text{mm}^2)</td>
<td>0.053</td>
<td>0.148</td>
<td>0.066</td>
<td>0.173</td>
<td>0.059</td>
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<tr>
<td>Perimeter, $P_j$ (\text{mm})</td>
<td>1.704</td>
<td>1.859</td>
<td>1.689</td>
<td>2.287</td>
<td>1.661</td>
</tr>
<tr>
<td>Hydraulic diameter, $d_j$ (\text{mm})</td>
<td>0.125</td>
<td>0.318</td>
<td>0.158</td>
<td>0.303</td>
<td>0.143</td>
</tr>
<tr>
<td>Hallowness ratio, $d_o/d_f$</td>
<td>39.4%</td>
<td>52.1%</td>
<td>48.2%</td>
<td>46.3%</td>
<td>48.6%</td>
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Table B.7: Fiber measurements for sample Fe

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<td>$d_o$</td>
<td>$d_f$</td>
<td>$d_o$</td>
<td>$d_f$</td>
</tr>
<tr>
<td>Area, $A_j$ (\text{mm}^2)</td>
<td>0.053</td>
<td>0.176</td>
<td>0.072</td>
<td>0.164</td>
<td>0.068</td>
</tr>
<tr>
<td>Perimeter, $P_j$ (\text{mm})</td>
<td>1.738</td>
<td>2.157</td>
<td>1.551</td>
<td>2.155</td>
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<tr>
<td>Hydraulic diameter, $d_j$ (\text{mm})</td>
<td>0.122</td>
<td>0.328</td>
<td>0.187</td>
<td>0.304</td>
<td>0.162</td>
</tr>
<tr>
<td>Hallowness ratio, $d_o/d_f$</td>
<td>37.3%</td>
<td>61.5%</td>
<td>53.4%</td>
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Table B.8: Fiber measurements for sample Ge

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<th>He-3</th>
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<td>$d_o$</td>
<td>$d_f$</td>
<td>$d_o$</td>
<td>$d_f$</td>
</tr>
<tr>
<td>Area, $A_j$ (\text{mm}^2)</td>
<td>0.069</td>
<td>0.177</td>
<td>0.070</td>
<td>0.165</td>
<td>0.070</td>
</tr>
<tr>
<td>Perimeter, $P_j$ (\text{mm})</td>
<td>1.576</td>
<td>2.166</td>
<td>1.657</td>
<td>1.840</td>
<td>1.583</td>
</tr>
<tr>
<td>Hydraulic diameter, $d_j$ (\text{mm})</td>
<td>0.174</td>
<td>0.327</td>
<td>0.169</td>
<td>0.359</td>
<td>0.177</td>
</tr>
<tr>
<td>Hallowness ratio, $d_o/d_f$</td>
<td>53.2%</td>
<td>47.1%</td>
<td>56.4%</td>
<td>44.8%</td>
<td>38.9%</td>
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Table B.9: Fiber measurements for sample He
## APPENDIX B. IMAGE ANALYSIS DATA

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<tbody>
<tr>
<td>$d_p$</td>
<td>0.465</td>
<td>0.654</td>
<td>0.522</td>
<td>0.677</td>
<td>0.444</td>
</tr>
<tr>
<td>$d_{p,j}$</td>
<td>0.719</td>
<td>0.862</td>
<td>0.794</td>
<td>0.880</td>
<td>0.733</td>
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**Table B.10:** Pore measurements for sample A

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<th>B-3</th>
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<tbody>
<tr>
<td>$d_p$</td>
<td>0.428</td>
<td>0.486</td>
<td>0.513</td>
<td>0.423</td>
<td>0.458</td>
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<tr>
<td>$d_{p,j}$</td>
<td>0.575</td>
<td>0.775</td>
<td>0.705</td>
<td>0.689</td>
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**Table B.11:** Pore measurements for sample B

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</thead>
<tbody>
<tr>
<td>$d_p$</td>
<td>0.483</td>
<td>0.428</td>
<td>0.426</td>
<td>0.401</td>
<td>0.421</td>
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<tr>
<td>$d_{p,j}$</td>
<td>0.688</td>
<td>0.588</td>
<td>0.633</td>
<td>0.526</td>
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**Table B.12:** Pore measurements for sample C

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</thead>
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<tr>
<td>$d_p$</td>
<td>0.533</td>
<td>0.418</td>
<td>0.433</td>
<td>0.425</td>
<td>0.433</td>
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<tr>
<td>$d_{p,j}$</td>
<td>0.765</td>
<td>0.566</td>
<td>0.715</td>
<td>0.651</td>
<td>0.609</td>
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**Table B.13:** Pore measurements for sample D
## APPENDIX B. IMAGE ANALYSIS DATA

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<tbody>
<tr>
<td>Perimeter, $P_j$ (mm)</td>
<td>11.381</td>
<td>8.699</td>
<td>12.491</td>
<td>11.412</td>
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<tr>
<td>Hydraulic diameter, $d_{p,j}$ (mm)</td>
<td>3.162</td>
<td>2.654</td>
<td>3.440</td>
<td>3.231</td>
<td>2.898</td>
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Table B.14: Pore measurements for sample E

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</thead>
<tbody>
<tr>
<td>$d_p$</td>
<td>4.744</td>
<td>5.416</td>
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<td>5.559</td>
<td>7.103</td>
</tr>
<tr>
<td>Perimeter, $P_j$ (mm)</td>
<td>9.177</td>
<td>8.767</td>
<td>10.059</td>
<td>9.159</td>
<td>9.397</td>
</tr>
<tr>
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Table B.15: Pore measurements for sample F

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Table B.16: Pore measurements for sample G

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Table B.17: Pore measurements for sample H
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Table B.18: Porosity measurements for Ni-40PPI
## Table B.19: Porosity measurements for Ni-10PPI

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Table B.20: Porosity measurements for Cu-40PPI
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Table B.21: Porosity measurements for Cu-10PPI
Appendix C

Pressure Drop Data

The data obtained from the fluid flow and pressure drop experiments described in Chapter 2 is given in this appendix in tabular form. The characteristics of the metal foam sample for which the data was collected are given in Table 3.5. The data is grouped as follows: in each table the first column lists the mass flow rate of air released into the channel. The next columns are the pressure drop values measured for each foam type. Each pressure drop value is measured between two fixed pressure ports and held constant at 250 mm. The measurements are repeated several times indicated by the trial number. The properties of air at 1 atm pressure are taken from [78].
### APPENDIX C. PRESSURE DROP DATA

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### APPENDIX C. PRESSURE DROP DATA

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# APPENDIX C. PRESSURE DROP DATA

## Table C.5: Pressure drop measurements: Trial 4

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Table C.7: Pressure drop measurements: Trial 7
Appendix D

Heat Flux Data

The data obtained in the forced convection experiments described in Chapter 5 is given in this appendix in tabular form. The data is organized as follows: the table is named according to the voltage input to the heater followed by the resulting heat flux amount, and the foam channel type. The first column is the mass flow rate at which air was fed into the channel. The next columns are the axial temperature values measured at a specific mass flow rate. For each mass flow rate the wall $T_s$, and bulk fluid temperature $T_f$ are indicated. The last two columns show the temperature of at the surface of the insulation $T_{loss}$ to measure the amount of heat loss to the surroundings. Also included in this section are results obtained during stagnate flow for heat conduction analysis. The properties of air at 1 atm pressure are taken from [25].
### Temperature at axial locations (°C)

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<th>127.0 mm</th>
<th>177.8 mm</th>
<th>228.6 mm</th>
<th>279.6 mm</th>
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Table D.1: Temperature results for 10PPI - Ni foam channel at 20V (428 W/m²)
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# APPENDIX D. HEAT FLUX DATA

## Temperature at Axial Locations (°C)

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Table D.3: Temperature results for 10PPI - Ni foam channel at 60V (3851 W/m²)

---

**Note:** The table represents temperature readings at various axial locations for different flow rates and phases. The temperatures are recorded in degrees Celsius (°C).

---

122
### Temperature at axial locations (°C)

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<th>127.0 mm</th>
<th>177.8 mm</th>
<th>228.6 mm</th>
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Table D.4: Temperature results for 10PPI - Ni foam channel at 80V (6846 W/m²)
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Table D.5: Temperature results for 40PPI - Ni foam channel at 40V \((1171 \text{ W/m}^2)\)
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Table D.6: Temperature results for 40PPI - Ni foam channel at 60V (3851 W/m²)
### Appendix D. Heat Flux Data

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Table D.7: Temperature results for 40PPI - Ni foam channel at 80V (6846 W/m$^2$)
## Table D.8: Temperature results for 10PPI - Cu foam channel at 20V (428 W/m²)

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Table D.9: Temperature results for 10PPI - Cu foam channel at 40V (1171 W/m²)
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Table D.10: Temperature results for 10PPI - Cu foam channel at 60V (3851 W/m²)
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Table D.11: Temperature results for 10PPI - Cu foam channel at 80V (6846W/m²)
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Table D.12: Temperature results for 40PPI - Cu foam channel at 20V (428W/m²²)
### Temperature at axial locations (°C)

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<th>127.0 mm</th>
<th>177.8 mm</th>
<th>228.6 mm</th>
<th>279.6 mm</th>
<th>$T_{\text{loss,1}}$ (°C)</th>
<th>$T_{\text{loss,2}}$ (°C)</th>
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<td>177.57</td>
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Table D.13: Temperature results for 10PPI - Cu foam channel at 40V (1171 W/m²)
## Temperature at axial locations (°C)

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<th>$Q$ (L/min)</th>
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<th>76.2 mm</th>
<th>127.0 mm</th>
<th>177.8 mm</th>
<th>228.6 mm</th>
<th>279.6 mm</th>
<th>$T_{loss,1}$ (°C)</th>
<th>$T_{loss,2}$ (°C)</th>
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<tbody>
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Table D.14: Temperature results for 10PPI - Cu foam channel at 60V (3851 W/m²)
### Table D.1: Temperature results for 10PPI - Cu foam channel at 80V (6846 W/m²)

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<th>127.0 mm</th>
<th>177.8 mm</th>
<th>228.6 mm</th>
<th>279.6 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( T_s )</td>
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### Table D.1.1: Heat flux data

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<th>( T_{loss,2} ) (°C)</th>
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### Table D.16: Temperature results for Ni foam channel during conduction.

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<th>$Q_H$ (W)</th>
<th>Temperature at axial locations (°C)</th>
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### Table D.17: Temperature results for Cu foam channel during conduction.

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<th>Temperature at axial locations (°C)</th>
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</table>

Table D.16: Temperature results for Ni foam channel during conduction.

Table D.17: Temperature results for Cu foam channel during conduction.
Appendix E

Heat Exchanger Data

The data obtained in the water-to-air heat exchanger experiments described in Chapter 6 is given in this appendix in tabular form. The data is organized as follows: each table corresponds to a constant flow rate of water. The first half of the table corresponds to temperatures obtained by the metal foam heat exchanger (MFHE) and the second half correspond to the plain tube heat exchanger (PTHE). Then, each row corresponds to the inlet and outlet temperature of the air and water at given flow rates.
### APPENDIX E. HEAT EXCHANGER DATA

<table>
<thead>
<tr>
<th>MFHE</th>
<th>Air</th>
<th>Water</th>
<th>Air</th>
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<tr>
<td>$Q_w$ (GPM)</td>
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<td>$T_{out}$ ($^\circ$C)</td>
<td>$Q_a$ (L/min)</td>
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</tr>
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#### Table E.1: Temperature results for heat exchanger experiments, water flow rate at 0.3 GPM

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<td>$Q_w$ (GPM)</td>
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<td>$Q_a$ (L/min)</td>
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#### Table E.2: Temperature results for heat exchanger experiments, water flow rate at 0.6 GPM
### Table E.3: Temperature results for heat exchanger experiments, water flow rate at 0.8 GPM

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### Table E.4: Temperature results for heat exchanger experiments, water flow rate at 0.9 GPM

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<th>Air</th>
<th>( Q_a ) (L/min)</th>
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Appendix F

Derivation of One-Dimension Heat Conduction Model

The following derivation is performed on control-volume shown in Figure 5.3 for the solid phase.

\[ Q_H + Q_{\text{Cond}_s} = Q_{\text{Cond}_{s+\Delta x}} \]  \hspace{1cm} (F.1)

Where \( Q_H \), provided by the heater accounts for the heat losses mentioned in Chapter 2, and is taken as a volumetric heat flux

\[ \frac{Q_H}{A_t L} = q''' \left[ \frac{W}{m^3} \right] \]  \hspace{1cm} (F.2)

Applying the same volumetric technique to the entire energy-balance equation, it becomes

\[ \lim_{\Delta x \to 0} \frac{Q_{\text{Cond}_{s+\Delta x}} - Q_{\text{Cond}_{s}}}{A_t \Delta x} = q''' \frac{A_s \Delta x}{A_t \Delta x} \]  \hspace{1cm} (F.3)

\[ \frac{1}{A_t} \frac{d}{dx} \left\{ -k_s A_s \frac{dT_s}{dx} \right\} = q''' \frac{A_s}{A_t} \]  \hspace{1cm} (F.4)

Where the volume fraction of the solid phase is accounted for by Equation (F.3), resulting in the axial temperature distribution of the solid phase as Equation (F.6)

\[ \frac{A_s}{A_t} = (1 - \varepsilon) \]  \hspace{1cm} (F.5)

\[ -k_s (1 - \varepsilon) \frac{dT_s}{dx^2} = q''' (1 - \varepsilon) \]  \hspace{1cm} (F.6)
Carrying out the same derivation for the fluid phase, the axial temperature distribution results in the following equation

$$-k_f(\varepsilon) \frac{d^2T_f}{dx^2} = q'''(\varepsilon) \quad (F.7)$$

Assuming local thermal equilibrium $T_s = T_f = T$, the overall temperature distribution can be simplified as

$$-k_s(1-\varepsilon) \frac{d^2T}{dx^2} - k_f(\varepsilon) \frac{d^2T}{dx^2} = q'''(\varepsilon) + q'''(1-\varepsilon) \quad (F.8)$$

$$[k_s(1-\varepsilon) + k_f(\varepsilon)] \frac{d^2T}{dx^2} = q''' \quad (F.9)$$

In order to solve the second-order differential equation, presented in Equation (F.9), the boundary conditions are chosen as specified temperature values at the inlet and outlet obtained from the experimental results:

$$T_1 = T_{x=0} = \frac{T_{s1} + T_{f1}}{2} \quad (F.10)$$

$$T_2 = T_{x=L} = \frac{T_{s6} + T_{f6}}{2} \quad (F.11)$$

Then the Buckingham-Pi theorem is applied to non-dimensionalize the equation, with six variables in three dimensions, or $7 - 3 = 3$ pi groups.

$$T(x) - T_1 = f_n[k_{eff}, x, L, q''', (T_2 - T_1)] \quad (F.12)$$

where

$$\Pi_1 = \frac{T - T_1}{T_2 - T_1} \quad \text{dimensionless temperature defined as } \theta$$

$$\Pi_2 = \frac{x}{L} \quad \text{dimensionless length defined as } X$$

$$\Pi_2 = \frac{q'''L^2}{k_{eff}(T_2 - T_1)} \quad \text{compares the rate of heat generation to the rate of heat loss, and is defined as } \phi$$

Applying the non-dimensional terms with its respective boundary conditions, Equation (F.9), results in the following:

$$\theta = \frac{\phi}{2} X^2 + \left(1 - \frac{\phi}{2}\right) X \quad (F.13)$$

where

$$X = \frac{x}{L} = 0 \quad \text{and} \quad \theta = \frac{T_1 - T_2}{T_1 - T_1} = 0$$

$$X = \frac{L}{L} = 1 \quad \text{and} \quad \theta = \frac{T_2 - T_1}{T_2 - T_1} = 1$$