

EXPERIMENTAL COMPARISON OF FLUID AND THERMAL
CHARACTERISTICS OF MICROCHANNEL AND METAL FOAM HEAT
SINKS

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CHARACTERISTICS OF MICROCHANNEL AND METAL FOAM HEAT
SINKS**

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ABSTRACT

EXPERIMENTAL COMPARISON OF FLUID AND THERMAL CHARACTERISTICS OF MICROCHANNEL AND METAL FOAM HEAT SINKS

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Doubling transistor count for every two years in a computer chip, transmitter and receiver (T/R) module of a phased-array antenna that demands higher power with smaller dimensions are all results of miniaturization in electronics packaging. These technologies nowadays depend on improvement of reliable high performance heat sink to perform in narrower volumes. Employing microchannels or open cell metal foam heat sinks are two recently developing promising methods of cooling high heat fluxes. Although recent studies especially on microchannels can give a rough estimate on performances of these two methods, since using metal foams as heat sinks is still needed further studies, a direct experimental comparison of heat exchanger performances of these two techniques is still needed especially for thermal design engineers to decide the method of cooling.

For this study, microchannels with channel widths of 300 μm , 420 μm , 500 μm and 900 μm were produced. Also, 92% porous 10, 20 and 40 ppi 6101-T6 open cell aluminum metal foams with compression factors 1,2, and 3 that have the same

finned volume of microchannels with exactly same dimensions were used to manufacture heat sinks with method of vacuum brazing. They all have tested under same conditions with volumetric flow rate ranging from 0,167 l/min to 1,33 l/min and 60 W of heat power. Channel height was 4 mm for all heat sinks and distilled water used as cooling fluid. After experiments, pressure drops and thermal resistances were compared with tabulated and graphical forms. Also, the use of metal foam and microchannel heat sinks were highlighted with their advantages and disadvantages for future projects.

Keywords: Microchannel Heat Sink, Metal Foam Heat Sink, Single-Phase Forced Convection Cooling, Fluid Flow through Metal Foam, Fluid Flow through Rectangular Microchannels

ÖZ

MİKROKANALLI VE METAL KÖPÜKLÜ ISI ALICILARI AKIŞKAN VE ISIL ÖZELLİKLERİNİN DENEYSEL OLARAK KARŞILAŞTIRILMASI

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Bilgisayar çiplerinde her iki senede bir ikiye katlanan transistör sayısı, faz dizili anten yapılarının küçülen boyutlu ve gittikçe daha yüksek güç ile çalışan almaç ve göndermeç (A/G) modül tasarımlarının gerçekleştirilmesi gibi konuların hepsi elektronik minyatürleştirmenin bir sonucu. Bugünlerde bu teknolojilerin daha dar hacimlerde çalışması ise güvenilir ısı alıcıların geliştirilmesine bağlı. Mikrokanallı ve açık hücreli metal köpüklerin yüksek performanslı ısı alıcılar olarak kullanılması son zamanlarda gelişen umut verici yöntemler. Son zamanlarda özellikle mikrokanallar üzerinde yayınlanan çalışmalar bu iki soğutma yönteminin performansları hakkında kaba bir değerlendirme imkânı sunsalar da, metal köpüklerin ısı alıcılar olarak kullanılmasında hala ileri çalışmaların gerekmesinden dolayı bu iki tekniğin deneysel olarak karşılaştırılmasına özellikle ısı tasarım mühendisleri tarafından ihtiyaç duyulmaktadır.

Bu çalışmada 300µm, 420µm, 500µm ve 900 µm kanal genişliğine sahip mikrokanallar üretilmiştir. Ayrıca %92 poroziteye haiz 10, 20 ve 40 PPI (bir inç başına düşen gözenek sayısı) 6101-T6 alaşımlı açık hücreli alüminyum metal köpükler 1,2 ve 3 kat sıkıştırılıp mikrokanalların kanallı hacimleriyle aynı hacme ve

boyutlara getirildikten sonra metal köpüklü ısı alıcılar vakumlu sert lehimleme yöntemi ile üretilmişlerdir. Üretilen ısı alıcıların hepsi 0,167 l/dak debiden 1,33l/dak debiye kadar 60W ısı güç gönderilerek aynı şartlar altında deneysel olarak test edilmişlerdir. Deneylerden sonra basınç düşüşü ve ısı dirençler tablolar ve grafiklerle karşılaştırılmışlardır. Ayrıca metal köpüklerin ve mikrokanalların ısı alıcılar olarak kullanılması avantaj ve dezavantajları ile gelecek projelere ışık tutması açısından incelenmiştir.

Anahtar Kelimeler: Mikrokanallı Isı Alıcılar, Metal Köpüklü Isı Alıcılar, Tek Fazlı Zorlanmış Konveksiyonel Soğutma, Metal Köpüklerde Akış, Mikrokanallarda Akış

To my son and my family

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TABLE OF CONTENTS

ABSTRACT.....	iv
ÖZ.....	vi
ACKNOWLEDGEMENTS.....	ix
TABLE OF CONTENTS.....	x
LIST OF TABLES.....	xiii
LIST OF FIGURES.....	xv
LIST OF SYMBOLS.....	xviii
CHAPTERS	
1. INTRODUCTION.....	1
1.1 Motivation.....	1
1.2 Cooling Technologies of Electronics.....	4
1.3 Goal of the Study.....	13
2. AN OVERVIEW OF STUDIES ON HEAT TRANSFER AND FLUID FLOW IN MICROCHANNELS AND METAL FOAM HEAT SINKS.....	15
2.1 Heat transfer and fluid flow studies on microchannels.....	15
2.1.1 Concept of microchannel and related early work.....	16
2.1.2 Experimental studies with single-phase flow.....	18
2.2 Heat transfer and fluid flow studies on metal foams.....	31
3. A CLOSER VIEW ON METAL FOAM TECHNOLOGY.....	50
3.1. Metal foams introduction and general properties.....	50

3.2 Categorization of Metal Foams.....	53
3.3 Cutting, finishing and joining of metal foams.....	55
3.3.1 Cutting of metal foams.....	55
3.3.2 Finishing metal foams.	55
3.3.3 Joining of metal foams.....	56
4. SINGLE-PHASE LIQUID FLOW AND HEAT TRANSFER IN MICROCHANNEL AND METAL FOAM HEAT SINKS.....	58
4.1 Single-phase pressure drop in microchannels.....	58
4.2 Forced convection in microchannels.....	64
4.3 A discussion on applicability of classical theory on microchannels.....	69
4.4 Single-phase pressure drop analysis of metal foam heat sinks.....	72
4.5 Forced convection analysis of metal foam heat sinks.....	76
5. EXPERIMENTAL SETUP AND DISCUSSION ON RESULTS.....	77
5.1 Motivation for experimental study.....	77
5.2 Experimental apparatus.....	77
5.3 Microchannel Heat Sinks.....	89
5.4 Metal Foam Heat Sinks.....	93
5.5 Uncertainty Analysis.....	97
5.6 Microchannel pressure drop results and discussion.....	102
5.7 Microchannel heat transfer results and discussion.....	108
5.8 Metal foam pressure drop results and discussion.....	110
5.9 Metal foam heat transfer results and discussion.....	124
5.10 Heat exchanger performance comparison.....	126
6. CONCLUSIONS.....	130

REFERENCES.....	134
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APPENDICES

A-TECHNICAL DRAWINGS OF TEST SPECIMENS.....	142
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B-SAMPLE UNCERTAINTY ANALYSIS.....	148
------------------------------------	-----

C-SAMPLE ANALYTICAL PRESSURE DROP AND NUSSELT NUMBER CALCULATION FOR MICROCHANNELS.....	150
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LIST OF TABLES

TABLES

Table 2.1: Summary of microchannel studies in the literature.....	23
Table 2.2: A synthesis of fluid and thermal transport models of metal foam heat exchangers.....	43
Table 3.1: Potential applications of metal foams [82]	50
Table 4.1: Nusselt numbers for thermally developing region [17]	67
Table 4.2: Nusselt numbers for thermally fully developed flow [17].....	68
Table 5.1: General properties of experimental instruments.....	79
Table 5.2: Dimensional parameters of microchannel test specimens.....	85
Table 5.3: Physical properties of tested metal foams.....	94
Table 5.4: 300 micron channel experimental and analytical results and related uncertainties.....	104
Table 5.5: 420 micron channel experimental and analytical results and related uncertainties.....	105
Table 5.6: 500 micron channel experimental and analytical results and related uncertainties.....	106
Table 5.7: 900 micron channel experimental and analytical results and related uncertainties.....	107
Table 5.8: Heat transfer calculated and experimental results of microchannels.	109
Table 5.9: Permeability and form coefficients of metal foams tested. Values of foam 10-01 were taken from Boomsma's study [16]	111
Table 5.10: Metal foam 10-03 pressure drop results.....	114
Table 5.11: Metal foam 20-01 pressure drop results.....	114

Table 5.12: Metal foam 20-02 pressure drop results.....	117
Table 5.13: Metal foam 20-03 pressure drop results.....	117
Table 5.14: Metal foam 40-01 pressure drop results.....	120
Table 5.15: Metal foam 40-02 pressure drop results.....	122
Table 5.16: Foam 40-03 pressure drop results.....	123
Table 5.17: Nusselt number and thermal resistances of metal foams.....	125
Table 6.1: Physical properties of tested heat sinks.....	131
Table D.1: Curve-fit equations for Tables 4.1, and 4.2.....	153

LIST OF FIGURES

FIGURES

Figure 1.1: Microprocessor transistor counts 1971-2011 and Moore's Law [5].....	2
Figure 1.2: Beam steering and focusing in phased array antenna system [7].....	3
Figure 1.3: Theoretical heat transfer coefficients vs. cooling method [8].....	4
Figure 1.4: Round pin fin heat sink with free convection fluid flow trajectories.....	5
Figure 1.5: AMD Heat sink and fan.....	6
Figure 1.6: Schematic of a piezoelectric fan oscillating under an applied voltage due to the contraction and expansion of a piezoceramic patch [10].....	7
Figure 1.7: Sketch of nanolightning [12]	8
Figure 1.8: Commercially available multiple jet impingement liquid cooling [12]....	9
Figure 1.9: Direct liquid spray cooling of chips [13]	10
Figure 1.10: Refrigeration cycle [14]	11
Figure 1.11: Thermal cycle inside a heat pipe [15]	12
Figure 1.12: Schematic of simple Peltier cooler [12]	13
Figure 2.1: Related figures of some dimensional parameters of correlations given in table 2.1 [81]	49
Figure 3.1: Open cell metal foam samples.....	53
Figure 3.2: Closer view of open cell metal foam.....	54
Figure 3.3: Closed cell metal foam [84]	54
Figure 3.4: Joining methods of metal foams.....	56
Figure 4.1: Rectangular duct figure.....	59
Figure 4.2: Contraction and expansion loss coefficients [89]	64

Figure 4.3: Cross-sectional view of a rectangular microchannel etched in silicon showing distortion in channel geometry from ideal rectangular profile [85].....	71
Figure 5.1: Schematic view of the experimental setup used to measure hydraulic and thermal characteristics of microchannel and metal foam heat sinks.....	78
Figure 5.2: Views of heat sink assembly in detail.....	85
Figure 5.3: Views of heat load carrier and thermocouple locations.....	86
Figure 5.4: Views of heat sink assembly with isolation.....	87
Figure 5.5: General view of test setup.....	88
Figure 5.6: Pre-brazing body of 300 micron channel.....	90
Figure 5.7: Brazed channel assembly.....	91
Figure 5.8: Microscope view of 300 micron channel.....	91
Figure 5.9: Final product.....	92
Figure 5.10: Compressing operation of metal foams.....	95
Figure 5.11: Metal foam pieces before brazing operation.....	96
Figure 5.12: Metal foam cutting with wire EDM.....	97
Figure 5.13: Experimental vs. analytical pressure drop results comparison for 300 micron channel.....	104
Figure 5.14: Experimental vs. analytical pressure drops results comparison for 420 micron channel.....	105
Figure 5.15: Experimental vs. analytical pressure drop results comparison for 500 micron channel.....	106
Figure 5.16: Experimental vs. analytical pressure drop results comparison for 900 micron channel.....	107
Figure 5.17: Nusselt number vs. Reynolds number experimental results.....	108

Figure 5.18: Experimental vs. Boomsma [16] results comparison for foam 40-03.....	112
Figure 5.19: Metal foam 10-03 pressure drop results.....	113
Figure 5.20 : Metal foam 20-01 pressure drop results.....	115
Figure 5.21: Metal foam 20-02 pressure drop result.....	116
Figure 5.22: Metal foam 20-03 pressure drop result.....	118
Figure 5.23: Metal foam 40-01 pressure drop result.....	119
Figure 5.24: Metal foam 40-02 pressure drop result.....	121
Figure 5.25: Nusselt number versus volumetric flow rate of metal foam samples.....	126
Figure 5.26: Pressure drop and thermal resistance comparison graphs.....	127
Figure 5.27: Pressure drop and thermal resistance comparison graphs.....	128
Figure A.1: Technical drawing of 300 μ m microchannel pre-brazing body.....	142
Figure A.2: Technical drawing of 420 μ m microchannel pre-brazing body.....	143
Figure A.3: Technical drawing of 500 μ m microchannel pre-brazing body.....	144
Figure A.4: Technical drawing of 900 μ m microchannel pre-brazing body.....	145
Figure A.5: Empty pre-brazing body for metal foams.....	146
Figure A.6: Pre-brazing cover for all pre brazing heat sink bodies.....	147

LIST OF SYMBOLS

A_c	total channel area
A_{con}	the convection surface area
A_{cns}	control surface that inlet or outlet of mass flow takes place
A_{cs}	flow cross-sectional area
A_p	total plenum cross-sectional area
A_t	total convective heat transfer area of heat sink
a	short side of rectangular channel
b	long side of rectangular channel
c_f	specific heat capacity of fluid
d_{ave}	average pore diameter
D_{le}	laminar-equivalent diameter
d_{hyd}	hydraulic diameter
dp	infinitesimal pressure difference
dx	infinitesimal length of a fluid element
D_S	channel size
$\frac{D}{Dt}$	directional derivative: $\frac{D}{Dt} = \frac{\partial}{\partial t} + u \frac{\partial}{\partial x} + v \frac{\partial}{\partial y} + w \frac{\partial}{\partial z}$
e	error of a measurement
F_n	force in n direction
$\vec{F}$	sum of the body forces in the x,y and z directions
f	fanning friction factor

f_{app}	apparent friction factor
h	convection coefficient
h_p	height of the porous layer
I	current
j	Colburn factor
k_f	thermal conductivity of coolant
k_h	hydraulic conductivity
K	permeability of porous medium
K_c	sudden contraction loss coefficient
K_e	sudden expansion loss coefficient
K_{90}	90° bend loss coefficient
$K(x)$	incremental pressure defect
$K(\infty)$	Hagenbach's factor
L_H	entrance length of the hydrodynamic developing region
L	length
L_i	length of thermal entrance region
M	compression factor of metal foam
m_{cv}	mass of the control volume at any moment in time
$\dot{m}$	time rate of mass flow
$\dot{m}_i$	rate of mass entering the control volume
$\dot{m}_e$	rate of mass leaving the control volume
m	mass of the control volume
$\hat{n}_{A_{cns}}$	normal vector of the surface area element dA

Nu	Nusselt number
Nu_T	Nusselt number for constant wall temperature condition
Nu_{H1}	Nusselt number for constant circumferential wall temperature
Nu_{H2}	Nusselt number for Constant wall heat flux
Nu_3	Nusselt number for three-side heating condition
Nu_4	Nusselt number for four-side heating condition
$Nu_{3,fd}$	Nusselt number for fully developed three-side heating condition
$Nu_{4,fd}$	Nusselt number for fully developed four-side heating condition
p	pressure
Po	Poiseuille number
Pr	Prandtl number
Q	volumetric flow rate
q''	heat flux from a solid surface
q	heat transfer rate
R	electrical resistance
Re	Reynolds number
Re_{le}	laminar equivalent Reynolds number
R_{th}	thermal resistance
P_t	pressure at the top of the sand layer (units of length)
P_b	pressure at the bottom of the sand layer (units of length)
P_w	wetted perimeter
$T_{c,out}$	temperature of coolant at the exit
$T_{c,in}$	temperature of coolant at the inlet

T_s	surface temperature of convection
T_∞	bulk fluid temperature
U_a	area-averaged velocity
U	uncertainty
U_f	uncertainty value given by manufacturer
U_{Re}	uncertainty in Reynolds number
$U_{\dot{m}}$	uncertainty in mass flow rate
U_μ	uncertainty in dynamic viscosity
U_a	uncertainty in dimension “a”
U_b	uncertainty in dimension “b”
U_Q	uncertainty in volumetric flow rate
U_ρ	uncertainty in density
$U_{\Delta P}$	uncertainty in pressure drop
U_L	uncertainty in a measure length
U_c	uncertainty in specific heat
$U_{T_{c,out}}$	in temperature of coolant at the exit
$U_{T_{c,in}}$	in temperature of coolant at the inlet
$U_{T_{pl}}$	uncertainty in temperature of plate
U_{k_f}	uncertainty in thermal conductivity of coolant
A_{con}	uncertainty in conduction surface area
U_W	uncertainty in pumping power
U_u	uncertainty in flow velocity
$U_{R_{th}}$	uncertainty in thermal resistance

u_m	mean flow velocity of fluid in the channel
V	volume of the control volume
$\dot{W}$	pumping power
X	body forces in x direction
X_m	measured value of some quantity
X_T	true value of measured quantity
ρ	density of fluid
μ	dynamic viscosity
α	aspect ratio of rectangular channel
α_f	fluid thermal diffusivity
u	flow velocities in x direction
v	kinematic viscosity
v_y	flow velocities in y direction
$\vec{v}$	three-dimensional velocity vector
$\vec{v}_n$	projection of the velocity vector in the n direction
τ_w	shear stress at wall
ΔP	pressure drop
ΔT	temperature difference that results heat to flow
ε	porosity
$\varepsilon_{compressed}$	porosity of compressed metal foam
$\varepsilon_{uncompressed}$	porosity of uncompressed metal foam

CHAPTER 1

INTRODUCTION

The purpose of this chapter is to present cooling techniques that has been applied to electronics briefly, including traditional methods and current state of the art with extreme cooling methods. The goal and general outline of this study is shortly presented at the end.

1.1 Motivation

A resulting fact of a survey made by U.S. Air Force in 1989 showed that more than fifty per cent of all electronics failures are temperature related [1]. Development in electronics towards miniaturization is another challenging fact of thermal management both in commercial and military products. These realities presented above are the main source of motivation for discovery of advanced extreme ways of cooling.

Continuous increasing performance of microelectronics with decreasing packaging sizes requires high rates of heat removal. A forward looking researcher and Intel co-founder Moore started his article with the following sentence “the future of integrated electronics is the future of electronics itself” in 1965 [2]. In this study he also forecasted future developments in electronics, the necessity of miniaturization and asked if it would be possible to remove the heat generated by tens of thousands of components in a single silicon chip or not. Today, a chip can have more than three billion transistors in an area of nearly 500 millimeters square and cooling solution is obviously came with this commercial product [3].

After the invention of transistor in 1947, advance in production techniques made it possible to have more transistors in denser areas. In fact, doubling the number of transistors on a chip for every two years justifies the prediction of Moore [4]. An updated version of transistor count versus the date of introduction of related chip is presented in the Figure 1.1. In this figure, continuous line represents doubling number of transistor in every two years (Moore's Law) and black dots are chips launched.

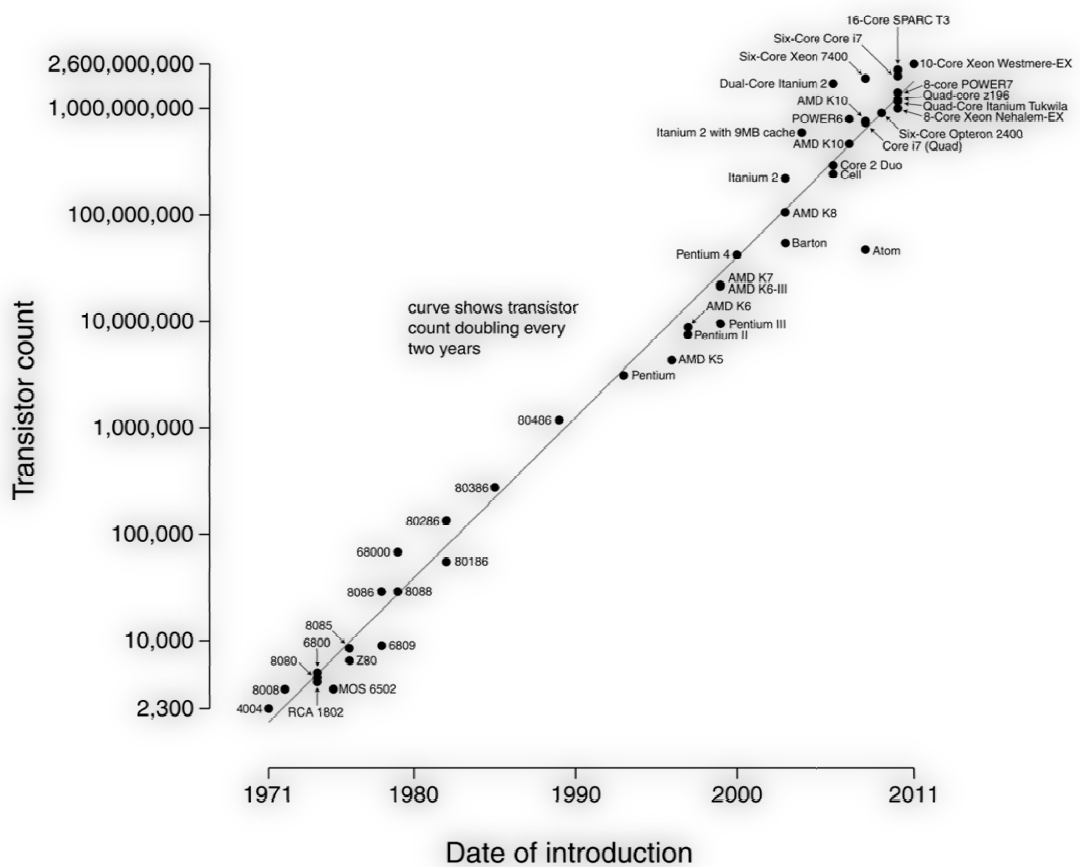


Figure 1.1 Microprocessor transistor counts 1971-2011 and Moore's Law [5]

Moore's law is expected to be valid until 2020 or later [6]. In the same study [1] Moore also predicted the future development in microwave area and mentioned that structures included in the definition of integrated electronics would become

increasingly important and successful realization of such items as phased-array antennas, for example, using a multiplicity of integrated microwave power sources, could completely revolutionize radar. It is that revolution that comes true nowadays made this study highly important to decide cooling technique of phased-array radars.

On the contrary of rotating or moving antenna systems phased array antennas stayed still because they have the ability of electronically steering and focusing electromagnetic beams. This capability, shown in Figure 1.2, comes with great advantage of having fast responding and flexible systems. Designing such a system requires an association of many engineering disciplines especially for Active Phased Array Radar (APAR), the system where the function of transmitting and receiving both handled electromagnetically in one antenna system. More detailed information is not given since it is beyond the scope of this thesis but it can be said that increasing the capability, i.e. delivering more power for increasing accuracy, speed and range, results in more heat dissipating antenna systems. Since traditional cooling techniques does not solve cooling problem of such systems, new applicable high heat removing techniques should be used.

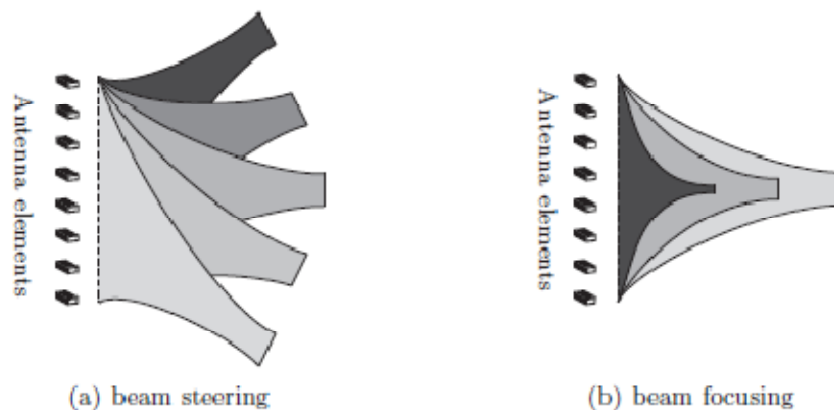


Figure 1.2 Beam steering and focusing in phased array antenna system [7]

1.2 Cooling Technologies of Electronics

Even in today's cooling applications forced air convection method is highly used since when compared to other sophisticated cooling techniques it is easier, cheaper and safer to put into practice. Since compactness and lightness are other important design criteria, instead of having electronics in large and heavy systems, alternative methods of cooling with high rates of heat removal looks inescapable. An illustrative figure from Bejan [8] shown in Figure 1.3 represents the theoretically within reach heat transfer coefficients in the temperature range of 0-100 °C, for air and water as coolant since they are mostly employed.

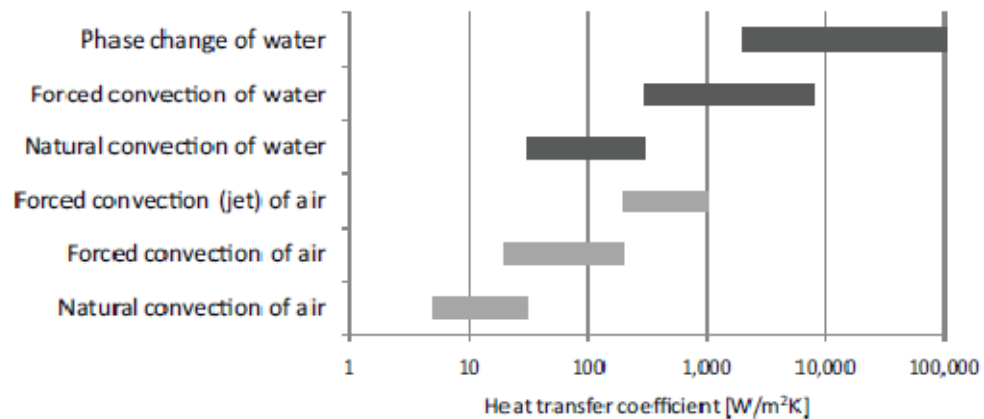


Figure 1.3 Theoretical heat transfer coefficients vs. cooling method [8]

As seen from Figure 1.3, usage of low capacity cooler air gave its place to water. Since water has more heat carrying capacity than air, its usage solved most of the cooling problems yet, for some advanced micro electronics applications additional considerations were required. Increasing heat exchange areas is one important method to achieve high heat removal rates. Other liquids also can be used if application temperature range is wanted to be shifted. Cooling with phase changing is also a promising method of heat removing and it is also beyond the scope of this study.

Later in this part of the chapter, cooling methods represented in Figure 1.3 is briefly discussed. When it comes to application, theoretical values are very difficult to achieve due to many reasons such as limitations in the volume, required place of cooling spot in the overall system, limited temperature gradient, acoustics etc. For example, theoretically, it is possible to cool down any electronics system by natural air convection if sufficient space is provided with unconstrained temperature gradient. However every unit has its own limit of high temperature durability.

Natural convection of air is the simplest way of cooling if applicable. Today, many electronics devices used at home such as CRT, LCD, LED televisions, stereos is being cooled down by this method. In most applications, to increase heat transfer area, heat is carried away from hot spots by conduction to designed fins that are exposed directly to open air. However even with a heat sink, total heat fluxes are limited to approximately 0.05 W/cm^2 [7]. In Figure 1.4, a CFD image of a round pin heat sink with free convection fluid flow trajectories was presented.

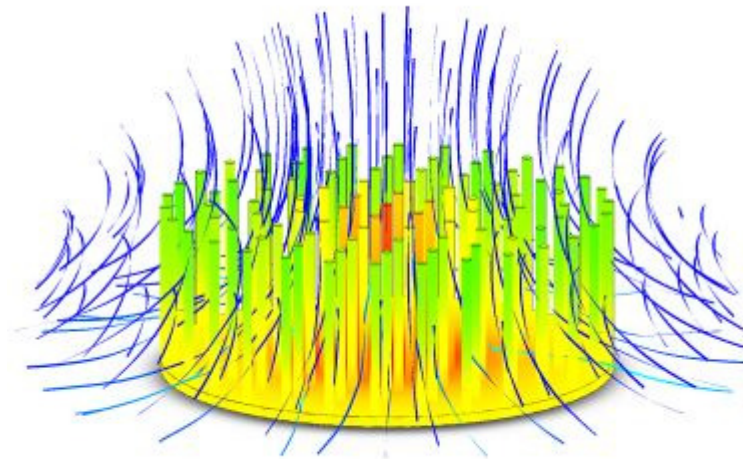


Figure 1.4 Round pin fin heat sink with free convection fluid flow trajectories

Forced convection of air is generally implemented with heat sink – fan couple. Many electronics devices use this method to carry heat away including many desktop and laptop computers. A standard fan can achieve a cooling capacity of 1 W/cm^2 , when a specially designed heat exchangers like shown in Figure 1.5 can carry heat rates of nearly 50 W/cm^2 [7]. Designing a forced air convection implied heat exchanger with fan is pretty straight-forward and mass production is highly available since it is reliable and inexpensive. If more heat is desired to be carried, a larger heat exchanger should be designed. Application limitations are limited volume and periodic maintenance required for fans, which are not acceptable for defense industry.

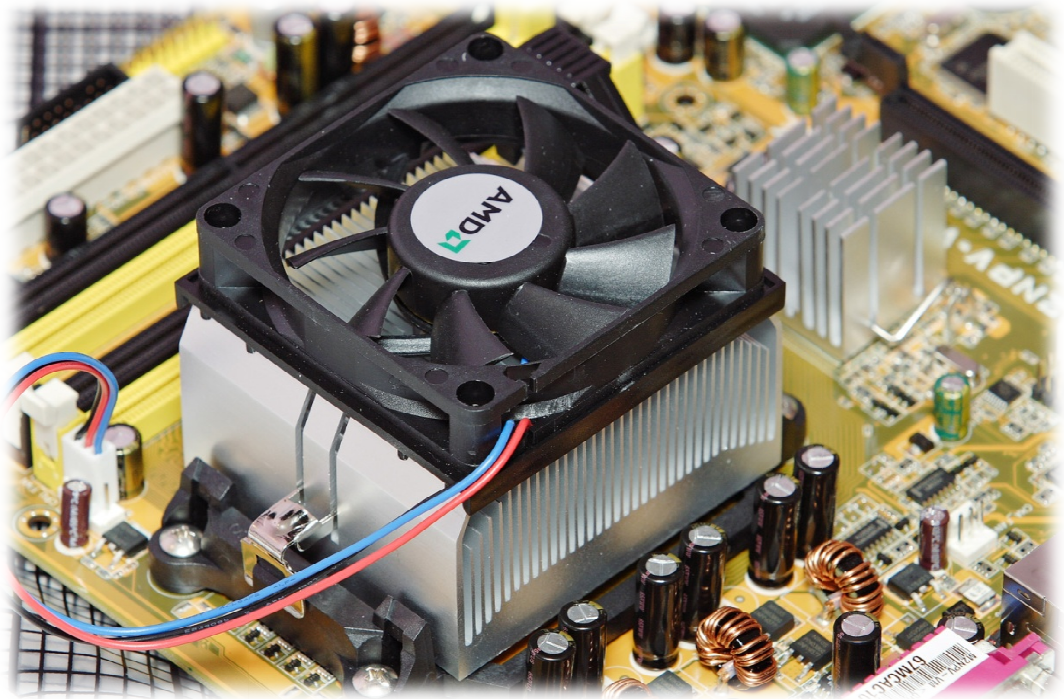


Figure 1.5 AMD Heat sink and fan

Instead of conventional fans, piezoelectric fans can be used. Piezoelectric effect results in blade shaking and thus forced air convection as seen in Figure 1.6. Since a piezoelectric fan is compact, has minimal noise, needs very low power to

operate and moreover is reliable, with growing portable electronics and greater capabilities, it can find a promising place in the industry. Piezoelectric fans are found to offer significant localized cooling, exceeding the enhancements in convective heat transfer coefficients by 100% when compared to natural convection [9].

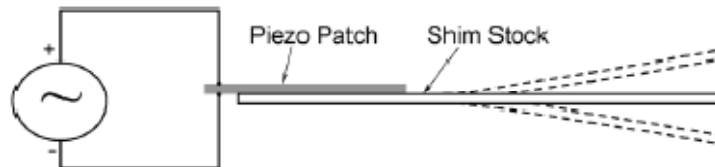


Figure 1.6 Schematic of a piezoelectric fan oscillating under an applied voltage due to the contraction and expansion of a piezoceramic patch [10]

Another new approach for forced convection air cooling is called nanolightning. Nano tubes produces very high electric fields resulting in micro-scale ion-driven airflow. As shown in Figure 1.7, the ionized air molecules are moved by another electric field, thereby inducing secondary airflow [11]. A heat flux of 40 W/cm^2 is possible with this method [12]. Respectively high power demand can be said to be the negative side of this technology.

Microscale Ion Driven Air Flow

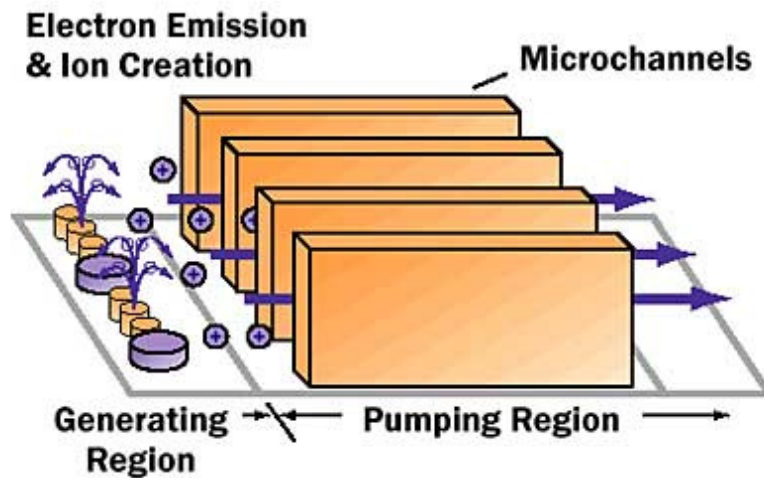


Figure 1.7 Sketch of nanolightning [12]

Jet air cooling includes applying air jet of 10-100 m/s to the hot surface. By locally breaking through the thermal boundary layer, higher heat transfer rates of 60 W/cm² can be attained [7]. This amount of heat removal rate is nearly the limit of cooling that can be achieved with air.

When air as coolant is not sufficient to carry dissipated heat away, liquid cooling can be employed. Liquid cooling can be directly or indirectly applied. Indirect cooling is satisfied with additional heat exchangers that have a contact with heated electronics part and liquid flows through the line and heat exchanger. In direct cooling, coolant directly makes contact with the chip. Since liquid in the cooling cycle needs to be recycled, sealing of the line is very important. If any leakage of conductive liquid coolant occurs, short circuits and unrecoverable damages can take place. Short circuit problem can be solved by using dielectric liquid, however in case of any leakage, system would become useless. So, liquid cooling can be said to be unreliable when compared to air cooling. Common liquids are water, glycol and alcohols.

Minichannels and microchannel heat sinks are kinds of indirect cooling application. Here “mini” refers to diameters in the order of millimeters and “micro” refers to channels that have hydraulic diameters less than these values. For laminar flow inside minichannels and microchannels, heat transfer coefficient is inversely proportional to the hydraulic diameter of the channel. This means that as hydraulic diameter gets smaller, heat transfer coefficient becomes larger. One great disadvantage is the high pressure drop. Cooling application with metal foam heat sinks is also a type of indirect cooling. The purpose is similar to that of the minichannels and microchannels namely, to increase heat transfer surface area. These heat sinks with high heat removal rates are promising. Since the scope of this study includes minichannels, microchannels and metal foam heat sinks, detailed information is given in later chapters.

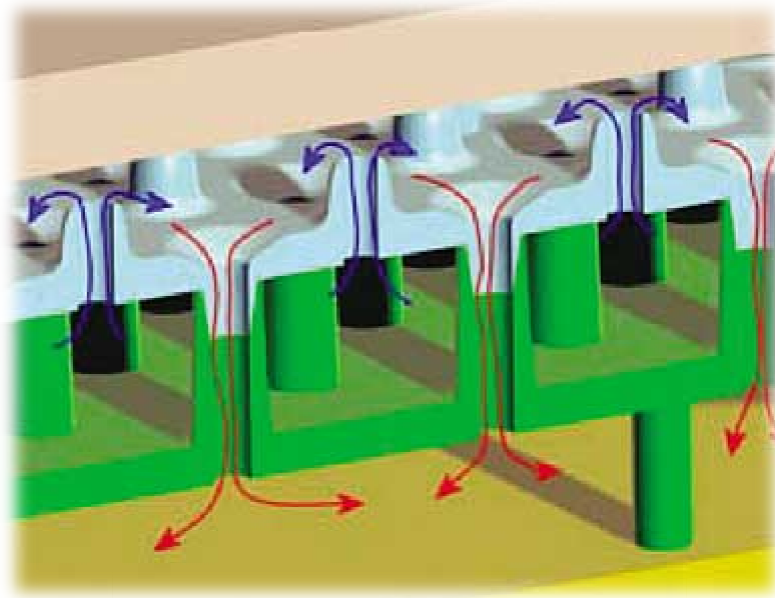


Figure 1.8 Commercially available multiple jet impingement liquid cooling [12]

Liquid jet impingement directly to the heated circuit is also an effective cooling method. With this method of cooling, a of heat removal rate of 460 W/cm^2 is possible [13]. Figure 1.8 represents liquid jet impingement schematically.

Direct dielectric liquid spraying onto chip surfaces, as shown in Figure 1.9 is employed as a heat removal technique commercially. 60 W/cm^2 of heat load can be carried [13] away from the heated surface.

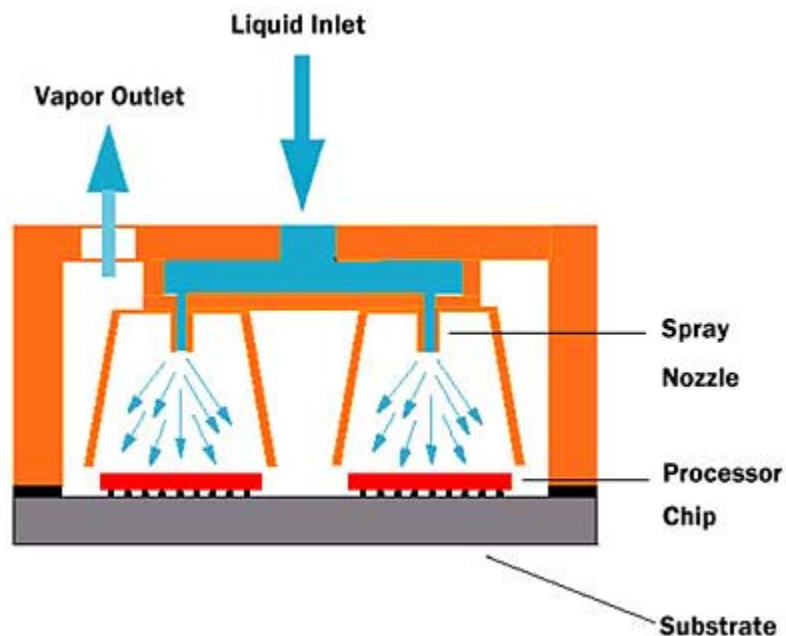


Figure 1.9 Direct liquid spray cooling of chips [13]

Refrigeration cycle can be used to cool down electronics devices. In the refrigeration cycle, fluid absorbs heat at low pressure and releases it at a higher pressure. Mostly a phase change occurs. Pressure difference is satisfied by a compressor. A schematic representation of this cycle is given in Figure 1.10.

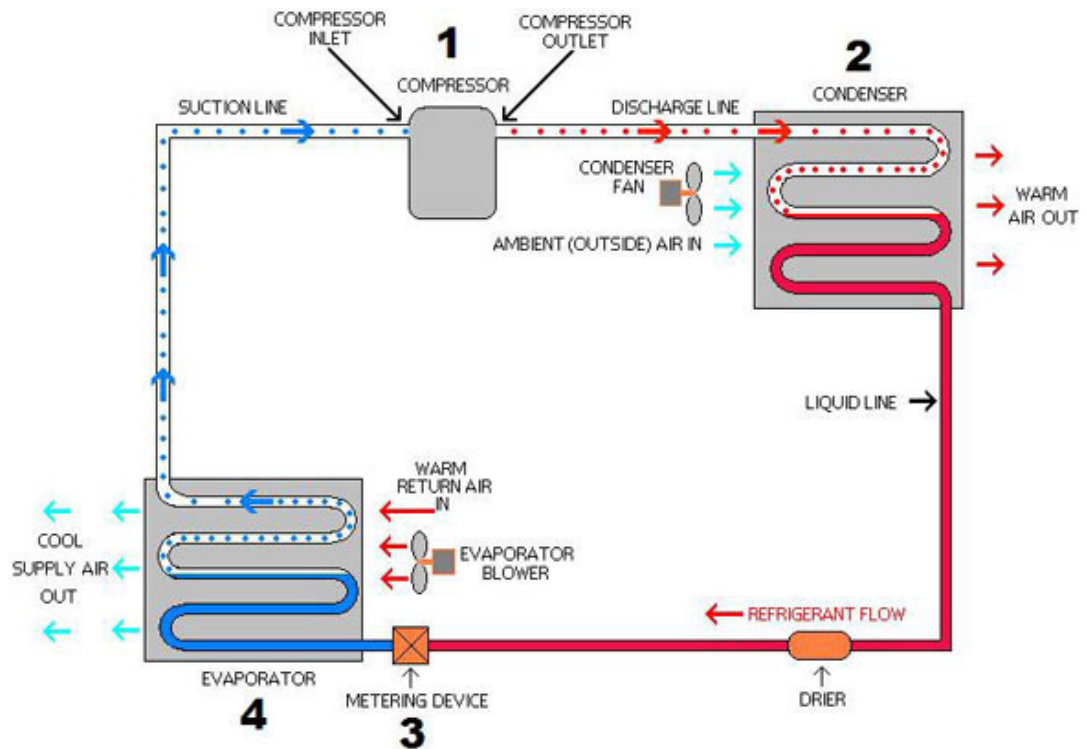


Figure 1.10 Refrigeration cycle [14]

Heat removal by phase changing is another important method used in electronics cooling. As a material changes its phase from solid to liquid or from liquid to gaseous, it absorbs a large amount of heat. The most common commercial product that employs phase changing is the heat pipe. Figure 3.11 shows the thermal cycle inside a heat pipe. Working fluid inside heat pipe absorbs thermal energy and evaporates. A pressure gradient exists and vapor migrates to lower temperature end. Vapor condenses and since internal surfaces of the pipe are lined with a porous medium (wick) that acts as a passive capillary pump, working fluid flows back to higher temperature end.

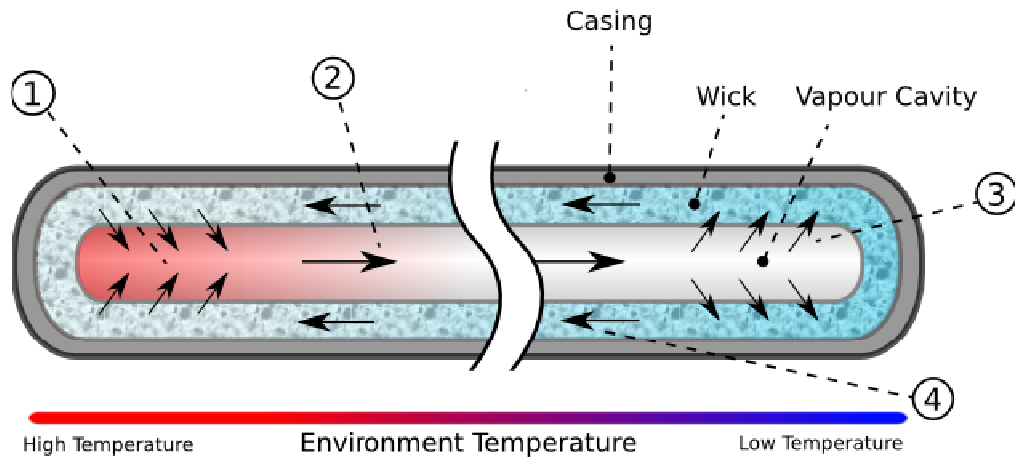


Figure 1.11 Thermal cycle inside a heat pipe [15]

Instead of employing fluid to cool down heated spots, solid-state cooling is also possible. A thermoelectric or a Peltier cooler is a small electronic heat pump that has the advantage of no moving parts and a silent operation [12]. Coolers can be activated by applying direct current and they can be used as heaters as well by reversing the direction of current flow. A schematic of simple thermoelectric module is given below. As shown in this figure when a DC potential difference is applied, electrons travel from the p- to the n- type thermoelement and heat is absorbed. One disadvantage is that the other side of the module is hot and since the power delivered is lost, the amount of energy dissipated on the hot side is greater than the amount of energy absorbed from cool side by I^2R .

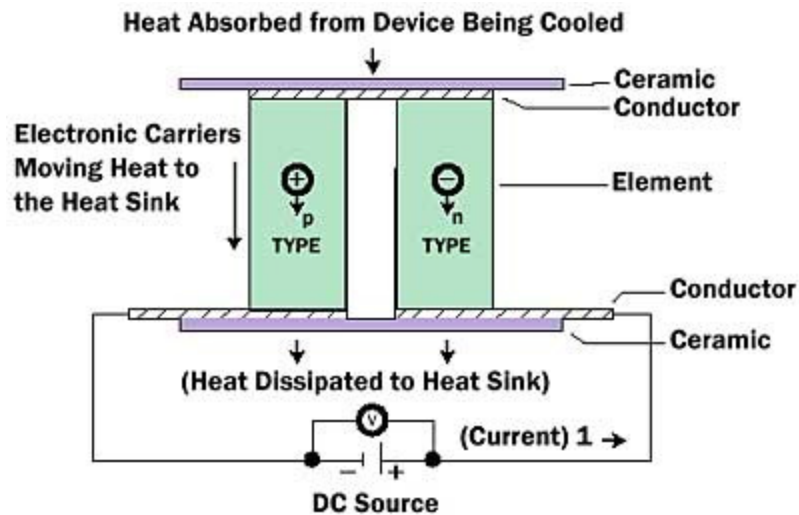


Figure 1.12 Schematic of simple Peltier cooler [12]

Other types of cooling methods such as magnetic refrigeration, cryogenic microcooling, thermotunneling, thermoacoustic refrigeration can properly not be used for the cooling of electronic components since these types lack the appropriate temperature range, practical dimensions or continuous cooling capability [7].

1.3 Goal of the Study

This study deals with two different types of possible future high-heat-flux cooling techniques. Namely; minichannels – microchannels and metal foam heat sinks. According to different classifications reviewed in later chapters, channels produced may be named as minichannels or microchannels, for simplicity channels fabricated for this thesis will be mentioned as microchannels from now on. For microchannels, there have been numerous studies that dealt with only numerical modeling, only experimentally investigation or that coped with both. The main purpose of the study is to make an extensive experimental comparison of thermal and fluidic characteristics of these two types of heat sinks. By just looking to the literature, it is possible to make only a rough judgment of their performances. However, as it will be seen in later chapters, there is no agreement on characteristics especially for microchannels on the literature. In the case of

open cell metal foam heat sinks, numerical models have restricted success and a limited number of successful experiments done on the subject. Boomsma [16] has an extensive work that successfully describes the performances of metal foam heat sinks. Since there is no full agreement between numerical models and experimental work, making an experimental comparison of characteristics between these two types of heat sinks come into prominence and this study fills the void in the literature. Performances of fabricated microchannels are compared to the performances of fabricated open cell metal foam heat sinks. All of these heat sinks have exactly the same size and shape, the main difference is the test section that four of them are filled with microchannels, while eight of them are filled with metal foams. So, actual comparison of characteristics of different types of metal foam and microchannel heat sinks can be made if proper experimental apparatus is used, which is the aim of this study.

CHAPTER 2

AN OVERVIEW OF STUDIES ON HEAT TRANSFER AND FLUID FLOW IN MICROCHANNELS AND METAL FOAM HEAT SINKS

Aim of this chapter is to have an overview of studies made on microchannel and metal foam heat sinks. Since these studies for microchannels and metal foam heat sinks were made separately, topics were examined in separate sub-chapters accordingly.

2.1 Heat transfer and fluid flow studies on microchannels

With developing microelectronics, high heat fluxes have been dramatically increased, while the area of heat dissipation continued to decrease. In addition to these difficulties, limitations on operational temperatures made problem even worse and a challenging problem emerged. Some techniques developed to deal with this difficulty were concisely investigated in the former chapter. One of the solutions was using microchannels as heat sinks.

Numerous studies have been made on microchannels that examine microchannels both theoretically and experimentally. Although there is no exact agreement on the results of those studies, one thing is remained a fact that when using as cooling devices, microchannels have high capacity of carrying heat away. It has been reported that they are capable of dissipating of heat load of 1000 W/cm^2 with a maximum surface temperature of 120°C [17]. This high level of capability attracted many researchers and subjects of fluid flow and heat transfer studied deeply focusing on different aspects like gas flow, liquid flow, single-phase flow,

boiling and two-phase flow inside microchannels. After a general overview on microchannels in the following sub-chapter, only studies related to this study that examine single-phase fluid flow experimentally will be overviewed in order to be purpose driven.

2.1.1 Concept of microchannel and related early work

First studies on microchannels that had a purpose of using them as heat sinks can be said to be done by Tuckerman and Pease [18]. Main aim of the study was to find a solution to the heating problem to very-large-scaled integrated (VLSI) circuits. To achieve that purpose, microchannels of 50 μm wide and 300 μm deep channels are directly etched onto silicon substrate of 1 cm x 1 cm area. According to that study, 790 W/cm^2 of heat load has been carried away with a substrate surface temperature difference of 71 $^{\circ}\text{C}$ above the inlet water temperature. As a result of experiments, thermal resistance values were obtained for different flow rates.

First work on friction factor of microchannels was conducted by Wu and Little [19, 20]. Channels widths were changing from 130 to 300 μm and depths were in the range of 30 to 60 μm . During experiments, employed fluid was nitrogen. Change in friction factor with Reynolds number was observed to be different than conventional Moody's chart derived for conventional channels. This deviation was attributed to the relative roughness being larger in small channels when compared to conventional ducts. An early transition to turbulent flow was also reported at a Reynolds number of nearly 400. New friction factor correlations both for laminar and turbulent flows and Nusselt number correlation for turbulent flows were proposed by the researchers.

Phillips et al. [17] was presented a theoretical model for developing and fully developed flows in microchannel heat sinks. The aim of the study was to develop a thermal-resistance model to calculate thermal and fluid flow performance of microchannels in the laminar and turbulent regimes. Material of microchannel heat sink was silicon and cooling liquid was water. It has been shown by

Tuckerman [21] that a good agreement was found with experimental data in laminar flow range.

Another experimental study has been done by Pfahler et al. [22] on microchannels. The purpose was to find out the channel size at which the conventional friction factors are no longer valid. It has been reported that for a microchannel with 100 μm width and 0.8 μm depth, experimental results were greatly deviated from conventional predictions. For another channel with the same width but with a 1.7 μm depth, a better agreement with conventional prediction has been found. Researchers concluded that there exists a critical channel size below which the conventional theory for friction factor no more holds.

In an experimental work carried out by Choi et al. [23], inner wall surface roughness, friction factors and convective heat transfer coefficients for flow of nitrogen gas in microchannels has been measured. Channel diameters were changing from 3 to 81 μm . Experimentally determined friction factor and heat transfer coefficient were significantly different from the conventional correlations. Friction factor dependence has been found as $53/Re$ instead of $64/Re$ in laminar flow regime. Measured friction factors in turbulent flow regime were lower than predicted. As a result of experimental measurements, new correlations offered by authors both for laminar and turbulent flow regimes. Contradictions with conventional heat transfer equations were also observed. New correlations of Nusselt number both for laminar and turbulent flow regimes were given.

Rahman and Gui [24] studied the fluid flow and heat transfer in series and parallel pattern of microchannels etched into silicon. Six microchannels of different aspect ratios were studied. Water was used as coolant. Local and average Nusselt numbers values were measured and results were compared with conventional correlations. Diversities identified. At the end of the study, it was concluded by researchers that microchannels have greater performance when compared to

other effective convective cooling methods like jet impingement and forced convection.

Until this point, early studies on microchannel have been briefly reported. These works included theoretical and analytical studies. In following sub-chapter, only studies related to this thesis will be investigated.

2.1.2 Experimental studies with single-phase flow

Both theoretical and experimental works have been reported about single-phase and two-phase flow and heat transfer in microchannels. Since examining all works that has been published is a tedious effort, only experimental studies that include single-phase flow and uses liquid as coolant will be observed in this section. Some of these studies reported agreement with conventional correlations and analytical solutions while some others reported discrepancies and offered new correlations based on experiments.

Experimental investigation on rectangular microchannels with water as coolant was done by Peng et al. [25,26]. Channels were fabricated from stainless steel and hydraulic diameters were changing from 0.133 to 0.367 mm and aspect ratios were between 0.33-1. Under steady-state conditions, liquid flow rates, liquid temperatures, inlet and outlet pressures, wall surface temperatures and heat input value to substrate were recorded during experiments. Deviation from classical correlations was observed both for laminar and turbulent flow regimes. An early transition between Reynolds numbers of 200 to 700 was reported. As a result, new correlations for Nusselt number were proposed.

Wang and Peng [27] made experiments with microchannels fabricated into stainless steel substrate with water and methanol as coolants. All channels had the same depth of 0.7 mm but different widths of 0.2, 0.4, 0.6 and 0.8 mm, and wall thicknesses also varied from 0.24 to 4 mm. For flows with different Reynolds numbers of flow, heat transfer coefficients and Nusselt numbers versus wall temperatures and Reynolds numbers graphs were plotted. An early transition to

turbulent flow was observed at Reynolds numbers of 1000 to 1500. Discrepancy from classical correlations has been reported.

Peng and Peterson [28] was investigated single phase forced convection with binary mixtures of water and methanol in microchannels having hydraulic diameters of 0.133 to 0.367 mm. Transitions to turbulent flow regime were observed to start at Reynolds numbers ranging from 70 to 400 and fully developed turbulent heat transfer conditions was achieved in the Reynolds number range of 200 to 700. As the hydraulic diameter has been reduced, transition Reynolds number has been observed to decrease. Hydraulic diameter and aspect ratio of channel were argued as the significant variables. Additionally, concentration of the mixture has also been said to be a factor that influenced heat transfer coefficient as expected.

Harms et al. [29] also studied heat transfer and fluid flow in a rectangular microchannels having a width of 251 μm and a depth of 1030 μm with deionized water. Critical Reynolds number was observed to be 1500. This early transition value found was attributed to inlet manifold conditions and channel roughness. Experimental results have been reported to be in good agreement with classical heat transfer correlations. As expected, volumetric flow rate, pressure drop and pumping power were inversely related to the thermal resistance and thermal resistance was reported to be smaller for deeper channels.

Adams et al. [30] made experiments with copper based microchannels produced by wire EDM. Distilled water used as coolant and turbulent flow regime was investigated. Measured Nusselt numbers were higher than those calculated by traditional correlations for large channels. It has been also reported that this discrepancy also became larger than predictions with the increasing Reynolds number. A new correlation for Nusselt number has been proposed as a result.

Qu and Mudawar [31] have conducted pressure drop and heat transfer experiments on a copper microchannel that have a width of 231 μm and depth of 713 μm . Single-phase experiments were made with deionized water. Two

different heat flux levels of 100 W/cm^2 and 200 W/cm^2 have been applied. Reynolds number ranged from 139 to 1672 for the first heat flux level and 385 to 1289 for the second heat flux level. Results were compared with the numerical analysis done and a good agreement have been reported. Researchers concluded that conventional Navier-Stokes and energy equations can be used to predict fluid flow and heat transfer characteristics of microchannel heat sinks.

Both single-phase and two-phase flows were investigated in small rectangular channels using FC-84 as test fluid by Warrier et al. [32]. Test section of the specimen used in experiments consisted of five parallel channels with a hydraulic diameter of 0.75 mm and a length to diameter ratio of 409.8. Uniform heat fluxes applied at top and bottom surfaces. Mass flow rate, inlet liquid temperature and heat flux were changed during experiments. For single-phase forced convection, experimental results for Nusselt number have been reported to be in agreement with results coming from conventional equations especially near the fully developed region. Close to the channel entrance, a discrepancy of %25 has been observed. This difference in the developing region was attributed to uncertainty of measurements and entrance effects.

Papautsky et al. [33] studied the effects of aspect ratio of microchannels on laminar friction constant experimentally. Rectangular metallic pipette arrays were used with widths varying from $150 \mu\text{m}$ to $600 \mu\text{m}$ and height ranging from $22.71 \mu\text{m}$ to $26.35 \mu\text{m}$. In order to reduce the entrance effects, a downstream port for static pressure measurements was also employed. Water was used as test fluid. At Reynolds numbers below 10, a 20 % increase in the friction factor when compared to macroscale predictions from the classical Navier-Stokes theory has been reported especially for low aspect ratios.

Liu and Garimella [34] investigated liquid flow in microchannels both experimentally and numerically. Microchannels with hydraulic diameters ranging from $244 \mu\text{m}$ to $974 \mu\text{m}$ at Reynolds numbers ranging from 230 to 6500 were tested. Pressure drop data was corrected by using global measurement for inlet and exit

losses. Onset of turbulence was visualized so that critical Reynolds number was investigated adequately. Agreement with classical theory has been reported and early transition to turbulent flow was not observed.

An experimental investigation of thermal behavior in single-phase flow through rectangular microchannels was conducted to explore the validity of classical correlations based on conventional sized channels by Lee et al. [35]. Width of microchannels tested varied from 194 μm to 534 μm . Aspect ratio was held constant for each channel as five. Deionized water was used as the test fluid and Reynolds numbers were varied from 300 to 1500. Results obtained were in good agreement with classical theory and an average deviation of only 5 % was observed. Finally, it has been concluded that when the entrance and boundary conditions imposed in the experiment were carefully matched, classical correlations can be used to predict the heat transfer in microchannels.

Akbari et al. [36] studied pressure drop in rectangular channels experimentally and compared the results to the ones obtained by using conventional channel flow equations. Microchannels were fabricated using soft lithography from polydimethylsiloxane with aspect ratios ranging from 0.13 to 0.76. Pressure drop data were measured for Reynolds numbers varying from 1 to 35. A discrepancy of 10 % has been reported from analytical model. Researchers concluded that a good agreement has been observed with the classical theory. They also emphasized that Poiseuille number, which is the product of the friction factor and Reynolds number product, $f\text{Re}$, only changes with aspect ratio of channels in laminar flow and it is constant over the wide range of microchannel cross-section aspect ratio.

As a conclusion of this section it can be said that experimental investigations on single-phase flow in microchannels have focused mainly on two aspects which are the effect of the channel dimensions and geometry on flow and heat transfer [37]. Some of them have been reported early transition to turbulent flow and offered new critical Reynolds numbers while some others concluded that there is no early transition. Some researchers proposed new correlations for pressure drop

calculation and heat transfer. However, according to recent trend, classical theory is applicable for pressure drop and heat transfer calculations and they can be employed in microchannel studies.

A very useful summary table of microchannel studies in the literature has been prepared by Garimella and Sobhan [37] given in Table 2.1. Only related part of the table was presented in this study.

Table 2.1 Summary of microchannel studies in the literature [37]

Configuration/Parameters	Nature of Work	Observations/Conclusions	Reference
MICROCHANNEL CONCEPTS AND EARLY WORK			
Rectangular cross section; water in silicon $W=50\ \mu\text{m}$ $H=300\ \mu\text{m}$, $Q=4.1, 6.5, 8.6\ \text{cm}^3/\text{s}$	Experiments on integral heat sink for silicon integrated circuits	Demonstrated use of microchannels for very high convective heat transfer in cooling integrated circuits ($790\ \text{W}/\text{cm}^2$ at a substrate-to-coolant temperature difference of 71°C)	Tuckerman & Pease [18]
Microchannels in cooling of integrated circuits	Microchannel fabrication and implementation details discussed	- Coolant selection, packaging/headering, microstructure selection, fabrication and bonding discussed - Etching and precision-sawing compared; fabrication and advantages of 'micropillars' using precision-sawing discussed - Expressions for Coolant Figure of Merit provided: CFOM = $(k_c \rho C / \mu)^{0.25}$ for given coolant pressure, and $(k_c \rho^2 C^2 / \mu)^{0.25}$ for given pumping power	Tuckerman & Pease [38]
Trapezoidal; nitrogen in silicon and glass $W=130\text{-}300\ \mu\text{m}$, $H=30\text{-}60\ \mu\text{m}$, $Dh=55\text{-}76\ \mu\text{m}$	Friction factors measured and compared with Moody's chart values for commercial channels	- Friction factor for glass channels 3-5 times larger than smooth-pipe predictions - Flow transition occurred at $Re \approx 400$ - Correlations for friction factor f: $(110 \pm 8)/Re$ $0.165(3.48 - \log Re)^{0.24} + (0.081 \pm 0.007)$ $(0.195 \pm 0.017)/Re^{0.11}$ $Re < 900$ $900 < Re < 3000$ $3000 < Re < 15000$	Wu & Little [19]

Table 2.1 (Continued)

As in [19]	Heat transfer experiments	Correlation for Nusselt number in the turbulent regime: $Nu = 0.0022 Pr^{0.4} Re^{1.09}$ for $Re > 3000$	Wu & Little [20]
Rectangular; air in silicon $W = 0.13\text{-}0.25\text{ mm}$, $H/W = 10$, $A_s = 47\text{-}63\text{ cm}^2/\text{m}^3$	Comparison of performance with conventional heat sinks, based on correlations	Micro-structured compact heat sinks attractive compared to conventional air circulation heat sinks	Mahalingam & Andrews [39]
Rectangular; water in silicon $W = 50\text{-}600\text{ }\mu\text{m}$	Theoretical model for fully developed, developing flows	Turbulent flow designs showed equivalent or better performance compared to laminar flow designs	Phillips et al. [17]
Rectangular; N-propanol in silicon $A_c = 80\text{-}7200\text{ sq. }\mu\text{ m}$	Experiments	- Channels with larger cross-sectional areas showed better agreement with theoretical predictions for the friction factor - Proposed $f = C/Re$ with C given as C vs Re graphs (laminar)	Pfahler et al. [22]
Microchannel structures for cooling applications	Microchannel applications discussed	Applications of microchannels to electronics cooling, compact heat exchangers, heat shields and fluid distribution systems discussed	Hoopman [40]

Table 2.1 (Continued)

Microtubes; nitrogen in silica $D = 3, 7, 10, 53, 81 \mu\text{m}$, $L = 24\text{-}52 \text{ mm}$	Experiments on friction and heat transfer	Correlations for friction factor and Nusselt number: Laminar ($Re < 2000$) $f = 64 / Re [1 + 30(v/Dc_a)^{-1} Pr^{1/3}]$; $Nu = 0.000972 Re^{1.17} Pr^{1/3}$; Turbulent ($2500 < Re < 20000$) $f = 0.140 Re^{-0.182}$ $Nu = 3.82 \times 10^{-6} Re^{1.96} Pr^{1/3}$	Choi et al. [23]
Rectangular; water in etched silicon $W = 1 \text{ mm}$, $H = 176\text{-}325 \mu\text{m}$, $L = 46 \text{ mm}$, $P = 2 \text{ mm}$	Experiments	Nusselt numbers higher than those predicted from analytical solutions for developing laminar flow	Rahman & Gui [24]
SINGLE-PHASE (LIQUID) EXPERIMENTS			
Rectangular; deionized water in stainless steel $W = 0.6 \text{ mm}$, $H = 0.7 \text{ mm}$, $T_i = 30\text{-}60^\circ\text{C}$, $v = 0.2\text{-}2.1 \text{ m/s}$	Experiments on single-phase forced convection	- In single-phase convection, a steep increase in wall heat flux with the wall temperature - Heat flux for microchannels higher than for normal-size tube	Peng & Wang [41]
Rectangular; water, methanol in stainless steel $W = 0.2, 0.4, 0.6, 0.8 \text{ mm}$, $H = 0.7 \text{ mm}$, $T_i = 10\text{-}35^\circ\text{C}$ (water), $14\text{-}19^\circ\text{C}$ (methanol), $v = 0.2\text{-}2.1 \text{ m/s}$	Experiments on forced convection flow and heat transfer	- Heat transfer augmented as liquid temperature was reduced and as liquid velocity was increased - Fully developed turbulent convection regime starts at $Re = 1000\text{-}1500$ - Correlation for turbulent heat transfer $Nu = 0.00805 Re^{4/5} Pr^{1/3}$	Wang & Peng [27]

Table 2.1 (Continued)

Rectangular; water in stainless steel $D_h = 0.133\text{-}0.367$ mm, $L = 50$ mm, $H/W = 0.333\text{-}1$, $T_i = 22\text{-}44^\circ\text{C}$, $v = 0.25\text{-}12\text{m/s}$, $Re = 50\text{-}4000$	Experiments on frictional behavior in laminar and turbulent flow	- Flow transition occurred for $Re = 200 - 700$ - Correlations proposed (values for $C_{f,i}$, $C_{f,t}$ provided in [42, Table 2]) $f = C_{f,i}/Re^{1.98}$ Laminar flow $f = C_{f,t}/Re^{1.72}$ Turbulent flow	Peng et al. [42]
As in [42]	Experiments on forced convection heat transfer characteristics	- Fully turbulent convective conditions reached at $Re = 400 - 1500$ - Transition Re diminished with a reduction in microchannel dimension $Nu = C_{h,i} Re^{0.62} Pr^{1/3}$ Laminar $Nu = C_{h,t} Re^{0.8} Pr^{1/3}$ Turbulent (values for $C_{h,i}$, $C_{h,t}$) available in [43]	Peng et al. [43]
As in [27] except $T_i = 11\text{-}28^\circ\text{C}$ (water), $12\text{-}20^\circ\text{C}$ (methanol) $v = 0.2\text{-}2.1$ m/s (water), $0.2 - 1.5$ m/s (methanol)	Experiments on effect of thermofluid properties and , geometry on convective heat transfer	- Changes in flow regimes and heat transfer modes initiated at lower Re in microchannels compared to conventional channels - Transition zone and heat transfer characteristics in laminar and transition flow influenced by liquid temperature, velocity, Re and microchannel size	Peng&Peterson [44]

Table 2.1 (Continued)

As in [25]	Experiments on single-phase flow and heat transfer	- Ratio of experimental to theoretical friction factor at critical Re plotted as a function of $Z = \min[H, W]/\max[H, W]$ - Correlations proposed $Nu = 0.1165 (D_h/P)^{0.81} (H/W)^{-0.79} Re^{0.62} Pr^{0.33}$ Laminar $Nu = 0.072 (D_h/P)^{1.15} [1 - 2.421 (Z - 0.5)^2] \times Re^{0.8} Pr^{0.33}$ Turbulent	Peng&Peterson [45]
Rectangular; water-methanol mixture in stainless steel $D_h = 0.133$ - 0.367 mm, $L = 50$ mm $W = 0.1, 0.2, 0.3, 0.4$ mm, $H = 0.2, 0.3$ mm, $T_i = 14$ - 36°C , $v = 0.04$ - 3.8 m/s, $Re = 6$ - 3500	Experiments	- Laminar heat transfer ceased for $Re \approx 70$-400 depending on flow conditions; fully developed turbulent heat transfer achieved at $Re = 200$-700, depending on D_h - Transition Re reduced with a reduction in microchannel size D_h, H/W and mixture mole fraction influenced heat transfer - Heat transfer increased for smaller mole fractions of the more volatile component	Peng&Peterson [46]
Rectangular; deionized water in silicon $W = 251$ μm , $H = 1030$ μm , $D_h = 404$ μm , $L = 2.5$ cm, $Q = 5.47$ - 118 cm^3/s	Experimental & theoretical study	- Critical Re of 1500 identified for onset of turbulence - Analysis showed that flow and heat transfer performance could be improved by increasing H, and that for the same pressure drop and pumping power, thermal resistance was smaller for deeper channels	Harms et al. [29]

Table 2.1 (Continued)

Rectangular; FC-72 and transformer oil in stainless steel $H = 0.10\text{-}0.58$ mm; nozzle dimensions (mm): Length = 35, $B = 0.146, 0.210, 0.234$, Height = 12, $v = 0.54\text{-}8.45$ m/s, $Re = 70\text{-}170$ (oil), 911-4807 (FC72)	Experiments in impingement on 2D microchannels	Empirical correlation for Nusselt number for the two liquids $Nu_x = 0.429 Re^{0.583} Pr^{1/3} (x/2H)^{0.349} (B/2H)^{-0.494}$	Zhuang et al. [47]
Circular; distilled water in copper $D = 0.102\text{-}1.09$ mm, $v < 18.9$ m/s, $Re = 2.6 \times 10^3 - 2.3 \times 10^4$, $Pr = 1.53\text{-}6.43$, $q'' < 3.0\text{MW/m}^2$	Experiments on turbulent single-phase flow	- Nusselt numbers higher than those predicted by large-channel correlations - Gnielinski [52] correlation modified for Nusselt number for turbulent flow in circular microchannels (f from Filonenko, 1954): $Nu = Nu_{Gn} (1 + F) \text{ where } F = CRe [1 - (D/D_0)^2]$$Nu_{Gn} = (f/8) (Re - 1000) Pr / [1 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)]$$C = 7.6 \times 10^{-5}; D_0 = 1.164 \text{ mm}, f = [1.821 \log(Re) - 1.64]^2$	Adams et al. [30]
Non-circular; water in copper $D_h = 1.13$ mm, $Re = 3.9 \times 10^3\text{-}2.14 \times 10^4$, $Pr = 1.22\text{-}3.02$	Experiments on turbulent convection	- Experimental Nusselt number well-predicted by Nu_{Gn} $Dh \approx 1.2$ mm proposed as reasonable lower limit for applicability of standard Nusselt-type correlations to non-circular channels	Adams et al. [48]

Table 2.1 (Continued)

Rectangular; laminar and transition flow	Dimensional analysis based on experimental data in the literature	- Attempted to explain the observation that Nu may decrease with increasing Re in laminar regime and may remain unaffected in transition regime - Proposed that Brinkman number may better correlate convective heat transfer	Tso & Mahulikar [49,50]
Almost circular; water in aluminum $D_h = 0.73$ mm	Experiments	Laminar flow data found to correlate well using Brinkman number	Tso & Mahulikar [51]
Rectangular copper; laminar and transition flow $W = 231$ μm , $H = 713$ μm , $Re = 139$ - 1672 for $q'' = 100\text{W}/\text{cm}^2$, $Re = 385$ - 1289 for $q'' = 200\text{W}/\text{cm}^2$	Experimental and numerical investigation	- Measured pressure drop data is in good agreement with predicted values - Early transition to turbulent flow was never observed - Very good agreement was achieved between experimental data and numerical predictions for the temperature distribution within the sink	Qu & Mudawar [31]
Narrow rectangular; FC-84 in aluminum laminar and transition flow $D_h = 0.75$ mm, $L/D_h = 409.8$, $T_i = 26$ $^{\circ}\text{C}$, 40 $^{\circ}\text{C}$, 60 $^{\circ}\text{C}$, $q'' = 0 - 5.99$ W/cm^2 , $Re = 418 - 2015$	Experimental investigation of single-phase and flow boiling	- Early transition to turbulent regime was observed between $500 \leq Re \leq 1500$ - Friction factor was well predicted by classical theory Nu number agreed with conventional theory within 25 %. Discrepancy was attributed to uncertainties and channel entrance effects	Warrier et al. [32]

Table 2.1 (Continued)

Rectangular; water in nickel laminar flow $W = 150\ \mu\text{m} - 600\ \mu\text{m}$, $H = 22.71\ \mu\text{m} - 26.35\ \mu\text{m}$	Experimental fluid flow investigation	- For small channels friction constants were approximately 20 % higher above predicted values - As channels getting larger agreement was also better for friction factors with theoretical predictions	Papautsky et al. [33]
Rectangular; Water in plexiglass laminar and turbulent flow $D_h = 250 - 1000\ \mu\text{m}$, $L = 25.4 - 41\ \text{mm}$, $Re = 230 - 6500$	Experimental fluid flow investigation	- No early transition to turbulent flow was observed - Friction factor and pressure losses predictions were in agreement with classical theory for laminar flow - For the turbulent regime satisfactory correlations for the losses are not readily available	Liu & Garimella [34]
Rectangular; water in copper laminar and turbulent flow $W = 194\ \mu\text{m} - 534\ \mu\text{m}$, $H/W = 5$, $Re = 300 - 3500$, $q'' \approx 45\ \text{W}/\text{cm}^2$	Experimental heat transfer investigation	Conventional analysis approach can be employed in predicting heat transfer behavior in microchannels of the dimensions considered in study	Lee et al. [35]
Rectangular; water in polydimethylsiloxane laminar flow $H/W = 0.13 - 0.76$, $W = 134\ \mu\text{m} - 780\ \mu\text{m}$, $Re = 1 - 35$	Experimental fluid flow investigation	- An agreement on frictional factor and Poiseuille number of 10 % with analytical model has been reported - Poiseuille number is only a function of aspect ratio of the channel considered - Pressure losses coming from developing region, minor flow contraction and expansion should be carefully included	Akbari et al. [36]

2.2 Heat transfer and fluid flow studies on metal foams

A more recent developing solution to cooling problems of microelectronics like computer chips, high frequency microwave systems, laser diodes, plasma processing, high energy beam dumps and stoppers and high-power optical structures is using metal foams as heat exchangers. Since it is a newer concept compared to employing microchannels as heat sinks, studies are limited. However some researchers have extensive publication on the topic and some related work will be presented in this section. Although many studies has been done on fluid flow in porous media, among these studies ones only deals with metal foam will be examined since it is more related to this study.

Antohe et al. [53] investigated fluid flow experimentally in metal foam that were considered to be used for cooling purposes of phased-array radars modules which is exactly the same inherent purpose of the present thesis. The purpose of mentioned work was to determine permeability and pressure drop in compressed metal foams accurately. A considered new design for cooling high frequency microwave system was consisting of brazing a mechanically compressed microporous metallic layer inside the coolant passage of the sink device. Material of porous matrix was 6101-0 which had a capability of brazing to main body of heat sink. To evaluate the coefficients of Forchheimer-extended Darcy equation (will be presented in later chapters), -a quadratic equation that defines the pressure drop in porous media over a length-, least square quadratic curve fit of experimental data was used. Using this technique, instead of extrapolation of experimental data, reduced uncertainties dramatically. Dimensions of porous matrix (heat load carrying volume) have been chosen according to phased-array radar being considered. A 40 ppi (porosity per inches) metal foam block have been split and compressed mechanically with several compression ratios ranging from four to fourteen. All compressed foam had exactly the same final dimensions. Porosities were directly measured from compressed foam samples and compared with expected porosities calculated. Some discrepancies of 10% have been

observed and that was attributed to unconstrained lateral sides during mechanical compression. Nine different metal foam samples with different densities and compression ratios have been tested with air and poly-alpha-olefin fluid, synthetic oil commonly used in cooling military avionic devices, have been used as testing fluid.

Observations after experimental measurements can be listed as below:

- Blocks with compressed microporous layer behaved as a porous medium, that is, the pressure drop varied quadratically with speed.
- Hydraulic behavior of a compressed matrix seems to become insensitive to changes in the initial matrix density beyond a certain threshold value for a defined compression ratio.
- Pressure drop for the same initial density but for different compression ratio is more sensitive to changes in compression ratio at high flow speeds.
- The permeability of the matrix decreases with the increase in initial aluminum matrix density and with the increase of its compression ratio.
- Darcy flow model was invoked for computing the permeability and inertia coefficient of the matrices and these hydraulic parameters are fundamental for the design and optimization of the micro heat exchangers.
- Pre-studies in the field presented high degree of inaccuracy since they have employed extrapolation and visual graphical estimation. Instead, least squares quadratic curve fit was used which was more precise and consistent.
- Since linear and quadratic effects are always present, determined hydraulic parameters were only valid for testing velocity range from which they were determined. So, presented permeability and Forchheimer coefficients are only valid for mentioned porous media in the range of testing velocities.

Bhattacharya et al [54] presented analytical and experimental investigation for the determination of the effective thermal conductivity, permeability and inertial

coefficient of high porosity uncompressed metal foams. 5, 10 20, 40 ppi aluminum and reticulated vitreous carbon (RVC) foam with porosities ranging from 0.905 to 0.978 was investigated analytically and experimentally. Air and water was considered as medium fluid for conductivity analysis, while only air was employed to determine permeability and inertial coefficient for analytical, empirical and experimental works. In analytical modeling of foam blocks, lump of metal at the intersection of two fibers was represented as a circular 'blub' of metal at the intersection of two sides of hexagon. A theoretical model was formulated for determination of effective conductivity of solid-fluid interface. An empirical correlation was also derived for estimating the effective thermal conductivity in terms of porosity, and the solid-fluid conductivities. Also for permeability and inertial coefficient of uncompressed metal foams, analytical models were proposed. As a result of this study, following observations can be made:

- Previous experimental studies for defining three parameters mentioned above included incorrect estimations.
- Pressure drop along metal foam block was increased along the axial direction as predicted but pressure gradient did not vary which indicates that the flow for all samples tested was fully developed.
- Proposed analytical and empirical models for parameters compared with experimental results and good agreement has been reported.
- Study is valid only for uncompressed open cell foams with high porosities.

A series of studies on metal foam heat exchangers has been published by Boomsma and Poulikakos. One of these researches was about investigation of the effects of compression and pore size variations on the liquid flow characteristics in metal foams experimentally [55]. In this study, 6101-T6 metal foams of porosities 92 % and 95 % and 10, 20 and 40 ppi were studied with water. To investigate the effects of compressing metal foams by several factors on fluid flow characteristics through metal foams, foams were compressed with a factor ranging from 2 to 8. During compression, lateral sides were not restricted so measured porosities after

compression were different than predicted values and this discrepancy was tabulated. Since Darcy's law is applicable only for relatively slow flows and as the velocity increases, the form drag becomes more effective and the importance of adding drag term C as suggested by Dupuit [56] by with adding a quadratic term to Darcy's equations was emphasized. In order to find out permeability and form coefficient, like Antohe et al. [53], least squares quadratic curve fit was employed. A transition from linear Darcy regime to quadratic relation was observed as expected. To describe the transition of the pressure drop from linear to quadratic curve, three different Reynolds numbers specific to porous medium have been proposed from literature and these Reynolds number models will also be discussed in this thesis in later chapters. Flow rate dependence of permeability and form coefficient was also discussed. Observations derived as a result of the study can be listed as follows:

- Structure of the open-cell metal foam is completely different from that encountered in packed beds of spheres that were used to investigate behavior of porous media. Because of these structural differences, the characterization of the pressure-drop through the open-cell metal foams requires renewed research effort.
- When measured pressure drop data after experiments were investigated, it has been observed that the pressure drop across the foam is a quadratic function of the flow velocity both for compressed and uncompressed foams.
- A transition from linear Darcy regime to quadratic relation was observed as expected.
- The effect on permeability of metal foams being originally 92 % and 95 % did not have an important effect. The post-compression porosity dominates the permeability and resulting pressure drop.
- Increasing the compression factor decreased the permeability of the foam. The trend was nearly the same for 92 % and 95 % metal foams.

- As porosity was increased, the permeability has become more sensitive to changes in flow velocity.
- For uncompressed metal foams that have same porosities but smaller pore sizes, flow resistance increased by reducing the permeability and increasing the form coefficient. This increase was attributed to the higher specific surface area generated by the smaller pore size.
- Reynolds number that is defined with the square root of permeability, K , was offered as appropriate Reynolds number for metal foams since it provides narrowest transitional Re number range. This Reynolds number is found by pressure drop data points from a zero velocity value up to and including the velocity at which the transition to the quadratic regime takes place.
- Transition regime from the linear Darcy regime to quadratic flow regime for all metal foams specimen tested range between unity and 26.5 based on the calculation of the permeability and form coefficient in this range.
- Narrowest range of Re was obtained when the permeability calculated using a flow rate ranging from zero to the flow velocity at which transition occurred.
- For another analysis using a different flow range will result in different permeability and form coefficient values. The flow velocity range over which these terms are calculated must also be stated for accuracy when permeability and the associated form coefficient are stated [53].

In another study of Boomsma et al. [57], the thermal performance of metal foams as heat exchangers has been investigated experimentally. 40 ppi 6101-T6 aluminum alloy metal foams with initial porosity of 92 % and 95 % were tested. As in [55], metal foams compressed by a ratio of 2 to 8. To use metal foams as heat sinks, metal foams were brazed to AA6092 aluminum heat spreader plate. Pressure drops have been recorded and Fanning friction factor of each sample have been calculated using experimental data. Nusselt numbers were also

determined by using inlet-outlet coolant temperatures and amount of heat delivered to heat sink. In order to compare the performances of new generation heat sinks produced with heat exchangers in the market, Colburn factors that gives a heat transfer performance estimate comparing the convection coefficient to the required flow rate of a heat exchanger has been calculated. Again for comparison purpose of thermal performances, pumping power versus thermal resistances graph has been plotted. The heat transfer rate for all varying flow conditions remained unchanged. Water and 50 % water-ethylene mixture have been used as testing fluid for comparison with exchanger in the market. The following conclusions were made by authors:

- Compressed aluminum metal foam performed well and made a significant improvement in the efficiency over several available heat exchangers which operate under nearly identical conditions.
- Metal foam heat exchangers decreased thermal resistance nearly by 50 % compared to currently used heat exchangers designed for the same application.

Zhang et al. [58] experimentally investigated fluid flow and heat transfer characteristics of 60 and 100 ppi copper metal foams with four different porosities ranging from 0.6 to 0.9. Among these porosities, the ones with porosities of 0.8 and 0.9 were directly produced without any compression by the manufacturer. Porosities of 0.6 and 0.7 were obtained by compressing highly porous foams according to certain ratios. To form foam heat sinks, all metal foams were brazed into the copper housings. Actually it is the vacuum brazing operation that turns metal foams into a heat sink. Power input has been held constant during all experiments. De-ionized water was used as the testing liquid. Also similar to this study, a microchannel with a height of 2 mm and a width of 0.2 mm tested and characteristics were compared with other metal foam heat sinks. Finned area of microchannel heat sink was similar to metal foam area and the flow and heat transfer characteristics of the microchannel heat sink were examined under the

same conditions as the metal foam heat sinks having an identical thermal interface material and the same power output. For all examined heat sinks, thermal resistance, pressure drop and pumping power were calculated. Illustrative graphs like pressure drop versus thermal resistance and pump power versus thermal resistance were presented to compare the overall performance of heat sinks. Also inertia coefficients and permeability constants of metal foams have been found. Several conclusions can be made after this research as follows:

- With increasing flow rate, thermal resistances of all metal foams were decreased as expected.
- As porosities were decreased, thermal resistances at a given flow rate were also observed to decrease. Porous foam materials with low porosities have higher solid content and larger fiber strut diameter surrounding the pore and thus enlarged solid surface area. The increased fiber strut size also helps to increase the effective thermal conductivity of the foam structure that is equivalent to the increase in fin efficiency for a finned heat sink.
- Low porosity also led to a higher flow resistance since the cross-sectional area for low porosity was also reduced. It has also been observed that pressure drops can be correlated as quadratic functions of flow rates.
- It was striking to observe that for the 60 ppi metal foams, the one with a porosity 0.8 had the lowest thermal resistance at the given pressure drop. It has also been observed that for lower pressure drops examined microchannel heat sinks had lower thermal resistance values over most of the flow rate range due to its higher solid content and effective transverse thermal conductivity. Thermal resistance values of metal foam heat sinks have been observed to decrease more rapidly when compared to microchannel heat sinks. As a result of that, thermal resistance versus pumping power curve of the metal foam heat sink of porosity 0.8 intersects with that for the microchannel and microchannel heat sinks at larger pumping power.

- For 100 ppi metal foams porosity of 0.9 has given the lowest thermal resistance among others for a given pressure drop.
- Manufacturability of heat sinks also plays an important role like thermal and hydraulic characteristics. Production of foam heat sinks with high foam height is an easy process with brazing when bulk material removal techniques were considered to manufacture a microchannel heat sink.

Another experimental study on metal foam for cooling purposes has been done by Hetsroni et al. [59]. The purpose of the research was to find out if metal foams could be employed as a transmission window. Transmission window is employed for dividing environment from vacuum space where high-energy particle beam is accelerated, from the atmospheric conditions, where it could be applied for irradiation surfaces, impact ionization, medical equipment sterilization, material joining and cutting etc. 40 ppi aluminum metal foam (AA6106-T6) has been used with a porosity of 85%. To represent working conditions of transmission window, inner heat generation was simulated by supplying D.C. power to the metal foam through copper rods. Water has been employed as testing liquid. Heat flux ranging from 0.02 to 0.25 kW/cm² has been applied and liquid velocity varied between 0.1 and 1.2 m/s. Permeability and inertia coefficients have been calculated from experimental data by fitting quadratic polynomial. To find pressure drop of the metal foam only, pressure drop of clear channel without the insert has been subtracted from the pressure drop of the channel with porous insert. Colburn factor has also been computed to compare heat transfer performance with other metal foam heat sinks studied by Boomsma [60]. Obtained results were compared to other studies on metal foams. Some important observations of study can be listed as follows:

- Heat flux up to 0.24 kW/cm² was obtained experimentally for a Reynolds number of 3800 with a temperature of 45 K difference between the wall and coolant liquid of. Pressure drop of above experiment was 20 bar/m.

- Calculated permeability and inertia coefficient values have been compared to the values calculated in another study of Bhattacharya et al [54]. Results were found to be in good agreement.
- When Colburn factor of studied metal foam heat sink was compared to that of compressed metal foam heat sinks of Boomsma's work [60], it has been observed that some compression of metal foam was useful to enhance heat sink performance at low fluid velocities. As Reynolds number increases, compression factor becomes independent of compression ratio.
- Study has been demonstrated that performance of metal foam heat sinks have the potential of using porous foam for cooling transmission window with internal heat generation.

Noh et al. [61] tested aluminum foam to determine fluid flow and thermal characteristics in an annulus. 10 ppi aluminum metal foam with a porosity of 0.90 has been placed between two concentric tubes. Outer wall was heated with a constant heat flux. Water was used as coolant. Inlet temperature of coolant has been held constant. A power input between 13.6 to 31.4 kW/m² has been applied and Reynolds number based on the permeability ranging from 2 to 150. As a result of the study, experimental data has been compared to the correlations developed to describe fluid flow and heat transfer in the literature. Newer friction factor and Nusselt number correlations were proposed.

N. Dukhan [62] studied the fluid flow characteristics of metal foams experimentally. Nine different 6101-T6 aluminum alloy metal foams of 10, 20 and 40 PPI with porosities ranging from 67.9 % to 92.4 % investigated experimentally. Foam sample has been placed inside a wind tunnel and air was used as testing fluid. During experiments, tests were conducted in incompressible range of air for Mach numbers significantly less than 0.3. Pressure drop before and after the test section was measured. Permeability values and form drag coefficients were calculated for tested metal foams. Some related conclusions of the research are listed as follows:

- Pressure drop increased with increasing Darcian velocity following the quadratic Forchheimer equation.
- Significantly higher pressure drop has been observed for lower porosity foam.
- New correlations were proposed both for permeability and form drag coefficient and these correlations predicted results of some previous studies well.

An experimental and numerical study has been done on aluminum metal foam sandwich panels to investigate convective heat transfer and size effects of the metal foam core and the effects of foam thickness on convective heat transfer by Salas and Waas [63]. Four 6101-T6 aluminum alloy 20 ppi test specimen with 92 % porosity and a thickness of ranging from 6.4 mm to 25.4 mm were used. Air was driven into sandwich metal foam structure at velocities 0 to 10 m/s. The temperature and pressure of the air entering into and exiting from the foam core were also measured. One side of structure was heated. A numerical model was also developed. Some conclusions are as follows:

- As the thickness of the foam was increased, the temperature difference of two opposing sides also increased. This means that for higher thickness of metal foam, heat transfer was also enhanced. Also as expected temperature difference decreased as air velocity increased. This performance enhancement however diminished with further increase in foam thickness.
- The slope of temperature profile obtained numerically and experimentally deviated especially for smaller thickness values. For thicker metal foams, agreement was better.

A recent study was conducted by Mancin et al. [78] on fluid flow and heat transfer performance of open cell metal foam. The main purpose of the research was to determine foam height effects on heat transfer performance. During experiments 20 ppi aluminum open cell metal foams with the same heated areas but with two

different foam core heights, 20 and 40 mm were employed. Each sample was brazed between two aluminum plates. Air was used as the testing fluid. Three different constant specific heat fluxes of 25, 32.5 and 40 kW m⁻² were applied at one side of metal foam sandwich. The experimental measurements obtained during single phase air heat transfer. Experimental heat transfer coefficients, permeability and inertia coefficient values of tested foams have been calculated and compared against the predictions obtained from two models recently suggested by other researchers. Some observations and conclusions related to the study made by authors were listed as follows:

- Heat transfer coefficient has been observed to increase as the mass velocity increased.
- Heat transfer coefficients measured for 20 mm high sample were higher than those obtained for a 40 mm high foam.
- Two samples exploited almost the same values of pressure drop for a given air mass velocity. The 20 mm high foam exhibits slightly lower values of both permeability and inertia coefficient than those of the 40 mm high sample.
- Calculated heat transfer coefficients from developed model of two studies of Mancin et al. [79,80] were compared with experimentally obtained values of heat transfer coefficients. A good agreement was observed for the 40 mm high sample while the model underestimates the data for the 20 mm high sample.
- Again calculated pressure drop estimations from developed models of same references above were compared with experimental data. A good agreement was observed.

A very useful study was conducted by Mahjoon and Vafai [81] on fluid flow and heat transfer of metal foams. Related researches on the topic were investigated and correlations presented as a result of these studies were tabulated. Since fluid flow in open cell metal foams and employing them as high performance heat sinks

is a comparably novel research area, no full agreement was observed. Some researches presented correlations to find the friction factor, pressure drop or heat transfer coefficient (Nusselt number). These correlations were presented in Table 2.2 [81]. For details of correlations, reader should refer to the references given in the last column. Also Figure 2.1 was given after the table in order to clarify some dimensional parameters of correlations.

Table 2.2 - A synthesis of fluid and thermal transport models of metal foam heat exchangers [81]

Nature of Correlations	Eq. Nos.	Correlation(s)	References
Category 1: Microstructural based correlations for metal foam heat exchangers			
Friction factor as a function of Reynolds number based on permeability	(1)	$\frac{(\frac{\Delta P}{L})\sqrt{K}}{\rho u^2} = f = \frac{1}{Re_K} + 0.105 = \frac{\mu}{\rho u \sqrt{K}} + 0.105$	Paek et al. [64]
Empirical correlations for friction factor of airflow with different porosities at different fluid regimes	(2-3)	$\text{For } 30 < Re_{dp} < 300: f = 22 \frac{1-\varepsilon}{Re_{dp}} + 0.22 \quad \text{For } Re_{dp} \gg 300: f = 0.22$	Liu et al. [65]
	(4-5)	$Re_{dp} = \frac{D_p \rho u}{\mu} (1 - \varepsilon) = \frac{6 \rho u}{\mu S_v} (1 - \varepsilon) = \frac{3 \rho u d_p}{2 \mu \varepsilon} (1 - \varepsilon)^2 \quad \frac{\Delta P}{L} = f \frac{\rho u^2}{D_p} \frac{1 - \varepsilon}{\varepsilon^3}$	
Theoretical model for prediction of permeability and inertial coefficient	(6-7)	$\frac{K}{d^2} = \frac{\varepsilon^2}{36 \mathcal{X} (\mathcal{X} - 1)} \quad F = \frac{2.05 \mathcal{X} (\mathcal{X} - 1) \sqrt{K}}{\varepsilon^2 (3 - \mathcal{X})} \frac{d}{d}$	Du Plessis et al. [66]
	(8)	$d = d_p / \sqrt[3]{\frac{\mathcal{X}}{3}}$	
	(9)	$\frac{1}{\mathcal{X}} = \frac{3}{4\varepsilon} + \frac{\sqrt{9-8\varepsilon}}{2\varepsilon} x \cos\left\{\frac{4\pi}{3} + \frac{1}{3} \cos^{-1} \left[\frac{8\varepsilon^2 - 36\varepsilon + 27}{(9-8\varepsilon)^{\frac{3}{2}}} \right] \right\}$	

Table 2.2 (Continued)

Inertial parameter and permeability were modeled as functions of average pore and fiber diameters and porosity. Also a model was developed for $\frac{d_f}{d_p}$	(10) (11) (12)	$\frac{K}{d^2} = 0.00073(1 - \varepsilon)^{-0.224} \left(\frac{d_f}{d_p}\right)^{-1.11} \quad F = 0.00212(1 - \varepsilon)^{-0.132} \left(\frac{d_f}{d_p}\right)^{-1.63}$ $\frac{d_f}{d_p} = \sqrt{\frac{(1 - \varepsilon)}{3\pi}} \frac{1}{G} = 2 \sqrt{\frac{(1 - \varepsilon)}{3\pi}} \frac{1}{1 - e^{-(1 - \varepsilon)/0.04}}$ $\frac{d_f}{d_p} = 1.18 \sqrt{\frac{(1 - \varepsilon)}{3\pi}} \frac{1}{G} = 1.18 \sqrt{\frac{(1 - \varepsilon)}{3\pi}} \frac{1}{1 - e^{-(1 - \varepsilon)/0.04}}$ Look at the figure 2.1 for d_f and d_p.	Calmidi [67]
Correlation for the inertial coefficient as a function of porosity, tortuosity, shape factor and form drag coefficient. For permeability the model proposed by Du Plessis [66] was suggested.	(13-14) (15-16) (17)	$F = 0.095 \frac{C_{D(\varepsilon=0.85)}}{12} G^{-0.8} \sqrt{\frac{\varepsilon}{3(\chi - 1)}} (1.18 \sqrt{\frac{(1 - \varepsilon)}{3\pi}} \frac{1}{G})^{-1} C_{D(\varepsilon=0.85)}$ $= 1.2$ For $0.85 < \varepsilon < 0.97$: $G = 1 - e^{-(1 - \varepsilon)/0.04}$ For $\varepsilon \geq 0.97$: $G = 0.5831$ $\frac{1}{\chi} = \frac{\pi}{4\varepsilon} \left\{ 1 - \left(1.18 \sqrt{\frac{(1 - \varepsilon)}{3\pi}} \frac{1}{G} \right)^2 \right\}$ $= \frac{\pi}{4\varepsilon} \left\{ 1 - \left(1.18 \sqrt{\frac{(1 - \varepsilon)}{3\pi}} \frac{1}{1 - e^{-(1 - \varepsilon)/0.04}} \right)^2 \right\}$	Bhattacharya et al. [54]

Table 2.2 (Continued)

Empirical correlations for permeability and inertial coefficient	(18-19)	$K = a_1 \exp[b_1 \varepsilon] \quad F = (a_2 \varepsilon + b_2) \sqrt{K}$ a_1, a_2, b_1 and b_2 are constants.	Dukhan [62]
Empirical model for inertial coefficient and permeability estimation.	(20-21)	$K = \frac{\varepsilon^3 d_f^2}{\alpha(1-\varepsilon)^2} \quad \frac{F}{\sqrt{K}} = \frac{\beta(1-\varepsilon)}{\varepsilon^3 d_f} \quad \alpha: 100 - 865, \beta: 0.65 - 2.6$ Look at the figure 2.1 for d_f .	Tadrist et al. [68]
Theoretical model for inertial coefficient and from drag coefficient. For permeability model proposed by Du Plessis et al. [66] suggested.	(22)	$F = (3 - \mathcal{X})(\mathcal{X} - 1) \frac{C_D \mathcal{X}^{1.5}}{24 \varepsilon^3}$	Fourie and Du Plessis [69]
	(23)	$C_D = 1 + \left(\frac{10}{Re^{0.667}} \right) = 1 + 10 \left(\frac{\rho u d (\mathcal{X} - 1)}{2 \mu \varepsilon} \right)^{-0.667}$	
	(8-9)	$d = \frac{d_p}{\sqrt{3}} \quad \frac{1}{\mathcal{X}} = \frac{3}{4\varepsilon} + \frac{\sqrt{9-8\varepsilon}}{2\varepsilon} x \cos\left\{ \frac{4\pi}{3} + \frac{1}{3} \cos^{-1} \left[\frac{8\varepsilon^2 - 36\varepsilon + 27}{(9-8\varepsilon)^{\frac{3}{2}}} \right] \right\}$ Look at the figure 2.1 for d_p .	
Correlations for specific solid-fluid interfacial surface area and interfacial Nusselt number as a function of fiber and pore diameter and porosity.	(24)	$Nu_{sf} = \frac{h_{sf} d_f}{k_f} = C_T Re_{d_f}^{0.5} Pr^{0.37} = C_T \left(\frac{u d_f}{\varepsilon \nu} \right)^{0.5} Pr^{0.37} \quad (C_T = 0.52)$	Calmidi and Mahajan [70]
	(25)	$a_{sf} = \frac{3\pi d_f}{(0.59 d_p)^2} [1 - e^{-\frac{1-\varepsilon}{0.04}}]$ Look at the figure 2.1 for d_f .	
Correlation for pore diameter Nusselt number as a function of pore diameter and porosity.	(26)	$Nu_{DP} = \frac{h D_P}{k_{se}} = a Re_{DP}^b = a \left(\frac{\rho u D_P}{\mu} \right)^b \quad a \text{ and } b \text{ are constants}$	Shih et al. [71]

Table 2.2 (Continued)

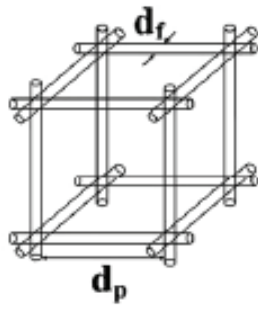
Category 2: Metal foam tube heat exchangers			
Analytical model for friction factor, pressure drop and heat transfer coefficient (overall Nusselt number) were presented. These correlations are developed for the foam filled inner tube of heat exchanger.	(27-29) (30-31)	$f = \frac{4\mu}{K\rho u} PR = \frac{8P}{DaxRe} \quad \frac{\Delta P}{L} = \frac{\mu}{K} uP \quad P = \frac{J_0(\sqrt{\frac{\varepsilon}{Da}})}{2\sqrt{\frac{\varepsilon}{Da}} J_1(\sqrt{\frac{\varepsilon}{Da}}) - J_0(\sqrt{\frac{\varepsilon}{Da}})}$ $Nu_i = \frac{h_i}{k_i} 2R = \frac{2Rq_w}{k_f(T_w - T_{f,bi})} = -2 \frac{k_{se}}{k_f \theta_{f,bi}} \quad Da = \frac{K}{R^2}$ Where J_0 and J_1 are Bessel functions. These correlations are developed for the foam filled inner tube of heat exchanger. Look at the figure 2.1 for R.	Lu et al. [72]
Analytical model for pressure drop and heat transfer coefficient (overall Nusselt number) were presented. These correlations are developed for foam filled outer tube of the heat exchanger.	(32) (33) (34)	$P = \frac{\frac{\Delta P}{L} = \frac{\mu}{K} uP}{(R_2/R_1)^2 - 1}$ $P = \frac{2\sqrt{\frac{Da}{\varepsilon}} \left(N_1 \left(\frac{R_2}{R_1} \left(\sqrt{\frac{\varepsilon}{Da}} \frac{R_2}{R_1} \right) - I_1 \left(\sqrt{\frac{\varepsilon}{Da}} \right) \right) + N_2 \left(\frac{R_2}{R_1} \gamma_1 \left(\sqrt{\frac{\varepsilon}{Da}} \frac{R_2}{R_1} \right) - \gamma_1 \left(\sqrt{\frac{\varepsilon}{Da}} \right) \right) \right)}{\gamma_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right) - \gamma_0 \left(\sqrt{\frac{\varepsilon}{Da}} \frac{R_2}{R_1} \right)}$ $N_1 = \frac{J_0 \left(\sqrt{\frac{\varepsilon}{Da}} \frac{R_2}{R_1} \right) \gamma_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right) - \gamma_0 \left(\sqrt{\frac{\varepsilon}{Da}} \frac{R_2}{R_1} \right) J_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right)}{J_0 \left(\sqrt{\frac{\varepsilon}{Da}} \frac{R_2}{R_1} \right) \gamma_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right) - \gamma_0 \left(\sqrt{\frac{\varepsilon}{Da}} \frac{R_2}{R_1} \right) J_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right)}$	Zhao et al. [73]

Table 2.2 (Continued)

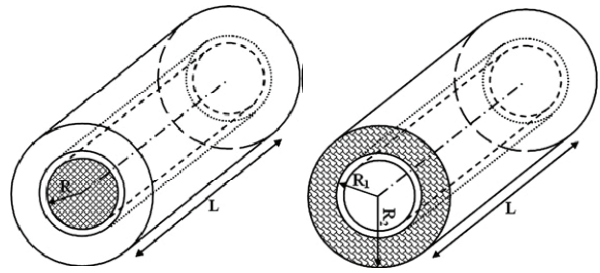
	(35)	$N_2 = \frac{I_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right) - I_0 \left(\sqrt{\frac{\varepsilon R_2}{Da R_1}} \right)}{\gamma_0 \left(\sqrt{\frac{\varepsilon R_2}{Da R_1}} \right) I_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right) - I_0 \left(\sqrt{\frac{\varepsilon R_2}{Da R_1}} \right) \gamma_0 \left(\sqrt{\frac{\varepsilon}{Da}} \right)}$ $Nu_o = \frac{2h_o}{k_f} (R_2 - R_1) = \frac{2q_w (R_2 - R_1)}{k_f (T_w - T_{f,bo})} = - \frac{k_{se}}{k_f \theta_{f,bo}} \frac{R_1}{R_2 - R_1} \quad Da = \frac{K}{R_1^2}$ Where J_0, J_1, γ_0 and γ_1 are Bessel functions. These correlations are developed for foam filled outer tube of the heat exchanger. Look at the figure 2.1 for R_2 and R_1.	
Category 3: Metal foam channel heat exchangers			
Correlations for friction factor and overall heat transfer coefficient for metal foam channel heat exchanger.	(38-40) (41)	$\frac{(\Delta P/L)H}{\rho u^2} = f = \frac{1}{Re_H Da} + \frac{0.105}{Da^{1/2}} \quad Re_H = \frac{uH}{\nu} \quad Da = \frac{K}{H^2}$ $For \ 270 < Re_H < 2050 \quad j = \frac{U}{\rho c_p u} Pr^{2/3} = 13.73 (Re_H^{-0.489} Da^{0.451})$ Look at the figure 2.1 for H and L.	Kim et al. [74]
Empirical correlation for Nusselt number.	(42)	$For \ 1000 < Re_H < 3000 \quad Nu = 0.0159 Re_H^{0.426} Pr^{1/3} Da^{-0.787}$	Kim et al. [75]
Analytical formulation of Nusselt number.	(43)	$Nu = 12 \frac{1 + \kappa}{\kappa} \frac{1}{1 + \frac{3}{Bi(1 + \kappa)} \left\{ 1 - \frac{1}{\sqrt{\frac{Bi(1 + \kappa)}{\kappa}}} \tanh \left(\sqrt{\frac{Bi(1 + \kappa)}{\kappa}} \right) \right\}}$	Lee and Vafai [76]

Table 2.2 (Continued)

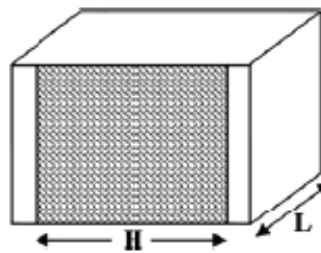
	(44-45)	$\kappa = \frac{k_{fe}}{k_{se}} \quad Bi = \frac{h_{sf} a_{sf} H^2}{4k_{se}}$ Look at the figure 2.1 for H.	
Analytical relationship for Nusselt number.	(46)	$Nu = \frac{h_w D_h}{k_{fe}} = \frac{-4}{\kappa \theta_{f,b}}$	Marafie and Vafai [77]
	(47)	$= \frac{-4}{\kappa \int_0^1 (2 \frac{\sum_{i=1}^4 \Gamma_i(Y)}{\alpha_1} - \frac{\mu}{2\alpha_1} Y^2 + \frac{C_3}{\alpha_1^2} \sinh(\alpha_1 Y) + \frac{C_4}{\alpha_1^2} \cosh(\alpha_1 Y) + C_5) dY}$ $Y = \frac{2y}{H}$ The variables in the above equations ($\Gamma_i, \alpha_1, C_3, C_4, C_5$) are specified in the reference. Look at the figure 2.1 for H.	



a. Cubic unit cell model



b. Metal foam tube heat exchangers



c. Metal foam channel heat exchangers

Figure 2.1 Related figures of some dimensional parameters of correlations given in Table 2.1 [81].

CHAPTER 3

A CLOSER VIEW ON METAL FOAM TECHNOLOGY

Another difference of metal foam heat sinks is that they can be obtained as a bulk material while channels of micro channel heat sinks were cut by machining. Production of microchannel heat sink will completely be investigated in later chapters. Since they are commercial products some additional information should be given about metal foams. First aim of this chapter is to investigate the production methods of metal foams. Usage of metal foams is not restricted to thermal applications. In fact, employing them as heat exchangers is comparably new developing side. Another purpose of this chapter is to see the capabilities and other possible applications of metal foams.

3.1. Metal foams introduction and general properties

Some unique properties like low densities, novel mechanical, thermal, electrical, acoustic and physical assets of metal foams, made them an important class of materials. Especially for energy absorption and thermal management they can offer relatively cheap solutions to the designers.

Table 3.1 Potential applications of metal foams [82]

Application	Comment
Light weight structures	Excellent stiffness-to-weight ratio when loaded in bending
Sandwich cores	Metal foams have low density with good shear and fracture strength

Table 3.1 (Continued)

Strain isolation	Metal foams can take up strain mismatch by crushing at controlled pressure
Mechanical damping	The damping capacity of metal foams is larger than that of solid metals by up to a factor of 10
Vibration control	Foamed panels have higher natural flexural vibration frequencies than solid sheet of the same mass per unit area
Acoustic absorption	Reticulated metal foams have sound-absorbing capacity
Energy management: compact or light energy absorbers	Metal foams have exceptional ability to absorb energy at almost constant pressure
Packaging with high-temperature capability	Ability to absorb impact at constant load, coupled with thermal stability above room temperature
Artificial wood (furniture, wall panels)	Metal foams have some wood-like characteristics: light, stiff, and ability to be joined with wood screws
Thermal management: heat exchangers/ refrigerators	Open-cell foams have large accessible surface area and high cell-wall conduction giving exceptional heat transfer ability
Thermal management: flame arresters	High thermal conductivity of cell edges together with high surface area quenches combustion
Thermal management: heat shields	Metal foams are non-flammable; oxidation of cell faces of closed-cell aluminum foams appears to impart exceptional resistance to direct flame
Consumable cores for castings	Metal foams, injection-molded to complex shapes, are used as consumable cores for aluminum castings

Table 3.1 (Continued)

Biocompatible inserts	Open-cell foams with controlled pore size have potential for high-temperature gas and fluid filtration
Filters	Good electrical conduction, mechanical strength and low density make metal foams attractive for screening
Electrical screening	High surface/volume ratio allows compact electrodes with high reaction surface area
Electrodes, and catalyst carriers Buoyancy	Low density and good corrosion resistance suggests possible floatation applications

Some advantages of using metal foams as heat exchangers can be listed as below:

- Elimination of many thermal interfaces between the main heat source of heat dissipation and heat sink is possible. They can be directly brazed or soldered to a substrate material, which may be semiconductor, ceramic or metal-matrix composite. [83]
- Mixing effect creates turbulence so film coefficient “ h ” becomes higher.
- Metal foams have large specific surface area varying between $500 \text{ m}^2/\text{m}^3$ to over $10000 \text{ m}^2/\text{m}^3$ when compressed. Thus, effective convection surface area is much larger than other heat sinks.
- Solid matrix can be manufactured from a high conductivity metal like aluminum and copper.
- Relative to other cooling techniques they are lighter.
- Since obtained as bulk materials, production cost from metal foam has comparably lower price.
- The performance is scalable meaning that, for higher heat fluxes with limited area metal foams can be compressed to be employed as heat exchangers.

Because of these advantages, metal foams can be used as high performance heat exchangers.

3.2 Categorization of Metal Foams

Foams are categorized according to two characteristics based on their liquid fraction and cell geometry [60]. Foams with extremely low liquid fractions (less than 1 %) are called dry foams. They have such a small fraction of liquid that every cell can be considered as a single surface. Soap bubble can be a good representative sample of this kind. On the contrary of dry foams, wet foams have a large amount of liquid fraction that can be as high as 50% or more. These foams are characterized by their three-dimensional border geometry [60]. They are divided into two sub groups as open and closed cell foams.

Open cell foams are foams with cells that have open windows with neighbor cells. A fluid may freely transport from one cell to the next cell. This type of foam generally produced after a post-operation of removing inter-cellular membranes of closed cell foam. An open cell metal foam figure is given below.



Figure 3.1 Open cell metal foam samples



Figure 3.2 Closer view of open cell metal foam

If neighboring cells are not opened and closed by an inter-cellular membrane that creates individual compartments, foam is named as closed cell. A view of such foam was given in below figure.

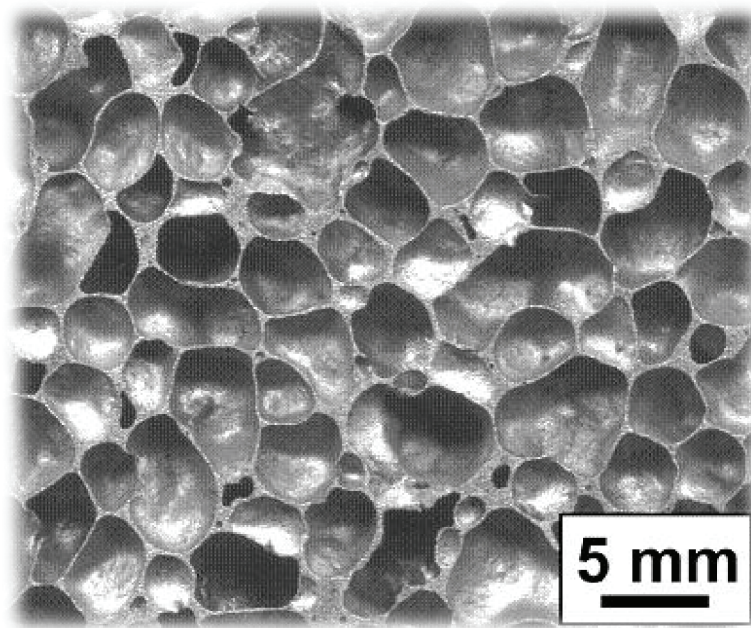


Figure 3.3 Closed cell metal foam [84]

3.3 Cutting, finishing and joining of metal foams

Since metal foams have cellular structure cutting and joining them requires some special techniques:

- Using conventional machine tools is practical for rough cutting since surface damage occurs.
- If high-quality surfaces are required electro-discharge machining, chemical milling, water-jet cutting or high-speed machining should be preferred.
- For bonding adhesives can be used.
- Welding, brazing and soldering are advanced bonding methods of metal foams.
- Threaded, embedded and bolted fasteners require careful design.

Some details are given briefly in following sub-chapters as described by Ashby et al. (2000).

3.3.1 Cutting metal foams

Traditional methods of cutting and machining (bandsawing, milling, drilling) cause severe surface distortion or damage to low-density metal foams. Electro-discharge machining, chemical milling, water-jet cutting and very high-speed flycutters are most appropriate accurate cutting methods of metal foams. Especially if the sample to be used is small it is very crucial to use electro-discharge machining or chemical milling since surface damages caused by other cutting methods influences the properties of the metal foams.

3.3.2 Finishing metal foams

Surfaces of metal foam that have been cut have open cells, a rough texture, and is vulnerable to damage. This opened cells after cutting operation can be filled with epoxy or other resin, or clad (creating a sandwich structure) with a skin of a material compatible with that of the foam. Syntactic structures have a natural skin, which can be polished, etched, anodized or coated by conventional methods.

3.3.3 Joining of metal foams

Metal foams can be soldered, brazed or welded. Foams have a cellular structure resembling, in some ways, that of wood. Because of this they can be joined in way developed for wood, using wood screws, glue joints or embedded fasteners. Figure below summarizes some joining methods.

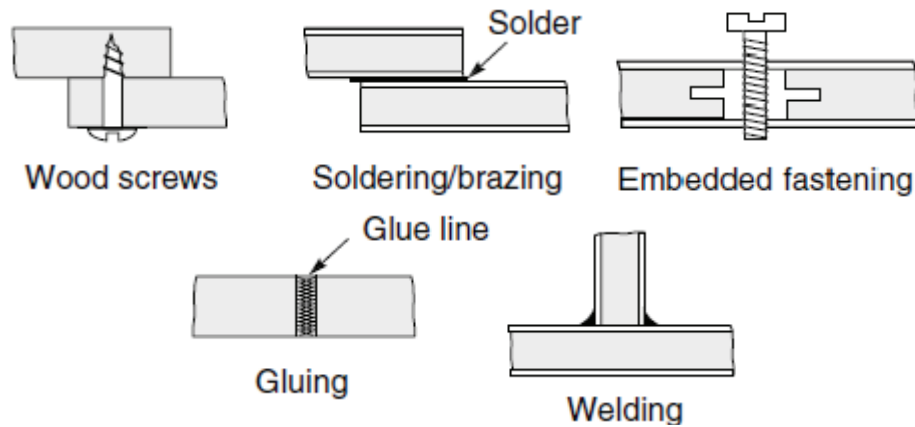


Figure 3.4 Joining methods of metal foams [82]

Welding and brazing are best used for foams with integral skins. Studies of laser welding show promise, but technique requires careful control. Brazing of Al-based foams with aluminum-eutectic alloys is practical. The soldering of aluminum foams requires a flux to remove the oxide film. If the flux penetrates the foam it causes corrosion, so soldering is only practical for sandwiches or skinned structures, restricting the solder to move outer surface of skin. Soldered joints weaken the foam, which fails at a stress less than the tensile strength of the foam itself. If metal foam is produced from an alloy suitable to brazing, this method is the most appropriate way of joining metal foams to a base plate. In brazing, since again an aluminum based filler sheet is used thermal resistances are minimized. Thus, for thermal management purposes, it is the most appropriate method of bonding.

Foams can be glued with the same adhesives used to bond the base metal. The glue joints are usually stronger than the foam itself. There are some drawbacks: low thermal stability, mismatch of expansion coefficient and the possible creation of a thermal and electrical isolation barrier. Provided these are not critical to the design, adhesives (particularly epoxies) allow simple, effective attachment. Their typical use is the attachment of face-sheets to metal foam cores in sandwich-panel construction.

Bonding with fasteners is available; however a careful design should be carried out especially if the bonding part is under the static or kinetic loads. Detailed design guidelines for bonding metal foams with fasteners are given in Ashby et al. [82].

CHAPTER 4

SINGLE-PHASE LIQUID FLOW AND HEAT TRANSFER IN MICROCHANNEL AND METAL FOAM HEAT SINKS

In this study, both microchannel and metal foam heat sinks were experimentally investigated to compare their fluid flow and heat transfer performances. So, detailed analytical models developed to predict fluid flow and heat transfer in microchannel and metal foam heat sinks are beyond the concept of this study. In this chapter, basic relations that could be used to examine experimental results of pressure drop and heat transfer for fluid flow in related heat sinks were presented. Subject was divided into four parts as given below:

- Single-phase pressure drop in microchannels
- Forced convection in microchannels
- Single-phase pressure drop analysis of metal foams
- Forced convection analysis of metal foam heat sinks

4.1 Single-phase pressure drop in microchannels

Pressure drop relations of single-phase liquid flow in microchannels presented in a logical order by first beginning with the pressure drop originated from frictional forces and then including the effects of developing laminar flow and other minor losses as presented by Kandlikar et al. [85]. Since especially latest studies have shown that pressure drop relations developed for classical channels are also applicable for microchannels, relations presented here are not different than

relations for conventional ducts. Applicability of those relations to microchannels will be discussed in this chapter later on.

Relations in this section were derived for pressure drop analysis in smooth circular pipes based on continuum assumption for Newtonian liquid flows. These relations are also applicable for rectangular ducts by using hydraulic diameter instead of the diameter of a circular pipe. Hydraulic diameter for non-circular pipes can be given by the following equation:

$$d_{hyd} = \frac{4A_{cs}}{P_w} \quad (4.11)$$

where A_{cs} is the flow cross-sectional area and

P_w is the wetted perimeter

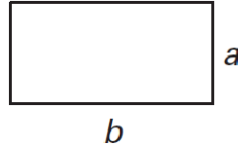


Figure 4.1 Rectangular duct figure

For a rectangular channel of sides a and b shown above Figure 4.3, the hydraulic diameter can be rewritten as:

$$d_{hyd} = \frac{4ab}{2(a+b)} = \frac{2ab}{(a+b)} \quad (4.12)$$

The frictional force caused by shear stress at the wall results in pressure drop.

$$\left(\frac{\pi}{4} d_{hyd}^2\right) dp = (\pi d_{hyd} dx) \tau_w \quad (4.13)$$

where dp is the infinitesimal pressure difference,

dx is the infinitesimal length of a fluid element and

τ_w is the shear stress at wall

Therefore, the pressure gradient and wall shear stress can be related as.

$$\frac{dp}{dx} = \frac{4\tau_w}{d_{hyd}} \quad (4.14)$$

The wall shear stress can be stated in terms of the velocity gradient for Newtonian fluids.

$$\tau_w = \mu \left. \frac{du}{dy} \right|_w \quad (4.15)$$

where μ is the dynamic viscosity of the fluid. At this point, it is useful to introduce Fanning friction factor since it is widely used in heat transfer due to its ability to represent the momentum transfer process of fluid flow in a manner consistent with the heat and mass transfer process representations.

$$f = \frac{\tau_w}{(\frac{1}{2})\rho u_m^2} \quad (4.16)$$

where u_m is the mean flow velocity of fluid in the channel and

ρ is the density of fluid

Combining Equations (4.14) and (4.16) for a channel length of L gives the frictional pressure drop as.

$$\Delta p = \frac{2f\rho u_m^2 L}{d_{hyd}} \quad (4.17)$$

The Fanning friction factor depends on some conditions like if the flow is laminar or turbulent, geometry of channel, if the flow is fully developed or is a developing flow, if the channel walls are smooth or rough.

Reynolds number for fluid flow in channel can be defined as following:

$$Re = \rho u_m \frac{d_{hyd}}{\mu} \quad (4.18)$$

Product of Reynolds number and friction factor is named as Poiseuille number and given as:

$$Po = fRe \quad (4.19)$$

For fully developed laminar flow, Poiseuille number is constant and depends on the flow-channel geometry. For a rectangular channel having a short side of a and a long side of b a channel aspect ratio defined as $\alpha = a/b$. Shah and London [86] presented Poiseuille number for the fully developed laminar flow as.

$$Po = fRe = 24(1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2357\alpha^5) \quad (4.20)$$

Above equation is valid only for fully developed laminar flow. However, as fluid enters a channel, velocity profile starts to develop along the direction of flow and finally reaches a fully developed velocity profile. Most of the studies assume uniform velocity condition at the inlet. But, since the channels of interest are relatively small, developing flow effects can be important. So entrance length for hydrodynamic developing region should be compared to overall length of the channel. Entrance length of the hydrodynamic developing region is given by the following equation [85]:

$$L_H = d_{hyd} 0.05Re \quad (4.21)$$

To include entrance effects into the pressure drop, apparent friction factor f_{app} is defined that accounts for the total affect originated both from friction and developing region. This factor represents an average value of the friction factor.

So if effect of developing flow wanted to be included in the pressure drop following equation may be used.

$$\Delta p = \frac{2f_{app}\rho u_m^2 L}{d_{hyd}} \quad (4.22)$$

The difference between the apparent friction factor over a channel length of x and the fully developed friction factor f is expressed in terms of an incremental pressure defect $K(x)$:

$$K(x) = (f_{app} - f) \frac{4x}{d_{hyd}} \quad (4.23)$$

If the channel length is larger than the hydrodynamic entrance length, $L > L_h$, the incremental pressure defect defined above becomes a constant value known as Hagenbach's factor $K(\infty)$. For rectangular channels Hagenbach's factor is given by Steinke and Kandlikar [87] as follows:

$$\begin{aligned} K(\infty) = & 0.6796 + 1.2197\alpha + 3.3089\alpha^2 - 9.5921\alpha^3 \\ & + 8.9089\alpha^4 - 2.9959\alpha^5 \end{aligned} \quad (4.24)$$

For developing and fully developed flow, Phillips [17], presented an accurate equation of Fanning friction factor for a circular tube [85] as.

$$f_{app} = A Re^B \quad (4.25)$$

where

$$A = 0.09290 + 1.01612/(L/d_{hyd}) \quad (4.26)$$

$$B = -0.26800 - 0.32930/(L/d_{hyd}) \quad (4.27)$$

For rectangular channels, Reynolds number in Equation (4.25) should be replaced with the laminar-equivalent Reynolds number given by Jones [88].

$$Re_{le} = \rho u_m \frac{D_{le}}{\mu} = \rho u_m \frac{[(2/3)+(11/24)(1/\alpha)(2-1/\alpha)]d_{hyd}}{\mu} \quad (4.28)$$

In above equation, D_{le} is laminar-equivalent diameter.

Equations given until now were to find the pressure loss due to wall friction effecting the major losses. Since pressure measurements in this study were taken before channel inlet and after the channel exit, some minor losses should be included to total pressure loss. These minor losses that are to be added are losses in bends at inlet and exit and sudden contraction and expansion losses. So, the total pressure loss equation given by Phillips [17] is:

$$\Delta p = \frac{2f_{app}\rho u_m^2 L}{d_{hyd}} + K_c \frac{\rho u_m^2}{2} + K_e \frac{\rho u_m^2}{2} + \left(\frac{A_c}{A_p}\right)^2 (2K_{90}) \frac{\rho u_m^2}{2} \quad (4.29)$$

where A_c is the total channel area,

A_p is the total plenum cross-sectional area,

K_c is the sudden contraction loss coefficient,

K_e is the sudden expansion loss coefficient,

K_{90} is the 90° bend loss coefficient.

Thus, above equation includes all pressure losses caused by frictional effects and additional losses caused due to developing region of flow, sudden expansion and sudden contraction losses and bending losses at the inlet and exit. For bending loss coefficient Phillips [17] recommended a value of approximately 1.2. For sudden contraction and expansion losses coefficients in rectangular channels Figure 4.4 derived by Kays and London [89] can be read.

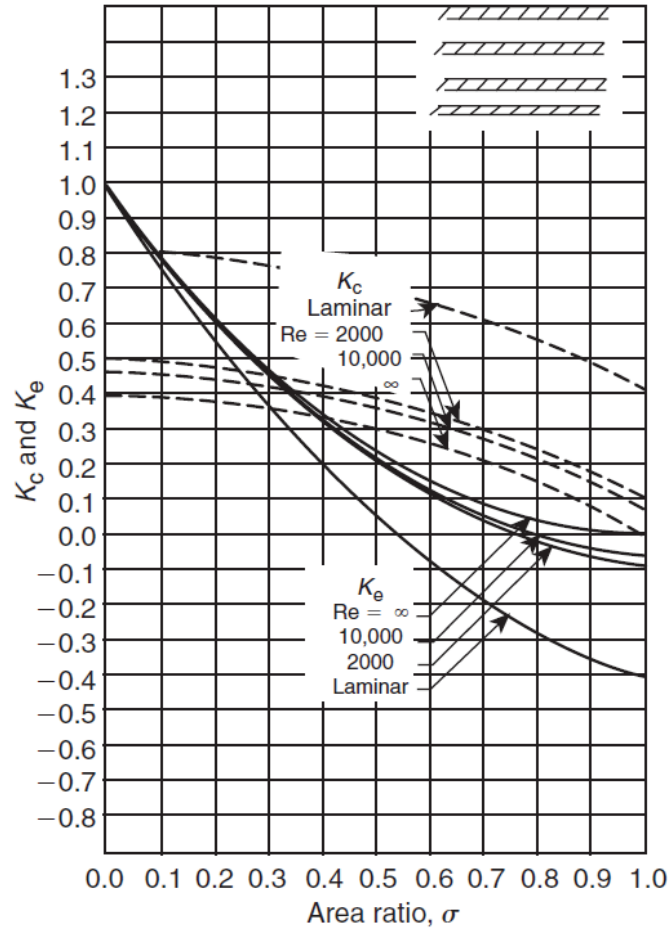


Figure 4.2 Contraction and expansion loss coefficients [89]

4.2 Forced convection in microchannels

Heat transfer is carried out by convection so Newton's law of cooling is also applicable. Heat flux from solid surfaces can be given as below:

$$q'' = \frac{q}{A_t} = h(T_s - T_\infty) \quad (4.30)$$

where q is the heat transfer rate,

A_t is the total convective heat transfer area of heat sink,

h is the convection coefficient,

T_s is the surface temperature of convection and

T_∞ is the bulk fluid temperature.

Due to the energy balance inside the system, heat transfer rate can also be found by considering the energy delivered to the fluid in the microchannel.

$$q = \dot{m}c_f(T_{c,out} - T_{c,in}) \quad (4.31)$$

where c_f is the specific heat capacity of fluid,

$\dot{m}$ is the mass flow rate of coolant,

$T_{c,out}$ is the temperature of coolant at the exit and

$T_{c,in}$ is the temperature of coolant at the inlet.

Ratio of convective and conductive heat transfer across the boundary is called as Nusselt number.

$$Nu = \frac{hd_{hyd}}{k_f} \quad (4.32)$$

where k_f is thermal conductivity of coolant. According to classical theory, the Nusselt number in fully developed laminar flow is expected to be constant and depends on the channel geometry and wall heat transfer boundary condition.

Nusselt number depends on the channel aspect ratio defined as before $\alpha = a/b$, and the wall boundary conditions. Nusselt numbers of rectangular channels for three boundary conditions are presented below [85]:

Constant wall temperature condition:

$$Nu_T = 7.541(1 - 2.610\alpha + 4.970\alpha^2 - 5.119\alpha^3 + 2.702\alpha^4 - 0.548\alpha^5) \quad (4.33)$$

Constant circumferential wall temperature, uniform heat flux:

$$Nu_{H1} = 8.235(1 - 2.0421\alpha + 3.0853\alpha^2 - 2.4765\alpha^3 + 1.0578\alpha^4 - 0.1861\alpha^5) \quad (4.34)$$

Constant wall heat flux:

$$Nu_{H2} = 8.235(1 - 10.6044\alpha + 61.1755\alpha^2 - 155.1803\alpha^3 + 176.9203\alpha^4 - 72.923\alpha^5) \quad (4.35)$$

As in fluid flow for forced convection, there also exists an entrance region called “thermal entrance region” and it is defined as follows [85]:

$$L_i = 0.1d_{hyd}RePr \quad (4.36)$$

The constant, 0.1, in above equation was suggested by Phillips [17] for rectangular channels and is different for other geometries. Prandtl number, Pr , is defined as the ratio of momentum to thermal diffusion.

$$Pr = \frac{c_f \mu}{k_f} \quad (4.37)$$

For rectangular channels with four-side being heated, the Nusselt number in the developing laminar region was given in Table 4.1 suggested by Phillips [17].

Table 4.1 Nusselt numbers for thermally developing region [17]

x^*	$\alpha < 0.1^*$	$\alpha = 0.25$	$\alpha = 0.333$	$\alpha = 0.5$	$\alpha = 1.0$	$\alpha > 10^{**}$
0.0001	31.4	26.7	27.0	23.7	25.2	31.6
0.0025	11.9	10.4	9.9	9.2	8.9	11.2
0.005	10	8.44	8.02	7.46	7.1	9.0
0.00556	9.8	8.18	7.76	7.23	6.86	8.8
0.00625	9.5	7.92	7.5	6.96	6.6	8.5
0.00714	9.3	7.63	7.22	6.68	6.32	8.2
0.00833	9.1	7.32	6.92	6.37	6.02	7.9
0.01	8.8	7	6.57	6.05	5.69	7.49
0.0125	8.6	6.63	6.21	5.7	5.33	7.2
0.0167	8.5	6.26	5.82	5.28	4.91	6.7
0.025	8.4	5.87	5.39	4.84	4.45	6.2
0.033	8.3	5.77	5.17	4.61	4.18	5.9
0.05	8.25	5.62	5.00	4.38	3.91	5.55
0.1	8.24	5.45	4.85	4.22	3.71	5.4
1	8.23	5.35	4.77	4.11	3.6	5.38

*-parallel plates, both sides heated; **-parallel plates one side heated

In Table 4.1, x^* is defined as:

$$x^* = \frac{L/d_{hyd}}{RePr} \quad (4.38)$$

For heating from three sides, the following conditions are applicable:

If $\alpha \leq 0.1$ and $\alpha \geq 10$ use four side heating as given in Table 4.1 without any modification.

If $0.1 \leq \alpha \leq 10$, following equation can be used to find Nusselt number for the three-side heating condition.

$$Nu_3 = Nu_4 \frac{Nu_{3,fd}}{Nu_{4,fd}} \quad (4.39)$$

where Nu_4 is the four-side heating condition given in table 4.1,

$Nu_{3,fd}$ is the fully developed three-side heating condition given in table 4.2 and

$Nu_{4,fd}$ fully developed four-side heating condition given in table 4.2.

Table 4.2 Nusselt numbers for thermally fully developed flow [17]

$\alpha = a/b$	$Nu_{fd,3}$	$Nu_{fd,4}$
0	8.235	8.235
0.10	6.939	6.700
0.20	6.072	5.704
0.30	5.393	4.969
0.40	4.885	4.457
0.50	4.505	4.111
0.70	3.991	3.740
1.00	3.556	3.599
1.43	3.195	3.740
2.00	3.146	4.111
2.50	3.169	4.457
3.33	3.306	4.969
5.00	3.636	5.704
10.00	4.252	6.700
> 10.00	5.385	8.235

Nusselt number predictions in heat transfer analysis presented in Table 4.1 and 4.2 are reasonable for water and some researchers obtained successful solutions when compared their results with experimental data [85].

To find Nusselt number experimentally, after obtaining successful data, the following Nusselt number definition [60] can be used.

$$Nu = \frac{hd_{hyd}}{k_f} = \frac{qd_{hyd}}{k_f A_{con} \Delta T} \quad (4.40)$$

where A_{con} is the convection surface area and

ΔT is the temperature difference that results heat to flow.

$$\Delta T = T_{pl} - T_{c,in} \quad (4.41)$$

where T_{pl} is average surface temperature of heat spreader plate. Substituting Equations (4.31) and (4.41) into Equation (4.40) Nusselt number can be calculated using steady state experimental data.

$$Nu = \frac{qd_{hyd}}{k_f A_{con}(T_{pl} - T_{c,in})} = \frac{\dot{m} c d_{hyd} (T_{c,out} - T_{c,in})}{k_f A_{con}(T_{pl} - T_{c,in})} \quad (4.42)$$

4.3 A discussion on applicability of classical theory on microchannels

Until today, many researches have been carried out on the topic of microchannels. Initial studies have reported discrepancies from classical fluid flow and heat transfer theories. Some of them observed early transition to turbulent flow and proposed new critical Reynolds number. Some of them has examined higher pressure drop than expected and offered new correlations according to experimental data that they gathered. Other surveyed huge differences from conventional heat transfer equations and proposed new relations. Recent trend of studies on microchannels have shown that classical theory developed for rectangular ducts might be applicable also for microchannels. At this point a classification of channels may be useful.

Generally channel size classification of Kandlikar et al. [85] accepted in the literature. Classification according to the smallest channel dimension offers the following:

$D_s < 0.1\mu m$	Nanochannels
$0.1\mu m < D_s < 1\mu m$	Transitional nanochannels
$1\mu m < D_s < 10\mu m$	Transitional microchannels
$10\mu m < D_s < 200\mu m$	Microchannels
$200\mu m < D_s < 3000\mu m$	Minichannels
$3000\mu m < D_s$	Conventional channels

Also, Obot [90] presented a channel classification based on hydraulic diameter and proposed that channels with hydraulic diameters less than 1 mm ($D_h < 1$ mm) are microchannels. Whichever classification was chosen, the main aim should be to investigate whether there is a disagreement with classical correlations. Researchers also have given dimensions of channels that they have studied with and when literature on microchannels was observed it can easily be seen that also for channels with similar dimensions there is no full agreement. As mentioned above, latest tendency tells us that there should be agreement with conventional correlations if experiments carried out carefully. Harms et al. [29] stated a good agreement with classical theory. Some discrepancies were attributed to inlet manifold conditions and channel roughness. Qu and Mudawar [31] reported successful experimental results that were compatible with analytical and numerical results. Also Warriar et al. [32], Liu and Garimella [34], Lee et al. [35], Akbari et al. [36], Kandlikar et al. [85] all concluded that conventional theory was enough to predict the fluid flow and heat transfer in microchannels. The reason of diversities reported by previous studies was carefully investigated by Kandlikar [85] and following conclusions were made.

- Some researchers assumed fully developed flow. However since the pressure gradients found in small diameter channels are quite high, the flow lengths are generally kept low. In many applications, the length of the channel in the developing region forms a major portion of the flow length through the channel. To include additional frictional effects caused by developing region, apparent friction factor should be used.
- Some studies did not include minor losses caused by sudden expansion and contraction and bending losses. Especially if pressure drop measurement is taken before the inlet and after the exit, overall pressure drop should include these kinds of pressure losses.
- Some test specimens were assumed to be rectangular, however they were not. Especially channel geometries produced by silicon etching can be different from exact rectangular shape as shown in Figure 4.3 and this

effects all geometry dependent solutions. So, after production, geometry should be well defined and the dimensions of the channel should be measured accurately.

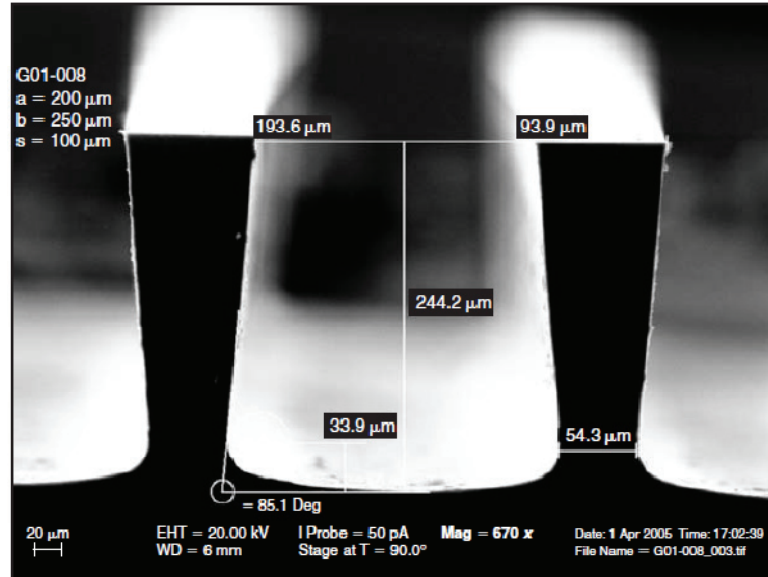


Figure 4.3 Cross-sectional view of a rectangular microchannel etched in silicon showing distortion in channel geometry from ideal rectangular profile [85]

- As channel dimensions become smaller roughness can play an important role. Average roughness values can be measured and included in the calculations as Kandlikar et al. [85] suggested in his study.
- Uncertainties can be a major problem. Analysis of uncertainties should be carefully carried out.

Same discussions can be made on forced convection issue. Again agreement with the classical flow theory is expected to hold, but reported data show significant scatter due to difficulties encountered in making local heat flux and temperature measurements. Main reasons can be:

- Entrance region effects: Generally fully developed flow assumption was also made for heat transfer analysis. Because microchannels are relatively short, neglecting entrance effects may result in high discrepancies. Especially for increasing Reynolds number, entrance length also increases and deviation becomes higher.
- Uncertainties in experimental measurements: Uncertainty analysis of Nusselt number can be made as suggested again by Kandlikar et al. [85].
- Right determination of the thermal boundary condition: Actual boundary conditions of microchannels are sometimes difficult to ascertain especially if they fall in between constant heat flux or temperature boundary conditions.

As argued above, after a careful investigation with careful experimental study, obtained analytical results should be parallel to the collected data. Also in this study, after a careful investigation on literature, experimental set-up was built carefully to limit uncertainties and analytical predictions were made considering possible effects with minimum possible simplifying assumptions.

4.4 Single-phase pressure drop analysis of metal foam heat sinks

An extensive study on metal foams was presented by Boomsma [16] and relations that are applicable for fluid flow and heat transfer analysis for metal foams are referred to this study. For flow of clear fluids through porous media, instead of traditional momentum equations or force balances in normal fluid dynamics Darcy's law is valid. Henry Darcy (1856) conducted experiments to investigate fluid flow with a packed of earth and sand and using experimental data derived new relations. In the equation developed through experimentation, volumetric flow rate of water through a column of packed earth related to the product of the height of the water in the column and a new factor named hydraulic conductivity was described.

$$Q = \frac{k_h A_{cs}}{h} (p_t + h_p - p_b) \quad (4.43)$$

where Q is the volumetric flow rate,

k_h is the hydraulic conductivity,

A_{cs} is the column cross-sectional area,

h_p is the height of the porous layer,

p_t is the pressure at the top of the sand layer (units of length) and

p_b is the pressure at the bottom of the sand layer (units of length).

In later years, Krüger (1918) was first to relate viscosity into the Darcy law. A more convenient form of this equation relates the area-averaged velocity to the pressure drop in x-direction through porous media, and it can be presented as follows:

$$U_a = -\frac{K}{\mu} \left(\frac{dp}{dx} \right) \quad (4.44)$$

where U_a is area-averaged velocity,

K is the permeability and

p is the pressure.

At this point, it is useful to make a dimensional analysis of permeability of a porous medium.

$$[K] = \frac{[\mu][U]}{\left[\frac{dp}{dx} \right]} = (length)^2 \quad (4.45)$$

Pressure drop per unit length becomes:

$$\frac{\Delta p}{L} = \frac{\mu U}{K} \quad (4.46)$$

For metal foams, it is possible to define two different Reynolds numbers. First one is based on the characteristic length (square root of permeability) and second one is based on average pore diameter. Using characteristic length gives following Reynolds number definition:

$$Re = \frac{\rho U K^{1/2}}{\mu} \quad (4.47)$$

Even though the porous media is not well-defined, this definition works well especially for porosities lower than 0,4. Since metal foam structure relatively has a more well-defined structure another Reynolds number definition based on average pore diameter (d_{ave}) is possible.

$$Re = \frac{\rho U d_{ave}}{\mu} \quad (4.48)$$

Darcy's law presented in Equation (4.44) performs well for lower porosities and lower flow velocities. Higher flow velocities cause a non-linear increase in the pressure drop and by many experiments this increase has been reported also for metal foams by many researchers as investigated in Chapter 2. Quadratic term has been added by Dupuit (1863) through experimentation and analysis of the related data.

$$0 = \frac{dp}{dx} - \frac{\mu}{K} U - \rho C U^2 \quad (4.49)$$

where C is the form coefficient.

Dimensional analysis of form coefficient can be given as follows:

$$[C] = \frac{\frac{[dp]}{[dx]}}{[\rho][U]^2} = (length)^{-1} \quad (4.50)$$

Pressure drop per unit length becomes:

$$\frac{\Delta p}{L} = \frac{\mu}{K} U + \rho C U^2 \quad (4.51)$$

where C is the form coefficient of porous media. After gathering experimental data, several methods are available to calculate the permeability and the form coefficient. First one is modifying Equation (4.51) to bring it into linear form and to extrapolate in order to determine the coefficients. This method has been shown lack of accuracy by Antohe et. al. [53] as discussed in Chapter 2. A more accurate way of finding those constants is using the least-squares curve fit through the pressure drop versus fluid speed data points. Using this method also makes an accurate uncertainty analysis possible. Substituting into Equation (4.51),

$$A = \frac{\mu}{K} \text{ and } B = \rho C \quad (4.52)$$

yields following simpler quadratic equation:

$$\frac{\Delta p}{L} = AU + BU^2 \quad (4.53)$$

Applying the least squares quadratic fit to Equation (4.53) gives the following results for A and B.

$$A = \frac{(\sum_{i=1}^n v_i p_i)(\sum_{i=1}^n v_i^4) - (\sum_{i=1}^n v_i^2 p_i)(\sum_{i=1}^n v_i^3)}{(\sum_{i=1}^n v_i^2)(\sum_{i=1}^n v_i^4) - (\sum_{i=1}^n v_i^3)(\sum_{i=1}^n v_i^3)} \quad (4.54)$$

$$B = \frac{(\sum_{i=1}^n v_i^2 p_i)(\sum_{i=1}^n v_i^2) - (\sum_{i=1}^n v_i p_i)(\sum_{i=1}^n v_i^3)}{(\sum_{i=1}^n v_i^2)(\sum_{i=1}^n v_i^4) - (\sum_{i=1}^n v_i^3)(\sum_{i=1}^n v_i^3)} \quad (4.55)$$

4.5 Forced convection analysis of metal foam heat sinks

Heat transfer analysis of gathered experimental data is done by using the same Nusselt number formulation given by Equation (4.42). Finding Nusselts number for each tested specimen gives an important measure of heat transfer performances. To compare heat exchanger performances, another characterization called Colburn factor which gives a heat transfer performance estimate comparing the convection coefficient to required flow velocity can be presented as follows:

$$j = \frac{h}{\rho c U} \left(\frac{v}{\alpha_f} \right)^{\frac{2}{3}} \quad (4.56)$$

where v is the kinematic viscosity: $v = \mu \rho^{-1}$,

α_f is the fluid thermal diffusivity: $\alpha_f = k \rho^{-1} c^{-1}$ and

U is the measured velocity to achieve corresponding convection coefficient.

Heat convection performance of the heat exchanger should be weighed against pumping power for comparison purposes. The most ideal exchanger is the one with less thermal resistance and less pumping power can be evaluated by:

$$\dot{W} = \Delta p Q \quad (4.57)$$

where Δp is the measured pressure difference through heat exchanger and

Q is the volumetric flow rate, while the thermal resistance is given by:

$$R_{th} = \frac{\Delta T}{q} = \frac{T_{pl} - T_{c,in}}{\dot{m} c (T_{c,out} - T_{c,in})} \quad (4.58)$$

where T_{pl} is the average surface temperature of heat spreader plate,

$T_{c,inlet}$ is the inlet fluid temperature and

$T_{c,outlet}$ is the fluid temperature at the exit.

CHAPTER 5

EXPERIMENTAL SETUP AND DISCUSSION ON RESULTS

5.1 Motivation for experimental study

Today, microchannels and metal foams can be employed as high performance heat sinks especially in military applications. With developing technology phased-arrayed radar transmitter and receiver (T/R) modules have become smaller and had high heat flux dissipating surfaces that should be cooled down in order to have reliable systems. Deciding in between these two recently developing cooling methods requires careful analysis. Although microchannels can be analyzed analytically and numerically with a good degree, the same cannot be said for metal foam heat sinks. Although some numerical and experimental studies were conducted on metal foams, since producing methods and the type of metal foams can vary greatly and every produced metal foam heat sink type has its own special characteristics, it becomes harder to make a comparison with microchannels without a direct experimental study. Experimentally comparing microchannels and metal foam heat sinks under same conditions becomes an important issue especially if one of these would be used as cooling method in future projects.

5.2 Experimental apparatus

Main aim of the study was to measure the hydraulic and thermal performances of microchannels and open cell aluminum metal foams when used as heat exchangers in a forced convection flow arrangement. The concept included a rectangular channel either filled with microchannels or metal foams and comparing them experimentally. Filled volumes, delivered heat flux, heated area,

volumetric flow rates i.e. all testing conditions held the same both for microchannel and metal foam heat sinks. The characterization of heat sinks included measuring the surface temperature of the heat load carrier plate, the inlet and outlet coolant temperature, the pressure drop across heat exchangers and delivered power to the heater for various coolant flow rates.

A general overview of the setup is given schematically in Figure 5.1.

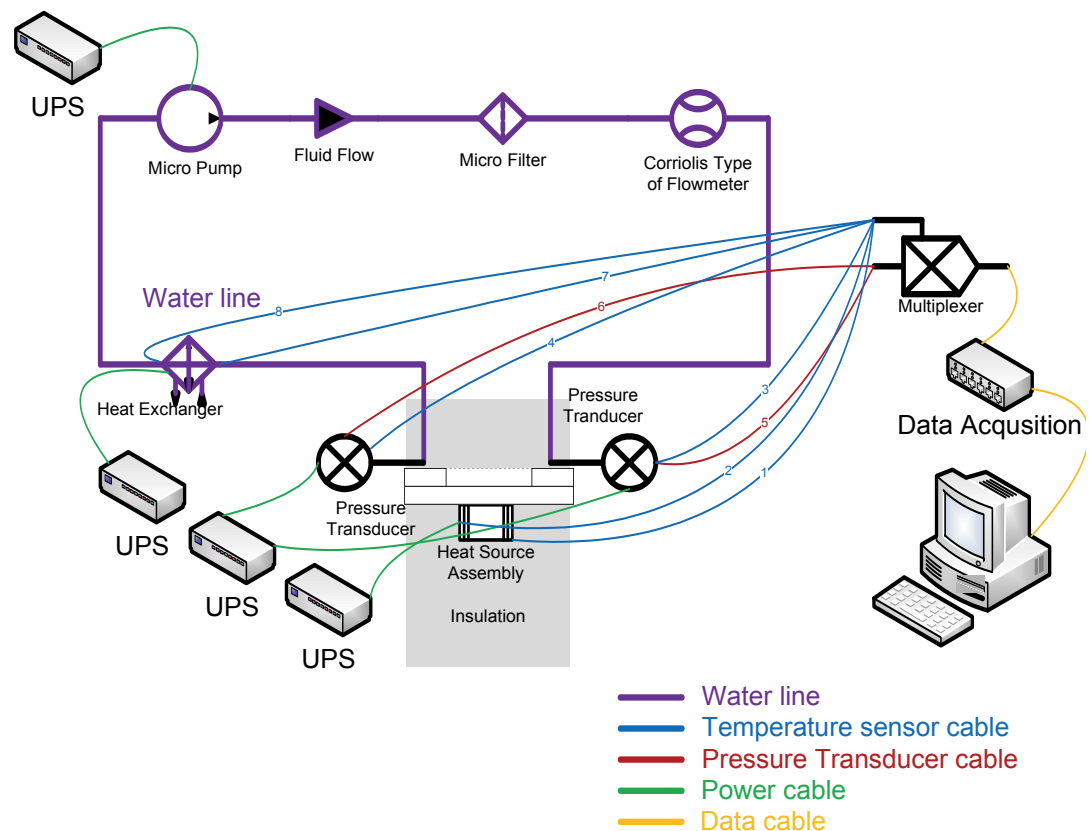


Figure 5.1 Schematic view of the experimental setup used to measure hydraulic and thermal characteristics of microchannel and metal foam heat sinks

Some general properties of experimental instruments used are presented in Table 5.1.

Table 5.1 General properties of experimental instruments

COMPONENT AND NO	BRAND	TYPE	FUNCTION	BRIEF PROPERTIES	VISUAL
01 UPS	AGILENT	E3634A	HEAT SOURCE POWER	0 to 25V/7A or 0 to 50V/4A power supply accuracy: 0.05%+10mV 0.2%+10mA	
02 UPS	AGILENT	E3620A	PRESSURE TRANSDUCER POWER	0 to 25V/1A power supply accuracy: 0.5%V 0.5%A	
03 UPS	AGILENT	E3617A	MICRO PUMP POWER	0 to 60V/1A power supply accuracy: 0.5%V 0.5%A	
04 UPS	AGILENT	E3615A	HEAT EXCHANGER FAN POWER	0 to 20V/3A power supply accuracy: 0.5%V 0.5%A	
05 DATA ACQUISITION	AGILENT	34970A	DATA ACQUISITION	60 channel data logger with multiplexer that collects and records voltage, current, temperature, frequency measuring with minimum sampling interval of 2 seconds	

Table 5.1 Continued

COMPONENT AND NO	BRAND	TYPE	FUNCTION	BRIEF PROPERTIES	VISUAL
06 FLOWMETER	SIEMENS	SITRANS F C	FLOW RATE MEASUREMENT	coriolis mass flow meter that measures flow rate, density, temperature of gases and liquids. Max. Mass flow rate: 1000kg/h flow rate error: $\leq 0,1\%$ of rate density error: $\leq 0,0015\%$ g/cm ³	
07 HEATSOURCE	LAIRD	6529	HEAT SOURCE	thermoelectric module max. operating temperature: 175°C max power can be altered if exchanger cools surface temperature of heater below 175°C	
08 HEAT LOAD CARRIER	NA	NA	CUSTOM COPPER HEAT LOAD CARRIER	custom designed copper heat load carrier to carry heat from heater to microchannel or metal foam heat exchanger.	
09 INSULATION	IZOCAM	TERRAMANTO IZOPOR PLUS	ISOLATING HEAT SOURCE AND HEATEXCHANGER ASSEMBLY	ekspande polistiren 50cmx50cmx5cm thermal conductivity: $\leq 0,035\text{W/mK}$	

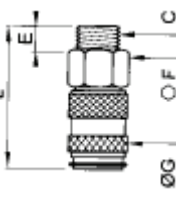
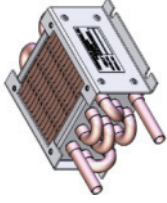
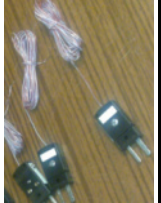
Table 5.1 Continued

COMPONENT AND NO	BRAND	TYPE	FUNCTION	BRIEF PROPERTIES	VISUAL
10 THERMAL PAD	THERMAGON	T-pli 240	THERMAL PAD WITH HIGH THERMAL CONDUCTIVITY	gap filling thermal pad that reduces thermal resistances originating from surface roughness with high conductivity. thickness: 1,02mm±0,05mm thermal conductivity: 6W/mK temperature range: -45°C to 200°C	
11 PRESSURE TRANSDUCER AND ADAPTATION PART	KULITE	HKL/T-1-235	PRESSURE/TEMP. SENSOR	combined pressure and RTD temperature sensor several types are available: 0-1,7Bar; 0-3,5Bar rated electrical excitation: 10VDC combined non-linearity, hysteresis and repeatability ±0,1% FSO (typ.), ±0,5% FSO (max.) full scale output (FSO): 100mV(Nom.) operating temperature range:-55°C to 175°C	
12 AXIAL FAN	COSTECH	D08A04HWP	AIR FLOW THROUGH HEAT EXCHANGER	12VDC 3100rpm axial fan. Air flow:67,3m³/h	

Table 5.1 Continued

COMPONENT AND NO	BRAND	TYPE	FUNCTION	BRIEF PROPERTIES	VISUAL
13 COMPUTER	HP	VECTRA	PENTIUM II PROCESSOR DESKTOP		
14 EXPANSION TANK	AQUARIUS	III	RESERVOIR	40mmx55mmx100mm expansion tank	
15 MICROFILTER	ECE	FLP 2421	FILTERING	10 micron filter	
16 PUMP	MICROPUMP	200 350-000 GB-P25	WATER PUMPING	displacement: 0,58ml/rev maximum flow: 4000ml/min maximum differential pressure: 8,7Bar maximum system pressure: 21 Bar working temperature: -46 to 177°C	

Table 5.1 Continued

COMPONENT AND NO	BRAND	TYPE	FUNCTION	BRIEF PROPERTIES	VISUAL
17 COUPLINGS	LEGRIS CONNECTIC	92012010 and 92852104	QUICK LOW PRESSURE LOSS COUPLINGS	low pressure loss quick couplings for water ethylene-glycol. working temperature : -20 to +100°C max working pressure : 35Bar	
18 HEAT EXCHANGER	THERMATRON ENGINEERING	72MSLS0A01	COOLING HEATED WATER IN LINE	Compact, 3-row liquid-to-air heat exchanger. Copper fins and tubing.	
19 THERMOCOUPLE	OMEGA	TT-J-36- SLE(ROHS)	MEASURING TEMPERATURES AT SEVERAL LOCATIONS	J-type thermocouple measuring range: 0-260°C measured error is: ±1°C checked in thermal bath	
20 PIPE	COLE PARMER	60422-10	PIPING	flexible translucent lab uses tubing working temp: -73 to 135 °C inner diameter : 6,4mm outer diameter: 9.525mm	

Pressure drop across heat sinks was measured by two Kulite (HKL/T-1-235) pressure transducers with an accuracy of 0.1% FS (± 0.0025 bar). These pressure transducers also have the capability of measuring temperature by means of integrated RTD type temperature sensor with an accuracy of 1%. Pressure transducer was screwed to an additional carrier part that was also screwed to the heat sink to be tested. 2 J-type thermocouples were employed to measure the surface temperature of heat load carrier plate. More than ten thermocouples were produced. These thermocouples and RTD temperature sensor was checked in ice-water and boiling water bath, two most reliable thermocouples with accuracy better than $\pm 1^\circ\text{C}$ were selected among others, and RTD temperature sensors were observed to have an accuracy better than $\pm 0.2^\circ\text{C}$ especially in the interested range of temperatures ($25\text{--}35^\circ\text{C}$). Two thermocouples were employed to measure the inlet and exit coolant temperature of heat exchanger just to monitor the behavior of exchanger and to have additional information whether steady-state conditions were reached or not. Another thermocouple was also used to follow heater surface temperature.

Heating system consisted of a heat load carrier and thermoelectric module. As discussed in Chapter 1, a thermoelectric module is a cooler and a heater. Good degree of uniform heat flux is possible with a power of I^2R . A custom design copper heat load carrier was employed to gradually reduce the heated area of thermoelectric module. One side of the carrier part has hot side dimensions of heater and other side has dimensions just lower than filled area of tested heat sinks ($12\text{ mm} \times 25\text{ mm}$). To prevent thermal resistance due to surface roughness, heater was glued with a conductive silver epoxy to heat load carrier plate. Again to avoid surface roughness effects Thermagon (T-pli 240) thermal pad with thickness: $1.02\text{ mm} \pm 0.05\text{ mm}$ and thermal conductivity of 6 W/mK was employed between heat load carrier and tested heat sink. Carrier part was assembled to heat sink by means of four polyoxymethylene (POM) M2 bolts to prevent heat flux to the heat sink with conduction. Detailed heat sink assembly is shown in Figure 5.2. Part numbers in this figure can be followed by using Table 5.1.

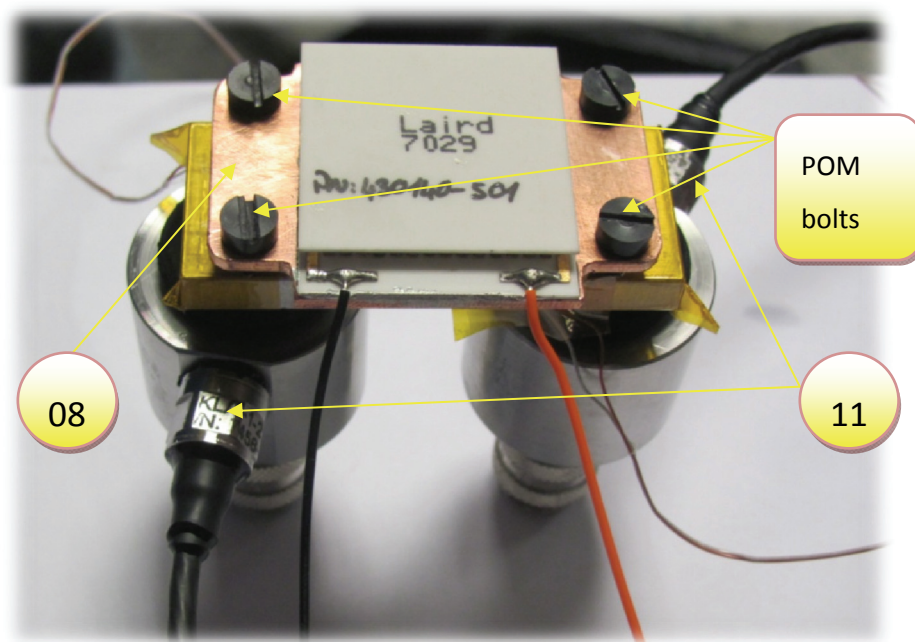
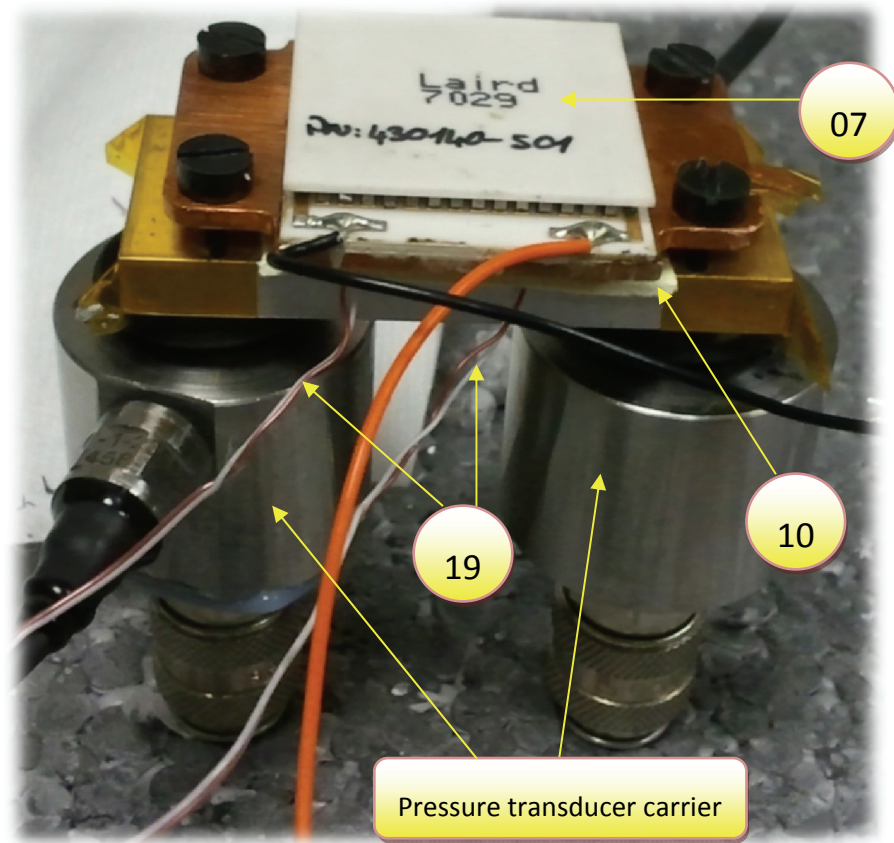


Figure 5.2 Views of heat sink assembly in detail

Surface temperatures of heating assembly that faces heat sink were taken at several locations before bonding it to heat sink and they did not vary more than 2°C. As seen from Figure 5.2, thermocouples were bonded at two different locations. These locations were shown in Figure 5.3 and they were observed to give good measurements. Softness property of thermal pad was exploited and thermocouples were buried in pad so that thermocouples did not create thermal resistances at bonding spots. Produced plastic bolts had two main functions which are to align heater with heat sink and to squeeze thermal pad in between to prevent thermal resistance adequately.

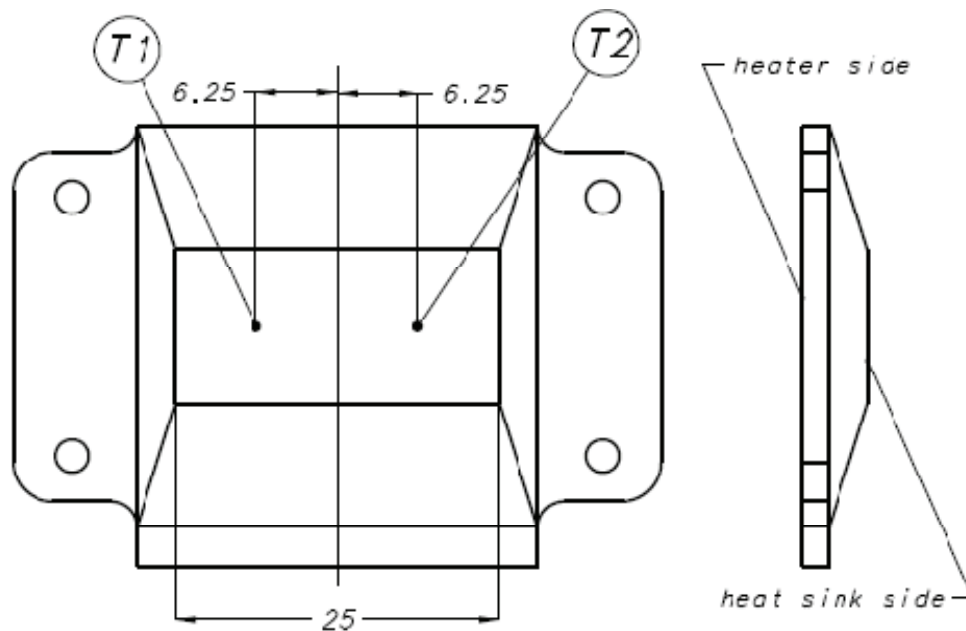


Figure 5.3 Views of heat load carrier and thermocouple locations

Ten 50 cm x 50 cm x 5 cm Izocam isolation panel with thermal conductivity less than 0,035 W/mK was employed to prevent any heat lost to the environment. Heat sink assembly was placed in a groove as shown in Figure 5.4 so that it was in the center of an isolation cube.

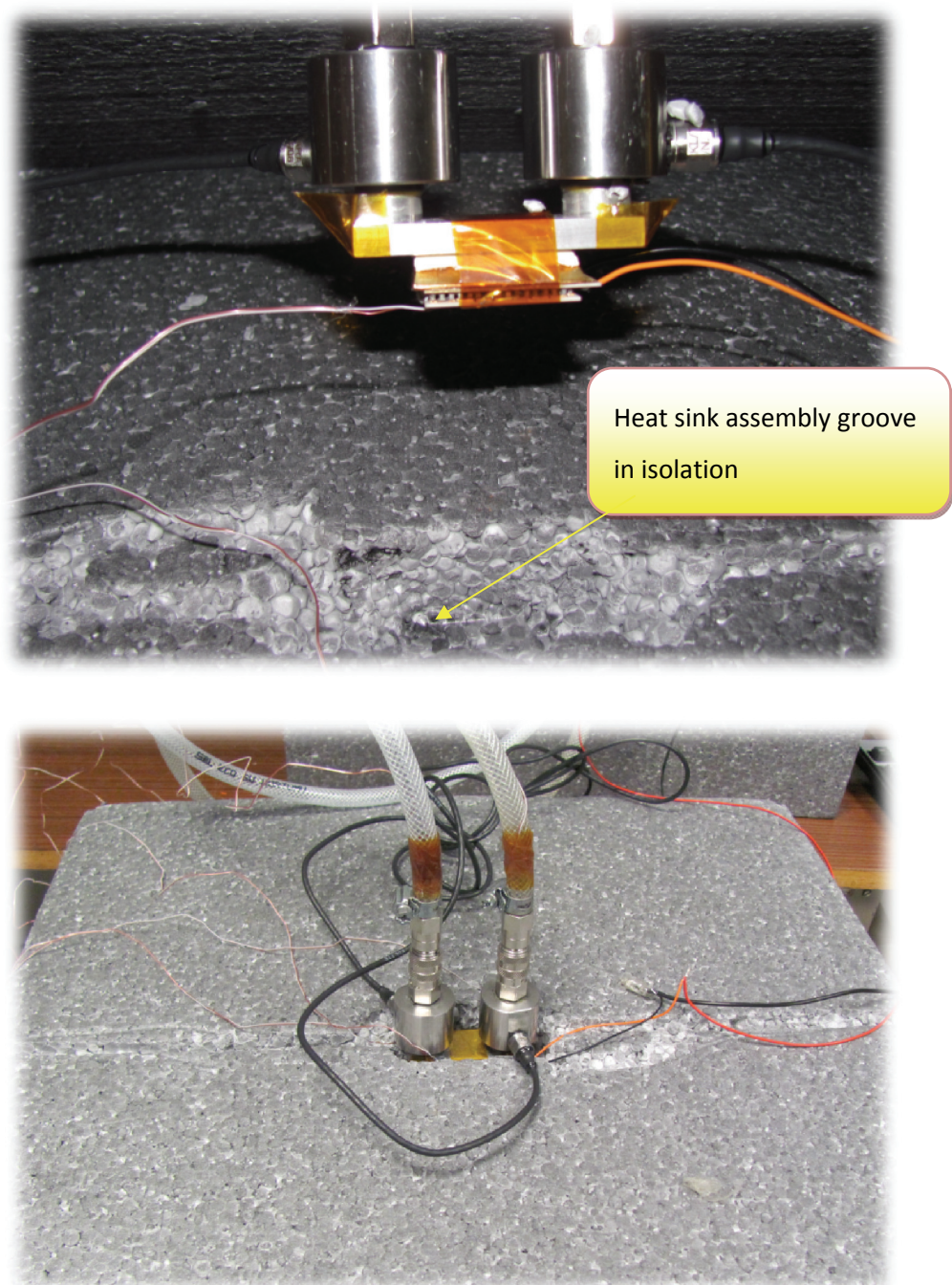


Figure 5.4 Views of heat sink assembly with isolation

Siemens (Sitrans F C) Coriolis type of flow meter was used to measure the flow rate with an error less than 0.1 %. However reading accuracy was lower and this situation was included in the uncertainty analysis. General view of the test setup is

The figure displays the experimental setup for the study. The main image shows a laboratory bench with various equipment labeled 01 through 18. A large stack of grey, rectangular samples is labeled 09. Two smaller inset images at the bottom show close-ups of power supplies labeled 03 and 04.

The acquisition of the signals from sensors which included both pressure transducers and thermocouples was handled by a RS-232 data acquisition device manufactured by Agilent. The device was attached via a RS-232 port on a PC running on Windows XP which controlled the acquisition device using Agilent

Benchmark. With this configuration, temperature and pressure were viewed and recorded to the PC hard drive in real-time.

Distilled water was employed as coolant. Each heat sink was tested by following identical procedure. After heat sink assembly brought together, it was placed in the groove, center of isolation, and then water was pumped. After pumping, at least 20 minutes were given to remove any possible trapped air especially in the metal foam. Following the stabilization period, starting from the maximum flow rate 14 pressure measurements at various flow rates almost for every specimen were taken. For pressure drop measurements inlet flow temperature was held at $25\pm1^{\circ}\text{C}$. After pressure drop measurements, 60 W of power was supplied to the heater and the entire experimental apparatus was allowed to reach steady-state. Although 20 minutes was observed to be sufficient to reach steady-state, 30 minutes were given. At four different flow rates temperature measurements were taken for every heat sink. Between subsequent measurements, at least 5 minutes were given to let the setup to reach steady-state.

5.3 Microchannel Heat Sinks

In his thesis Alpsan E. [94] has modeled microchannel heat sinks with dimensions suitable for cooling transmitter and receiver (T/R) modules designed in ASELSAN. Pieces were produced by a foreign company and bought as ready to test. Since the purpose is the same, same dimensions were used when producing new microchannels. Channels were produced by using national resources to ensure that mass production of such pieces are possible within Turkey in case of using them as heat sinks in phased-array projects. Brazing was chosen as production method, after machining channels with wire EDM. Wire EDM operation was handled by a sub-contractor of ASELSAN. For brazing operation pre-brazing cover and body pieces were designed. In Appendix A, technical drawings were given. A produced pre-brazing channel body view was given in Figure 5.6. After brazing, bonded pieces have been sent to machining in order to have final form of heat

sinks. Vacuum brazing is only possible way to have a robust system of cooling and only aluminum vacuum brazing furnace owner company is ASELSAN in Turkey.



Figure 5.6 Pre-brazing body of 300 micron channel

Brazing is a joining method that has a nature of inter-atomic metallic bond. Joining has equal or even greater strength than base material. In vacuum, pieces are heated to the melting temperature of filler sheet material but base metal never melts down. Strong, uniform and leak proof joints are possible with this joining method. Since the filler material is again another alloy of aluminum, brazed aluminum assemblies conduct heat uniformly. High cost of operation and restriction on alloy type to be used as base material can be said as negative aspects of vacuum brazing operation. A view of microchannel heat sink after brazing was given below in Figure 5.7.



Figure 5.7 Brazed channel assembly

6063-T6 aluminum alloy was used as material of microchannel heat sink. Heat sinks with four different channel widths were produced. A closer view of produced channels with brazing fillet formed at contact edges is given in Figure 5.8.

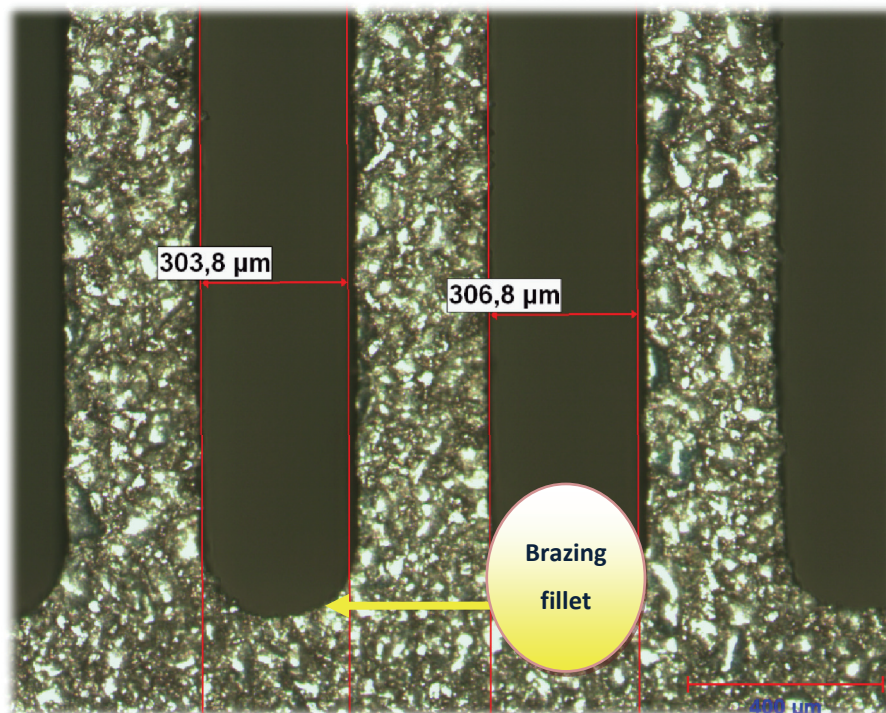


Figure 5.8 Microscope view of 300 micron channel

Microchannel test specimens produced are listed in Table 5.2.

Table 5.2 Dimensional parameters of microchannel test specimens

Specimen name	Channel width	Channel height	Hydraulic diameter	Aspect ratio	Number of channels
300 micron	300 μm	4 mm	558 μm	13,33	24
420 micron	420 μm	4 mm	760 μm	9,52	20
500 micron	500 μm	4 mm	888 μm	8,00	18
900 micron	900 μm	4 mm	1469 μm	4,44	12

A view of final product is given in Figure 5.9.

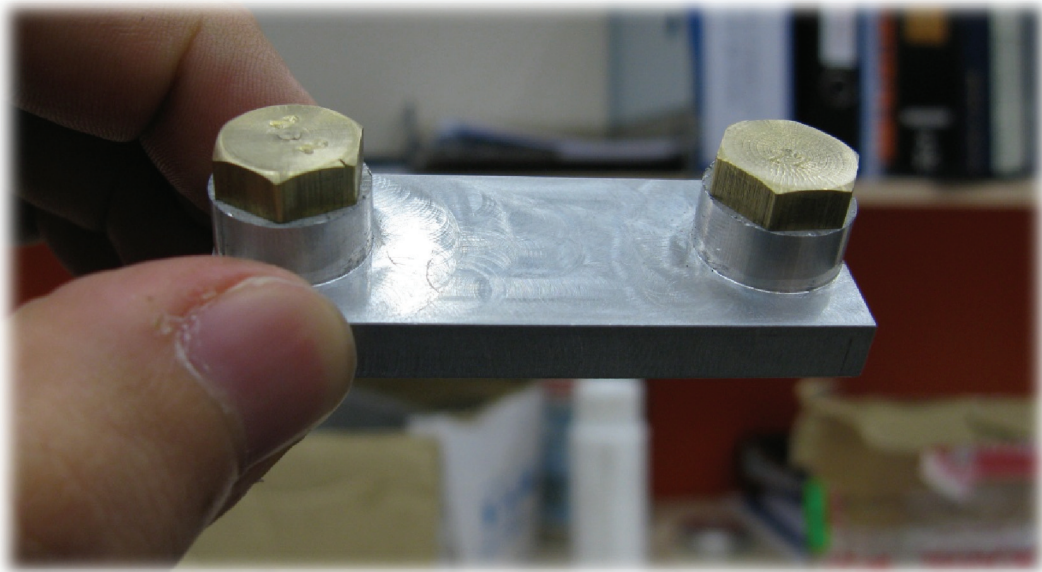


Figure 5.9 Final product

5.4 Metal Foam Heat Sinks

Open cell metal foams used were manufactured by ERG from 6101-T6 aluminum alloy. In this thesis, only aluminum open cell metal foam of 92 % porosity was used. Three different pore dimensions were available: 10ppi (pores per linear inch), 20ppi and 40ppi. Both uncompressed and compressed metal foams were tested. Three types of foams with different pore diameters were compressed with a factor of two and three. Compressed and uncompressed foams were machined to their final dimensions by using the most appropriate cutting method, wire EDM, as explained in Chapter 3. Final dimensions were the same as microchannel heat sink filled volume with channels (31,9 mm x 14,1 mm x 4 mm). Flow cross section for all heat sinks is 14,1 mm x 4 mm.

Bonding metal foams to the heat sink body again accomplished with brazing. As discussed in Chapter 3, although other methods are available, most appropriate method for bonding aluminum metal foam is vacuum brazing. This is the method that turns metal foam into high performance heat sinks. ERG DUOCEL aluminum metal foams are suitable for brazing and actually they were produced with that special alloy intentionally.

Although they have same porosity values, uncompressed foams are labeled by their characteristic pore diameter in inches which are 10, 20, and 40 ppi. Open cell aluminum metal foam production of ERG has almost uniform homogenous properties. These foams have open pore diameters of 6.9 mm, 3.6 mm, and 2.3 mm, respectively. The porosity value for each foam with their final dimensions was calculated by dividing its weight by the volume, after measuring external dimensions, and then comparing this value to the density of the specified solid metal alloy. Some physical properties of metal foams that were tested are presented in Table 5.3 below.

Table 5.3 Physical properties of tested metal foams

Physical Properties of Uncompressed Foam				
Specimen name	Foam	Pore diameter (mm)	Specific surface Area (m²/m³)	Measured porosity (%)
foam 10-01	10 ppi	6.9	820	92.1
foam 20-01	20 ppi	3.6	1700	92.0
foam 40-01	40 ppi	2.3	2700	92.8
Physical Properties of Compressed Foam				
Specimen name	Initial uncompressed foam	Compression factor (M)	Expected porosity (%)	Measured porosity (%)
foam 10-03	10 ppi	3	76	81,0
foam 20-02	20 ppi	2	84	86,6
foam 20-03	20 ppi	3	76	79,1
foam 40-02	40 ppi	2	84	87,1
foam 40-03	40 ppi	3	76	80,1

Compression factor “*M*” was defined as the ratio of the uncompressed foam height to the final compressed height. ε was designated as porosity and with the following simple relationship expected porosity can be calculated:

$$\varepsilon_{compressed} = 1 - M(1 - \varepsilon_{uncompressed}) \quad (5.1)$$

To compress metal foam under press in a controlled manner, a compression jig was designed and produced. With this jig, six foam samples were compressed as seen in Figure 5.10.



a. Compressing under press



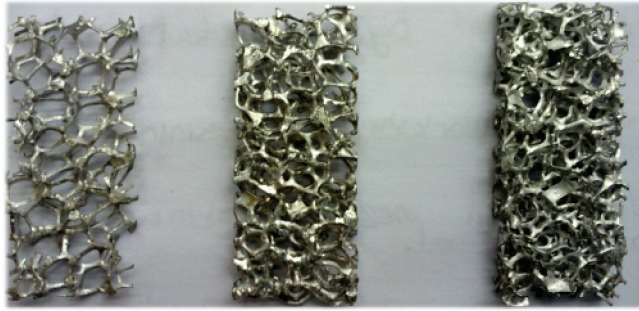
b. Compression jig



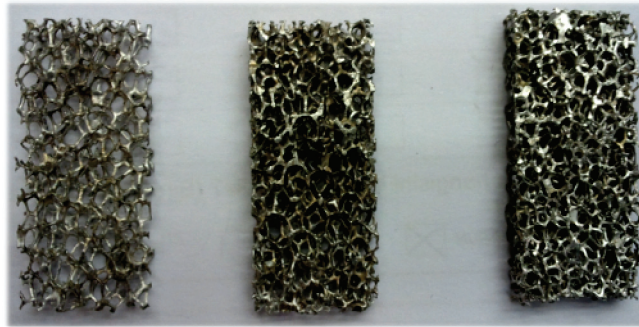
c. Final foam thickness

Figure 5.10 Compressing operation of metal foams

During compression, one of the lateral sides of foam was allowed to expand freely and other hold constant. As the foam was being compressed, mass of foam that extended beyond the original lateral dimensions was lost when the foam was machined to its final tolerances. Due to this reason final porosities were higher than expected porosities.



a. 10ppi metal foam pieces. Compression factor 1,2, and 3



b. 20ppi metal foam pieces. Compression factor 1,2, and 3



c. 40ppi metal foam pieces. Compression factor 1,2, and 3

Figure 5.11 Metal foam pieces before brazing operation

After compression, metal foams were cut to their final dimensions via wire EDM, and a view of this operation is given in Figure 5.12.

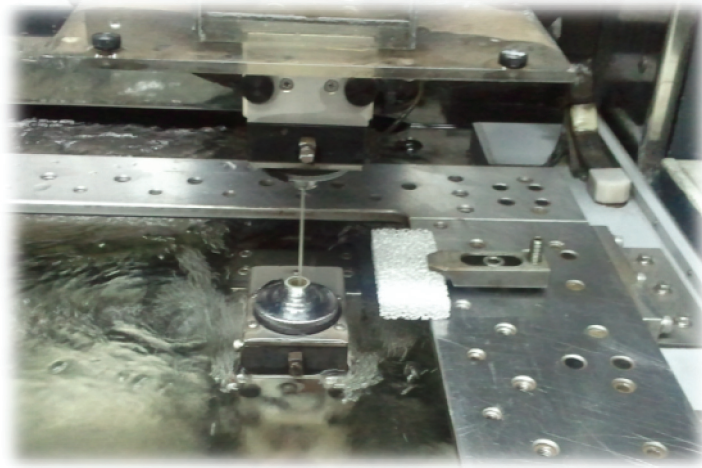


Figure 5.12 Metal foam cutting with wire EDM

Finally, metal foams were brazed to AA6063-T6 empty bodies and brazed assemblies were machined to their final dimensions which are exactly the same as the microchannel heat sinks. In Appendix A, technical drawing of empty body was also given.

5.5 Uncertainty Analysis

Uncertainty analysis has been conducted for calculated properties and they were presented both in tabulated and graphical form in subsequent sections of this study. Before the discussion of experimental data and results gained after tests, in this section analysis method followed is briefly explained.

Unfortunately measurements taken can never be exact. Every measurement has its own uncertainty. Especially for experimental studies it is crucial to inform the reader about uncertainties of presented data to let him/her be aware of the accuracy of results. Possible reasons for uncertainties can be:

- Measuring device calibration is not properly done
- Repeated measurements results in different results
- Reading accuracy is not sufficient, e.g. measuring device shows only a few digits

- Constants that are used are not exact
- Change in some external variable like temperature, is not constant but deviates slightly

Therefore, statement of accuracy of presented data becomes an important issue. In this study, experimental results have been presented generally with graphs and these graphs included error bars. Uncertainty analysis for related measurements and products of measured variables has been conducted and the method was presented in this section.

Detailed information of uncertainty analysis is beyond of this text therefore only required formulations and subjects are given.

Error of a measurement can be defined as follows:

$$e = X_m - X_T \quad (5.2)$$

where X_m is the measured value of a quantity and

X_T is the true value of the measured quantity.

Generally measured value is in between $X + e$ and $X - e$. According to rectangular distribution uncertainty can be calculated using known error [91].

$$U = e/\sqrt{3} \quad (5.3)$$

For imported errors originated from sensor employed in experimental setup generally full scale output (FSO) uncertainty given by manufacturer. This percentage uncertainty can be converted by using following equation [91]:

$$U = 1.5U_f/\sqrt{3} \quad (5.4)$$

In above equation, U_f is uncertainty value given by manufacturer.

If error is caused when display value has some number of digits like 15,0006 real measured value can be in between 15,00055 and 15,00065. In this case uncertainty due to parameter resolution can be estimated as [92]:

$$U = 0,00005/\sqrt{3} \quad (5.5)$$

Let $S(A,B,C)$ be a result that depends on the measured quantities A , B and C . Consider that measured A has an uncertainty U_A . S has some uncertainty due to its dependence on A [93]:

$$U_{SA} \cong \frac{dS}{dA} U_A \quad (5.6)$$

In a similar fashion uncertainties of S caused by B and C are:

$$U_{SB} \cong \frac{dS}{dB} U_B \quad (5.7)$$

and

$$U_{SC} \cong \frac{dS}{dC} U_C \quad (5.8)$$

When A, B and C vary randomly relative to each other as it is the general case, overall product of uncertainty can be formulated [93]:

$$U_S \cong \sqrt{U_{SA}^2 + U_{SB}^2 + U_{SC}^2} \quad (5.9)$$

Uncertainty is always positive and if it is wanted to be converted to a plus and minus error, uncertainty can be expressed as $\pm k$ times the uncertainty. This factor k is generally accepted as 2.

In this chapter, graphs were plotted as a function of Reynolds number and derivation of uncertainty becomes important. Reynolds number was given in Chapter 4 as:

$$Re = \rho u \frac{d_{hyd}}{\mu} \quad (5.10)$$

Since hydraulic diameter and flow velocity cannot be read directly, they should be replaced with following equations:

$$u = Q/(ab) \text{ and } d_{hyd} = \frac{4ab}{2(a+b)} = \frac{2ab}{(a+b)} \quad (5.11)$$

where “a” and “b” are channel dimensions given in Chapter 4. Then, Equation (5.9) becomes:

$$Re = \rho \frac{Q}{ab} \frac{\frac{2ab}{(a+b)}}{\mu} = \frac{2\dot{m}}{\mu(a+b)} \quad (5.12)$$

Uncertainty of Reynolds number can be defined as follows:

$$\frac{U_{Re}}{Re} = \sqrt{\left(\frac{1}{Re} \frac{\partial Re}{\partial \dot{m}} U_{\dot{m}}\right)^2 + \left(\frac{1}{Re} \frac{\partial Re}{\partial \mu} U_{\mu}\right)^2 + \left(\frac{1}{Re} \frac{\partial Re}{\partial a} U_a\right)^2 + \left(\frac{1}{Re} \frac{\partial Re}{\partial b} U_b\right)^2} \quad (5.13)$$

The derivatives can now be computed as:

$$\frac{1}{Re} \frac{\partial Re}{\partial \dot{m}} U_{\dot{m}} = \frac{\mu(a+b)}{2\dot{m}} \frac{2\rho}{\mu(a+b)} U_{\dot{m}} = \frac{U_{\dot{m}}}{\dot{m}} \quad (5.14)$$

$$\frac{1}{Re} \frac{\partial Re}{\partial \mu} U_{\mu} = \frac{-\mu(a+b)}{2\dot{m}} \frac{2\rho Q}{(a+b)\mu^2} U_{\mu} = \frac{-U_{\mu}}{\mu} \quad (5.15)$$

$$\frac{1}{Re} \frac{\partial Re}{\partial a} U_a = \frac{-\mu(a+b)}{2\dot{m}} \frac{2\rho Q}{\mu(a+b)^2} U_a = \frac{-U_a}{a+b} \quad (5.16)$$

$$\frac{1}{Re} \frac{\partial Re}{\partial a} U_b = \frac{-\mu(a+b)}{2\dot{m}} \frac{2\rho Q}{\mu} \frac{1}{(a+b)^2} U_b = \frac{-U_b}{a+b} \quad (5.17)$$

Substituting Equations (5.13), (5.14), (5.15), and (5.16) into (5.12):

$$\frac{U_{Re}}{Re} = \sqrt{\left(\frac{U_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{-U_{\mu}}{\mu}\right)^2 + \left(\frac{-U_a}{a+b}\right)^2 + \left(\frac{-U_b}{a+b}\right)^2} \quad (5.18)$$

In a similar fashion, expression for uncertainty in estimating the product fRe for rectangular channels was conducted by Steinke and Kandlikar [87] as:

$$\frac{U_{fRe}}{fRe} = \sqrt{3\left(\frac{U_Q}{Q}\right)^2 + 2\left(\frac{U_{\rho}}{\rho}\right)^2 + \left(\frac{-U_{\mu}}{\mu}\right)^2 + 2\left(\frac{-U_a}{a+b}\right)^2 + 2\left(\frac{-U_b}{a+b}\right)^2 + \left(\frac{U_{\Delta P}}{\Delta P}\right)^2 + \left(\frac{U_L}{L}\right)^2 + 5\left(\frac{U_a}{a}\right)^2 + 5\left(\frac{U_b}{b}\right)^2} \quad (5.19)$$

Nusselt number formulation was given as:

$$Nu = \frac{qd_{hyd}}{k_f A_{con}(T_{pl} - T_{c,in})} = \frac{\dot{m} c d_{hyd}(T_{c,out} - T_{c,in})}{k_f A_{con}(T_{pl} - T_{c,in})} = \frac{\dot{m} c 2ab(T_{c,out} - T_{c,in})}{k_f A_{con}(T_{pl} - T_{c,in})(a+b)} \quad (5.20)$$

Uncertainty was derived like Reynolds number and here only the related result was given:

$$\frac{U_{Nu}}{Nu} = \sqrt{\left(\frac{U_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{U_c}{c}\right)^2 + \left(\frac{U_a}{a}\right)^2 + \left(\frac{U_b}{b}\right)^2 + \left(\frac{U_{T_{c,out}}}{T_{c,out}}\right)^2 + \left(\frac{-U_{T_{c,in}}}{T_{c,in}}\right)^2 + \left(\frac{-U_{k_f}}{k_f}\right)^2 + \left(\frac{-U_{A_{con}}}{A_{con}}\right)^2 + \left(\frac{-U_{T_{pl}}}{T_{pl} - T_{c,in}}\right)^2 + \left(\frac{U_{T_{c,in}}}{T_{pl} - T_{c,in}}\right)^2 + \left(\frac{-U_a}{a+b}\right)^2 + \left(\frac{-U_b}{a+b}\right)^2} \quad (5.21)$$

Uncertainty in pumping power was given by:

$$\frac{U_{\dot{W}}}{\dot{W}} = \sqrt{\left(\frac{U_{\Delta p}}{\Delta p}\right)^2 + \left(\frac{U_Q}{Q}\right)^2} \quad (5.22)$$

Uncertainty in volumetric flow rate can be conducted as:

$$\frac{U_Q}{Q} = \sqrt{\left(\frac{U_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{U_{\rho}}{\rho}\right)^2} \quad (5.23)$$

Uncertainty in measured flow velocity is

$$\frac{U_u}{u} = \sqrt{\left(\frac{U_Q}{Q}\right)^2 + \left(\frac{U_A}{A}\right)^2} \quad (5.24)$$

where U_A and A are uncertainty in flow area and flow area respectively.

Finally, uncertainty of thermal resistance can be found as:

$$\frac{U_{R_{th}}}{R_{th}} = \sqrt{\left(\frac{U_{T_{pl}}}{T_{pl}}\right)^2 + \left(\frac{-U_{T_{c,in}}}{T_{c,in}}\right)^2 + \left(\frac{-U_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{-U_c}{c}\right)^2 + \left(\frac{-U_{T_{out}}}{T_{c,out}-T_{c,in}}\right)^2 + \left(\frac{U_{T_{c,in}}}{T_{c,out}-T_{c,in}}\right)^2} \quad (5.25)$$

A sample calculation of uncertainty values is given in Appendix B.

5.6 Microchannel pressure drop results and discussion

For each tested channel, pressure drop in laminar flow was measured generally at 14 different Reynolds numbers. Measured pressure drops for 300 micron channel were compared to conventional theory in Figure 5.13. Also both calculated and measurements as a result of experimental study were presented in Table 5.4 with related uncertainties in data obtained. Results for other channels were given in subsequent pages both in graphical and tabulated form. When Figure 5.13 is investigated, a good agreement in trend with classical theory is observed.

Quadratic curve that fully represents the effect of increasing flow velocity is clear. There is an offset in measurements with respect to the analytical results that resulted in high ratio of error especially for lower Reynolds number values. In literature, although it is possible to find higher pressure drops and Poiseuille number values compared to the classical theory, this result was not attributed to discrepancy from conventional theory. This systematic error might be assigned to:

- Uncertainties as given in Figure 5.13 and Table 5.4.
- Instead of employing differential pressure transducer, using vacuum pressure sensor due to availability. Although, before every measurement, pressure transducers were leveled at atmospheric pressure, since error in measurements were doubled, a shift in pressure drop value due to the shift in pressure reading in either pressure transducer is possible within uncertainties supplied by manufacturer.
- Pressure drop due to inlet and exit manifold design. To determine pressure drop analytically Equation (4.29) is employed. Even though minor pressure losses are included in the calculation, design may include more minor losses.

Alpsan E. [94] also studied microchannels of same channel width experimentally and gathered pressure drop measurements at three different flow rates. Measurements were also compared to his study.

For other microchannels tested, although agreement has been observed to be better for lower Reynolds number values, divergence has been seen and above reasons might also be valid for other channel dimensions. Discrepancy at high Reynolds numbers in 500 micron channel was even more difficult to explain; although great care was given, some channels appeared to be clogged.

A sample analytical calculation for pressure drop in microchannel was presented in Appendix C.

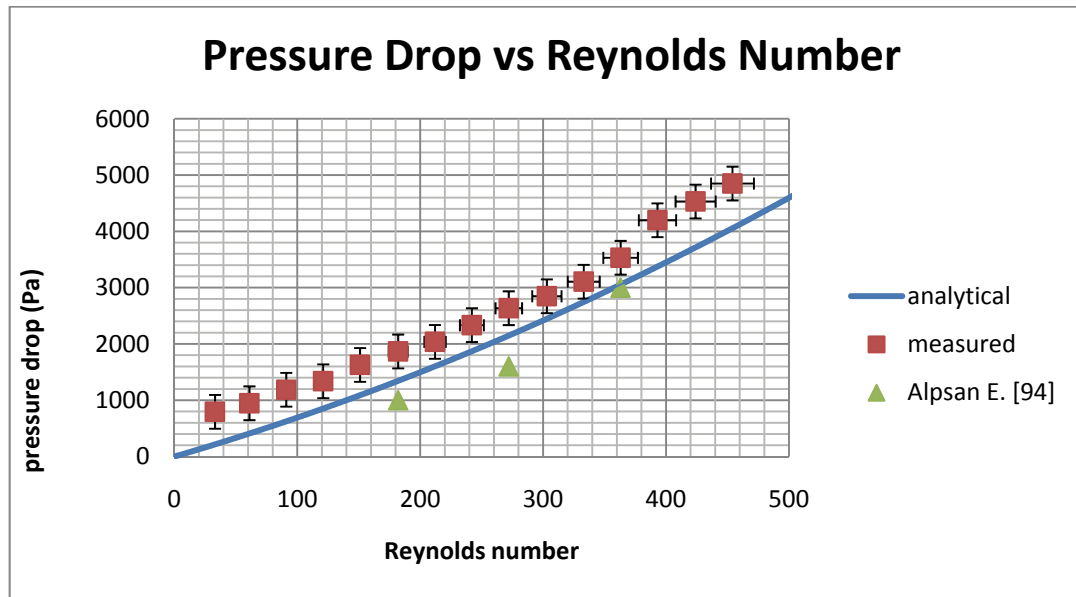


Figure 5.13 Experimental vs. analytical pressure drop results comparison for 300 micron channel

Table 5.4 300 micron channel experimental and analytical results and related uncertainties

Mass flow rate (kg/h)	Re	Calculated		Measured		Uncertainties		
		Δp (Pa)	$ff_{app} * Re$	Δp (Pa)	$ff * Re$	Re (%)	Δp (%)	$ff * Re$ (%)
5	33	205,6	21,9	396,1	43,0	5,7	21,9	13,4
10	61	389,3	22,0	547,4	31,9	3,5	15,8	10,8
15	91	595,5	22,1	787,6	30,5	2,7	11,0	10,1
20	121	811,4	38,9	938,8	27,0	2,4	9,2	9,8
25	151	1037,0	22,3	1229,3	28,2	2,2	7,0	9,7
30	182	1280,4	22,4	1468,3	27,7	2,1	5,9	9,6
35	212	1525,8	22,5	1638,0	26,2	2,1	5,3	9,6
40	242	1781,0	22,6	1935,0	27,0	2,0	4,5	9,6
45	272	2045,9	22,7	2236,7	27,5	2,0	3,9	9,6
50	303	2329,8	22,8	2449,1	26,8	2,0	3,5	9,5
55	333	2614,5	22,9	2707,2	26,7	2,0	3,2	9,5
60	363	2908,9	23,0	3132,3	28,3	2,0	2,8	9,5
65	393	3213,0	23,2	3799,6	31,8	2,0	2,3	9,5
70	424	3537,4	23,3	4132,0	31,8	1,9	2,1	9,5
75	454	3861,3	23,4	4451,1	31,8	1,9	1,9	9,5

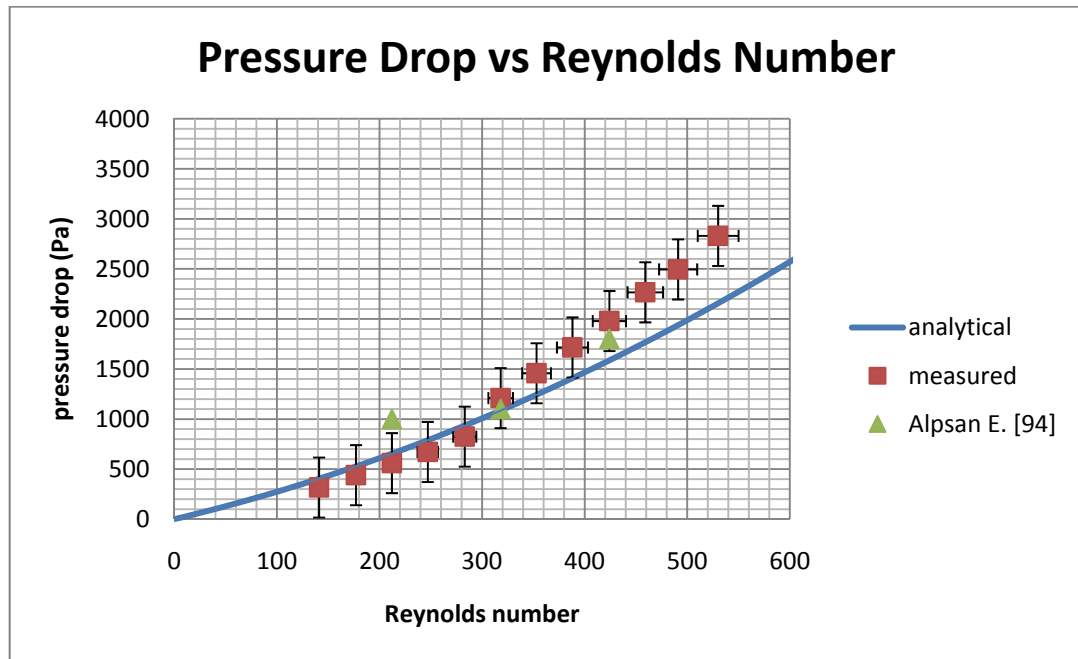


Figure 5.14 Experimental vs. analytical pressure drops results comparison for 420 micron channel

Table 5.5 420 micron channel experimental and analytical results and related uncertainties

Mass flow rate (kg/h)	Re	Calculated		Measured		Uncertainties		
		Δp (Pa)	$ff_{app} * Re$	Δp (Pa)	$ff * Re$	Re (%)	Δp (%)	$ff * Re$ (%)
20	141	384.6	21.8	317.0	19.0	2.4	27.3	7.8
25	177	501.6	21.9	440.7	20.8	2.2	19.7	7.6
30	212	622.6	22.1	561.0	21.8	2.1	15.4	7.5
35	247	743.6	22.3	672.3	22.1	2.1	12.9	7.5
40	283	864.6	22.5	783.6	23.6	2.0	12.5	7.4
45	318	985.6	22.6	894.9	33.2	2.0	4.2	7.4
50	353	1106.6	22.8	1006.2	32.7	2.0	5.9	7.4
55	388	1227.6	23.0	1117.5	35.6	1.9	5.0	7.4
60	424	1348.6	23.2	1228.8	38.0	1.9	4.4	7.4
65	459	1469.6	23.3	1340.1	40.6	1.9	3.8	7.4
70	491	1590.6	23.5	1451.4	41.3	1.9	3.5	7.4
75	530	1711.6	23.7	1562.7	43.5	1.9	3.1	7.3

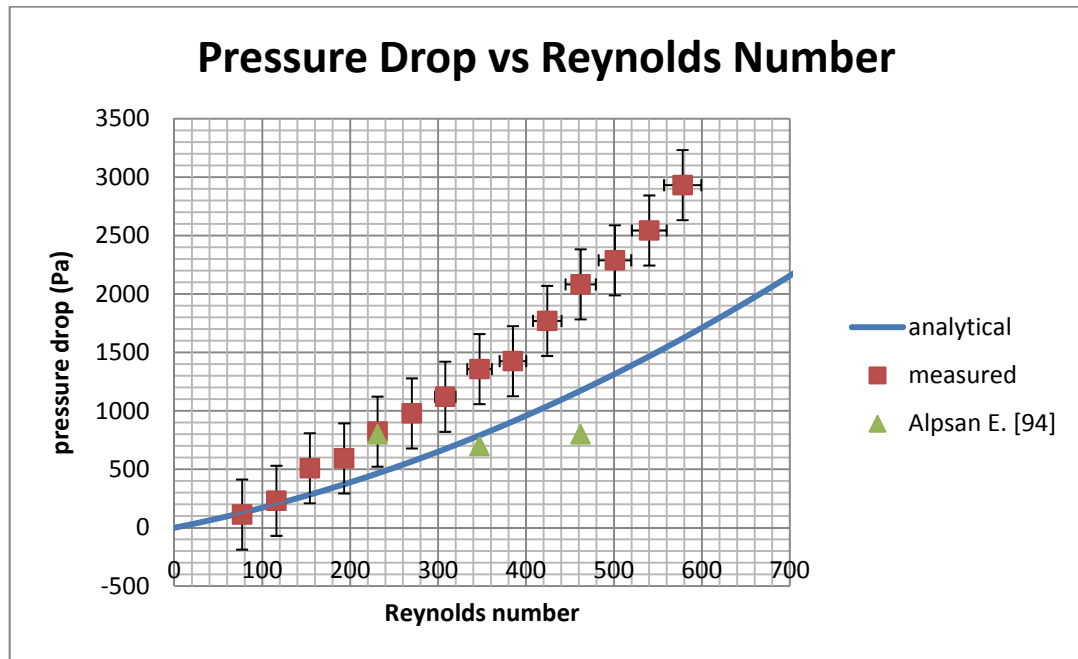


Figure 5.15 Experimental vs. analytical pressure drop results comparison for 500 micron channel

Table 5.6 500 micron channel experimental and analytical results and related uncertainties

Mass flow rate (kg/h)	Re	Calculated		Measured		Uncertainties		
		Δp (Pa)	$ff_{app} * Re$	Δp (Pa)	$ff * Re$	Re (%)	Δp (%)	$ff * Re$ (%)
5	77	121.8	21.1	113.2	20.4	3.5	76.5	8.3
10	116	193.3	21.3	231.0	27.5	2.7	37.5	7.3
15	154	269.3	21.5	509.6	46.3	2.4	17.0	7.0
20	193	353.8	21.8	594.1	42.4	2.2	14.6	6.8
25	531	442.5	22.0	822.9	53.9	2.1	10.5	6.8
30	270	540.1	22.2	979.2	54.6	2.1	8.8	6.7
35	308	641.5	22.5	1121.8	52.1	2.0	7.7	6.6
40	347	752.8	22.7	1358.2	50.0	2.0	6.4	6.6
45	385	866.2	22.9	1426.4	47.9	1.9	6.0	6.6
50	424	989.8	23.2	1770.5	55.5	1.9	4.9	6.6
55	462	1116.6	23.4	2083.5	61.1	1.9	4.2	6.6
60	501	1253.2	23.6	2289.4	62.1	1.9	3.8	6.6
65	540	1396.5	23.9	2544.4	64.0	1.9	3.4	6.5
70	578	1542.4	24.1	2933.0	68.8	1.9	3.0	6.5

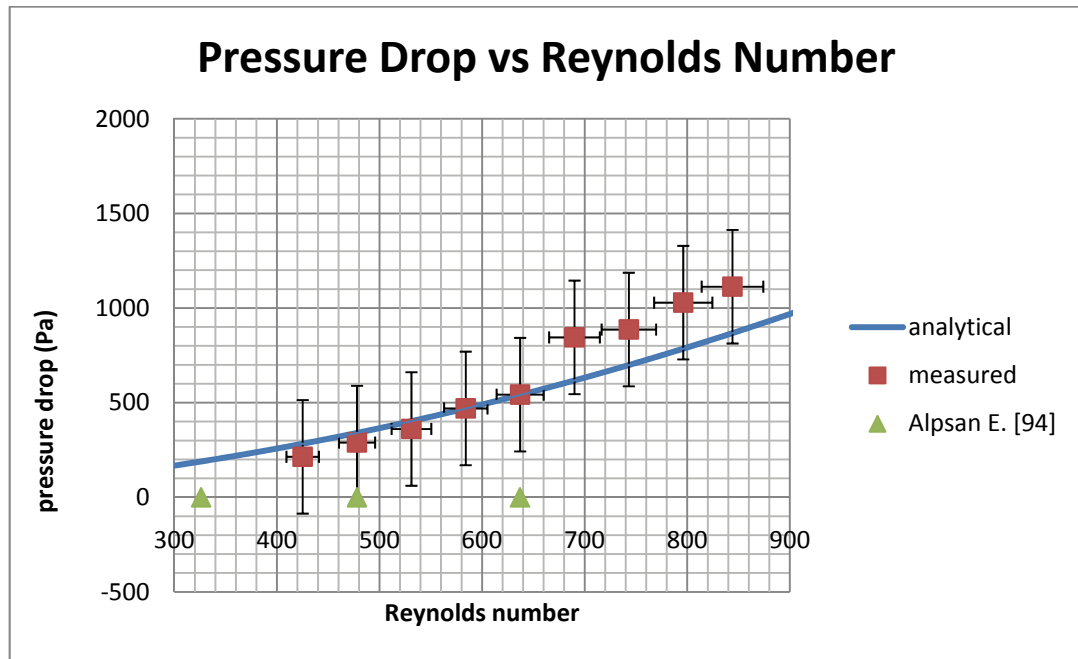


Figure 5.16 Experimental vs. analytical pressure drop results comparison for 900 micron channel

Table 5.7 900 micron channel experimental and analytical results and related uncertainties

Mass flow rate (kg/h)	Re	Calculated		Measured		Uncertainties		
		Δp (Pa)	$ff_{app} * Re$	Δp (Pa)	$ff * Re$	Re (%)	Δp (%)	$ff * Re$ (%)
40	425	269.1	23.7	214.3	25.4	1.9	40.4	5.0
45	478	353.5	24.3	289.6	31.0	1.9	29.9	5.0
50	531	382.6	25.0	361.3	34.9	1.9	24.0	4.9
55	584	446.2	25.6	469.9	42.0	1.9	18.4	4.9
60	637	514.5	26.2	542.9	44.0	1.9	16.0	4.9
65	690	587.4	26.8	845.5	67.7	1.8	10.2	4.9
70	743	665.0	27.5	886.8	64.4	1.8	9.8	4.9
75	796	747.1	28.1	1028.9	69.9	1.8	8.4	4.9
80	844	825.5	28.7	1113	70.6	1.8	7.8	4.9

5.7 Microchannel heat transfer results and discussion

In Figure 5.14, variation of the Nusselt number sketches of microchannels with Reynolds number is presented for microchannels. Also, calculated and measured Nusselt number and measured thermal resistance values were presented with uncertainties in Table 5.5.

Analytical Nusselt numbers are calculated by using Table 4.1 and sample calculation is performed in Appendix C. It is observed that analytical results underestimated experimental results. Actually, lower Nusselt number values are expected since heated area is smaller than effective finned area of channels. This may result in higher heater plate surface temperatures for the same heat power due to higher heat flux when inlet and outlet temperature difference remains the same. Since heat insulation was employed adequately, calculated power loss was lower than %10.

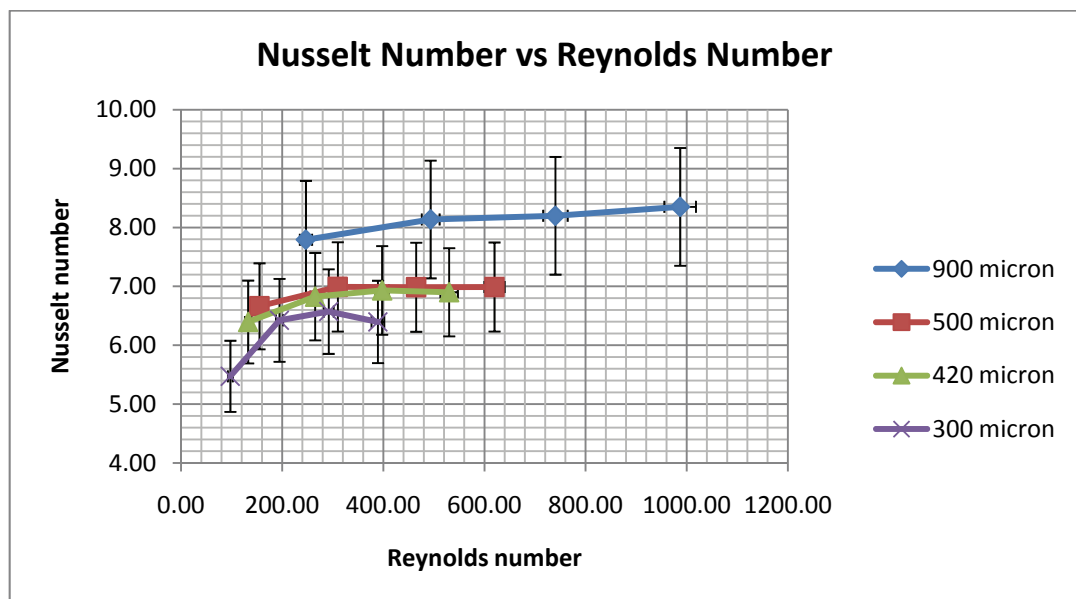


Figure 5.17 Nusselt number vs. Reynolds number experimental results

Table 5.8 Heat transfer calculated and experimental results of microchannels

								Uncertainties		
	$\dot{m}$ (kg/h)	Q (l/min)	Re	Nu	L_t	Nu_c	R_{th} (°C/W)	Re (%)	U_{Nu} (%)	U_{Rth} (%)
300 micron	15	0,25	97,28	5,47	29,8	8,26	0,55	2,47	11,04	3,27
	30	0,50	194,57	6,42	59,7	8,25	0,47	1,83	10,96	4,97
	45	0,75	291,85	6,57	89,5	8,27	0,46	1,68	10,93	7,09
	55	1,00	389,13	6,40	119	8,32	0,47	1,63	10,93	9,68
420 micron	15	0,25	132,50	6,40	55,3	8,25	0,64	2,47	11,00	3,30
	30	0,50	265,00	6,83	110	8,30	0,60	1,83	10,88	5,05
	45	0,75	397,49	6,93	166	8,45	0,59	1,68	10,86	7,38
	56	1,00	529,99	6,90	221	8,50	0,60	1,63	10,86	9,90
500 micron	15	0,25	154,93	6,66	75,7	8,16	0,72	2,47	10,97	3,18
	30	0,50	309,86	6,99	151	8,36	0,69	1,83	10,85	4,93
	45	0,75	464,79	6,99	227	8,61	0,69	1,68	10,83	7,38
	60	1,00	619,73	6,99	302	8,88	0,69	1,63	10,82	9,89
900 micron	15	0,25	246,73	7,79	207	7,28	1,03	2,45	10,93	3,28
	30	0,50	493,46	8,14	414	8,29	0,98	1,79	10,81	5,22
	45	0,75	740,19	8,20	620	9,17	0,97	1,64	10,79	7,84
	60	1,00	986,92	8,35	827	9,97	0,96	1,58	10,78	10,38

In Table 5.5, $\dot{m}$ is the measured mass flow rate, Q is the volumetric flow rate, Nu is the measured Nusselt number, L_t is the thermal entrance length and Nu_c is the calculated Nusselt number by using Table 4.1.

As seen in Table 5.5 and Figure 5.14 with increasing Reynolds number, Nusselt number also increased and thermal resistance was observed to decrease except for 300 micron channel at 55 kg/h mass flow rate which was attributed to uncertainties. Developing boundary effects greatly influence the Nusselt number and thermal resistance values. As given in the table all thermal entrance length estimations are greater than the channel length, 31.9 mm, except for 300 micron channel at 15 kg/h mass flow rate. Therefore, including thermal entrance effects has vital importance.

5.8 Metal foam pressure drop results and discussion

As microchannels, metal foam heat sinks were tested in a flow range of 0.17 l/min to 1.33 l/min.

Modified Darcy law was given in Chapter 4 as follows:

$$\frac{\Delta p}{L} = \frac{\mu}{K} U + \rho C U^2 \quad (5.26)$$

$$A = \frac{\mu}{K} \text{ and } B = \rho C \quad (5.27)$$

Yields following simpler quadratic equation:

$$\frac{\Delta p}{L} = AU + BU^2 \quad (5.28)$$

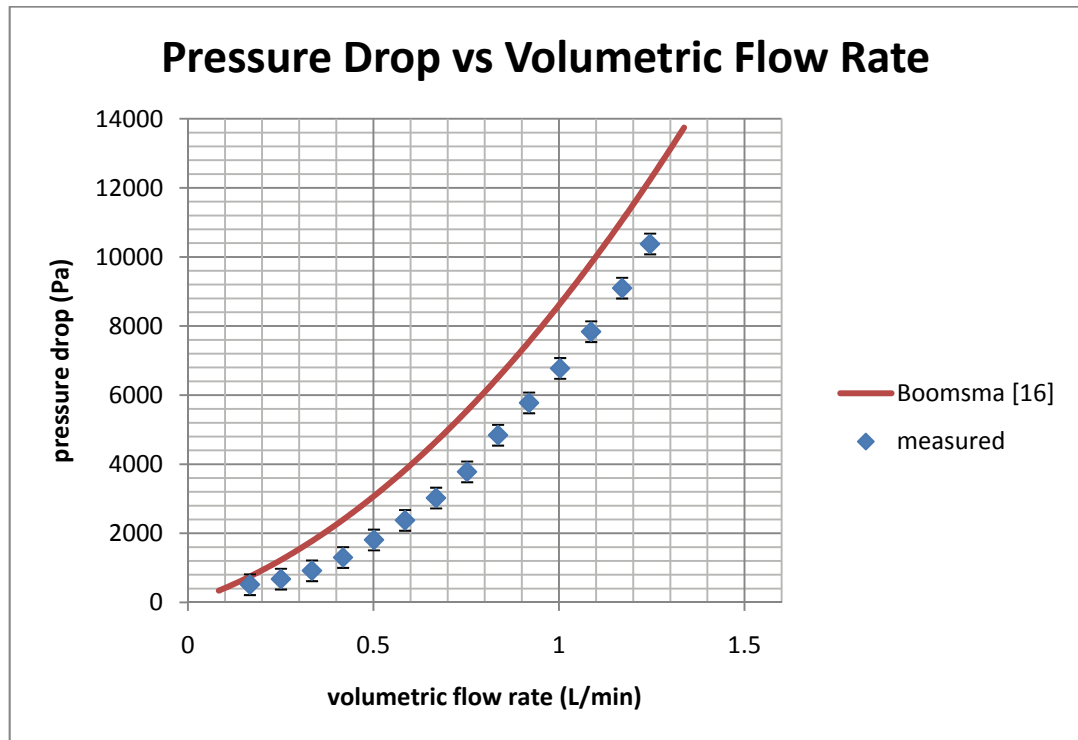
A and B in Equation 5.28 was calculated by employing least squares quadratic curve to fit to the experimentally collected data. In Table 5.6, permeability K and form coefficient C values were presented. Since pressure drop in foam 10-01 could not be measured due to the high uncertainties at low pressure drop values, for that heat sink pressure drop was directly referred to Boomsma's study [16].

As Antohe et al. [53] emphasized that since linear and quadratic effects are always present, determined hydraulic parameters are only valid for testing velocity range from which they are determined. Hence, presented permeability and form coefficients are only valid for mentioned porous media in the range of testing velocities.

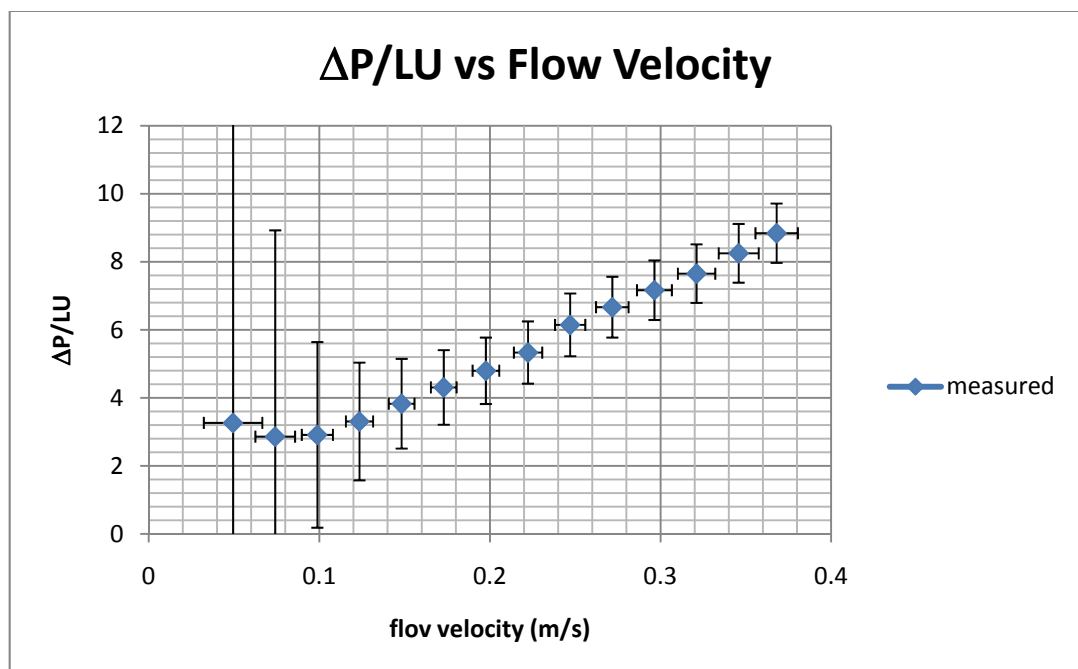
Table 5.9 Permeability and form coefficients of metal foams tested. Values of foam 10-01 were taken from Boomsma's study [16]

Foam	$K (10^{-10} \text{ m}^2)$	$C (\text{m}^{-1})$	$A (\text{kPa-s/ m}^2)$	$B (\text{kPa- s}^2/\text{ m}^3)$
foam 10-01	3529	120,0	2,46	119,65
foam 10-03	916	1828,8	9,49	1823,59
foam 20-01	2250	227,0	3,86	226,68
foam 20-02	103	713,1	84,38	711,02
foam 20-03	350	1478,3	24,85	1473,98
foam 40-01	251	233,5	34,56	232,84
foam 40-02	53	931,0	163,8	928,28
foam 40-03	154	2234,3	56,41	2227,78

Boomsma also studied with similar metal foams. If available, tested metal foam was compared graphically with nearest foam studied by Boomsma. In his study similar to this one, Boomsma tested 10, 20, 40 ppi uncompressed aluminum metal foams and 40 ppi compressed metal foams by a compressing factor 2 and 3. Testing range for uncompressed foams is similar, however testing range of compressed foam is different from this study. Results of foam 40-03 are presented in tabular and graphical form below, results for other foams were given in subsequent pages. Quadratic relationship is obvious as seen from pressure drop versus volumetric flow rate graphs. Also, to observe where flow leaves Darcian behavior, $\Delta p/LU$ versus flow velocity graphs were drawn. At low flow rates $\Delta p/LU$ becomes constant except for metal foams 10-03 and 20-02 due to uncertainties.

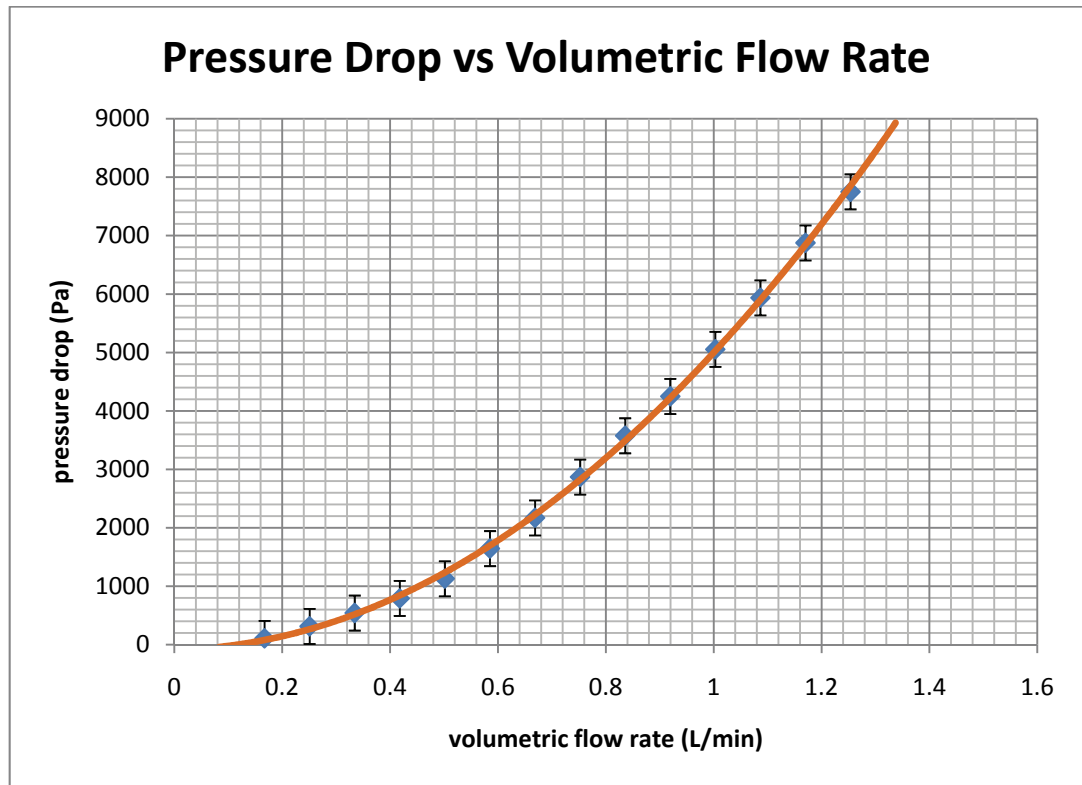


a. Pressure drop vs. volumetric flow rate comparison graph

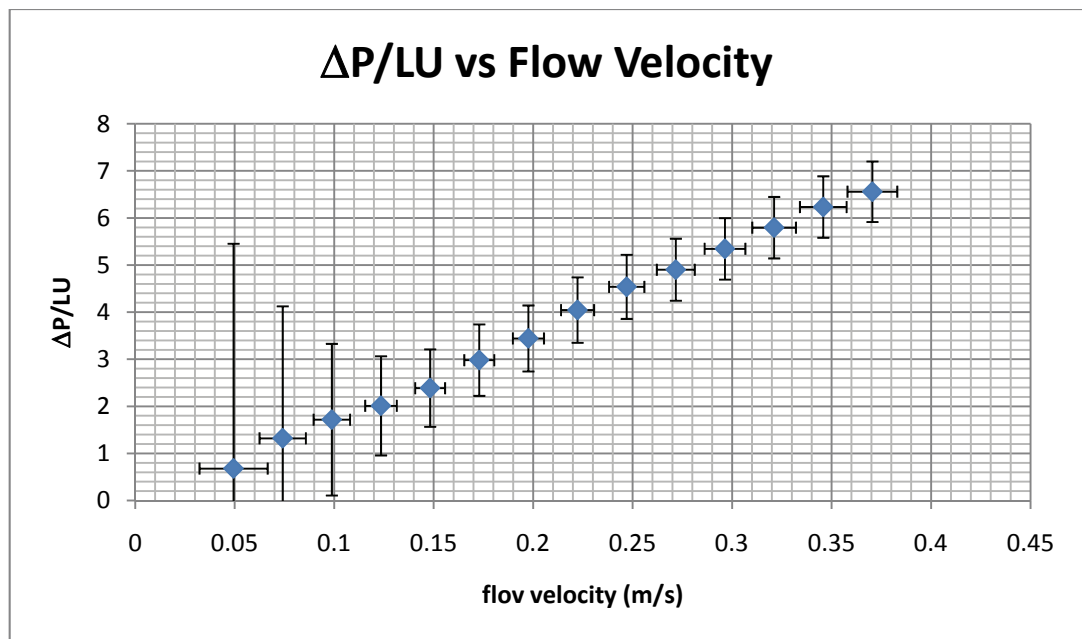


b. $\Delta p/LU$ vs. flow rate comparison graph

Figure 5.18 Experimental vs. Boomsma [16] results comparison for foam 40-03



a. Pressure drop vs. volumetric flow rate graph



b. $\Delta p/LU$ vs. flow rate graph

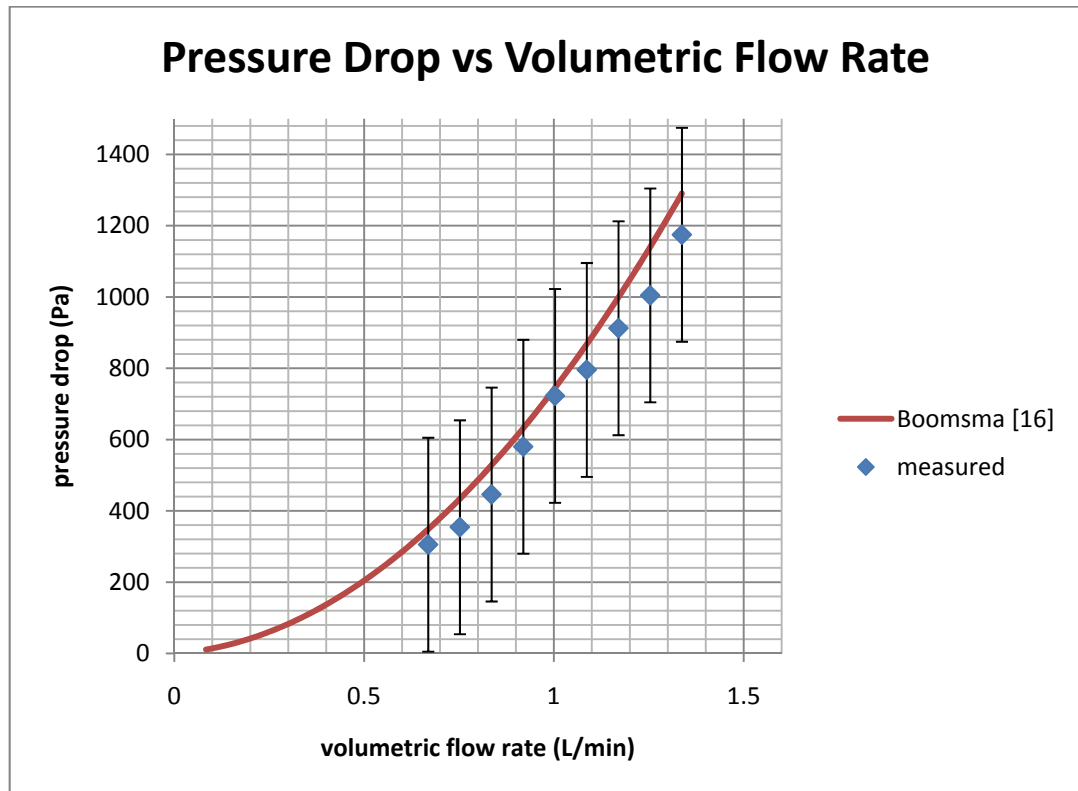
Figure 5.19 Metal foam 10-03 pressure drop results

Table 5.10 Metal foam 10-03 pressure drop results

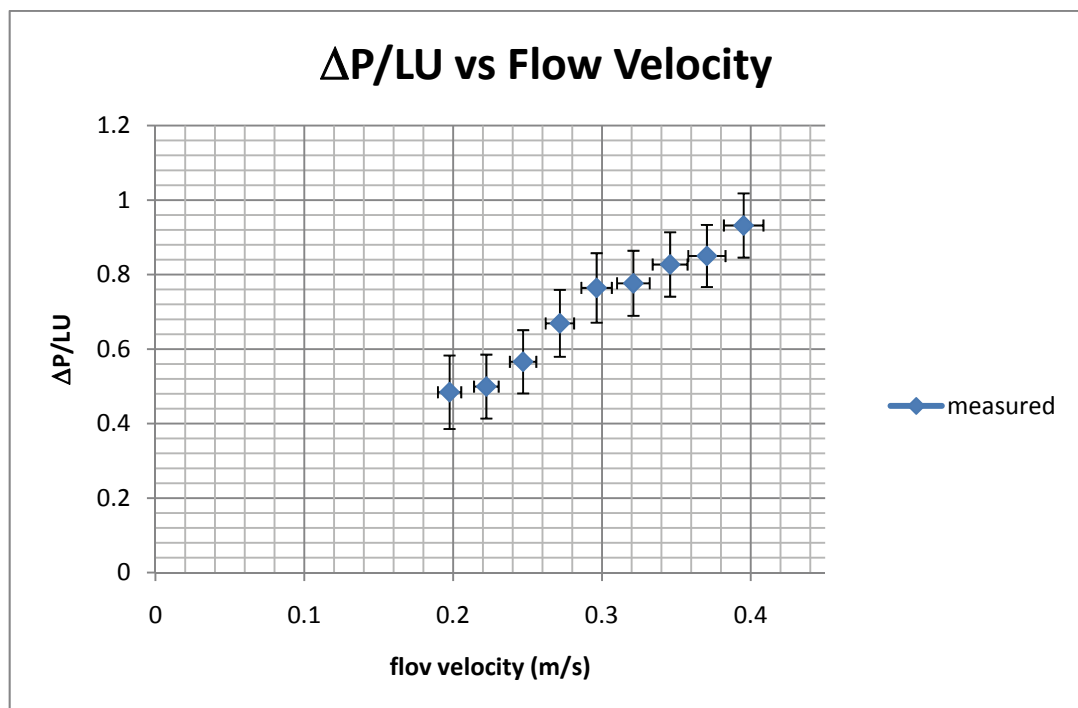
Mass flow rate (kg/h)	Measured values	Uncertainties			
	Re_k	Δp (Pa)	$\Delta p/L$ (Pa/m)	Δp (Pa)	$\Delta p/L$
5	17.2	107.1	3357.4	80.9	80.9
10	25.8	312.5	9796.2	27.7	27.8
15	34.3	541.6	16978.1	16.0	16.1
20	42.9	792.6	24846.4	10.9	11.1
25	51.5	1129.1	35395.0	7.7	7.9
30	60.1	1645.1	51570.5	5.3	5.6
35	68.6	2170.2	68031.3	4.0	4.4
40	77.2	2869.1	89940.4	3.0	3.5
45	85.8	3576.4	112112.9	2.4	3.0
50	94.4	4249.3	133206.9	2.0	2.7
55	103.0	5054.5	158448.3	1.7	2.4
60	111.5	5936.4	186094.0	1.5	2.3
65	120.1	6786.4	215561.1	1.3	2.2
70	128.7	7751.4	242990.6	1.1	2.1

Table 5.11 Metal foam 20-01 pressure drop results

Mass flow rate (kg/h)	Calculated by using Boomsma's K and C for 20PPI		Measured values			Uncertainties	
	Δp (Pa)	$\Delta p/L$ (Pa/m)	Re_k	Δp (Pa)	$\Delta p/L$ (Pa/m)	Δp (Pa)	$\Delta p/L$
40	347.0	10878	107.6	305.4	9572.1	28.4	28.4
45	432.1	13546	121.1	354.3	11107	24.4	24.5
50	526.5	16505	134.5	446.1	13984	19.4	19.5
55	630.2	19755	148.0	580.1	18185	14.9	15.0
60	743.1	23295	161.5	722.8	22658	12.0	12.1
65	865.3	27126	175.0	795.7	24944	10.9	11.0
70	996.8	31247	188.4	912.6	28608	9.5	9.7
75	1137.5	35659	201.8	1004.8	31498	8.6	8.8
80	1287.6	40362	215.3	1174.9	36831	7.4	7.6

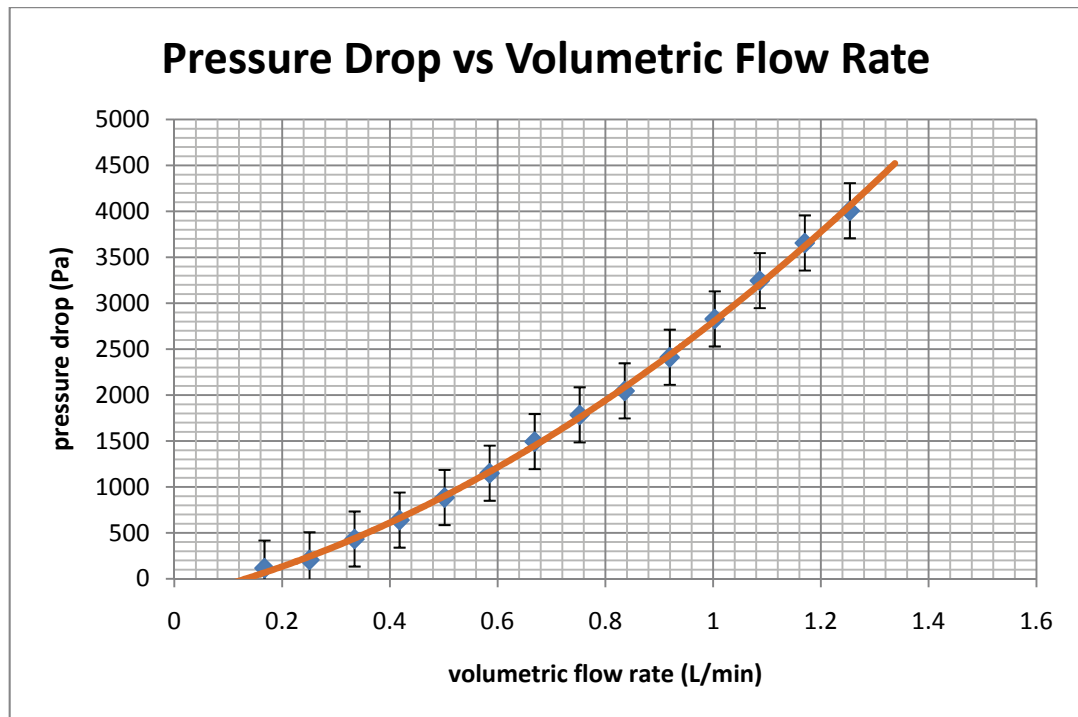


a. Pressure drop vs. volumetric flow rate comparison graph

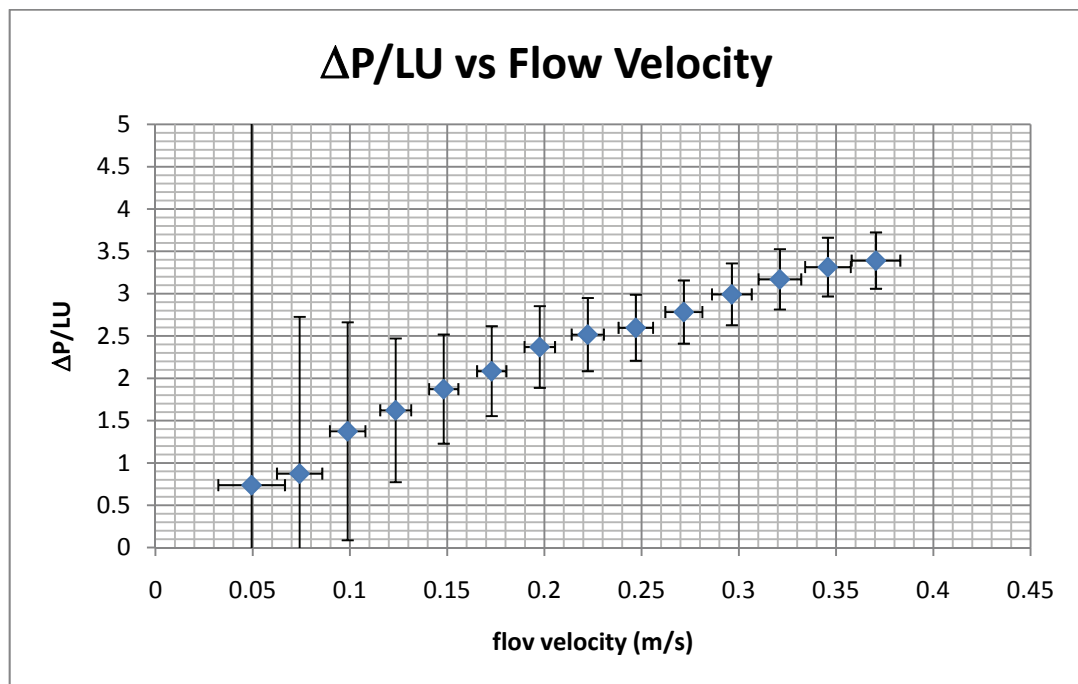


b. $\Delta p/LU$ vs. flow rate comparison graph

Figure 5.20 Metal foam 20-01 pressure drop results



a. Pressure drop vs. volumetric flow rate graph



b. $\Delta p/LU$ vs. flow rate graph

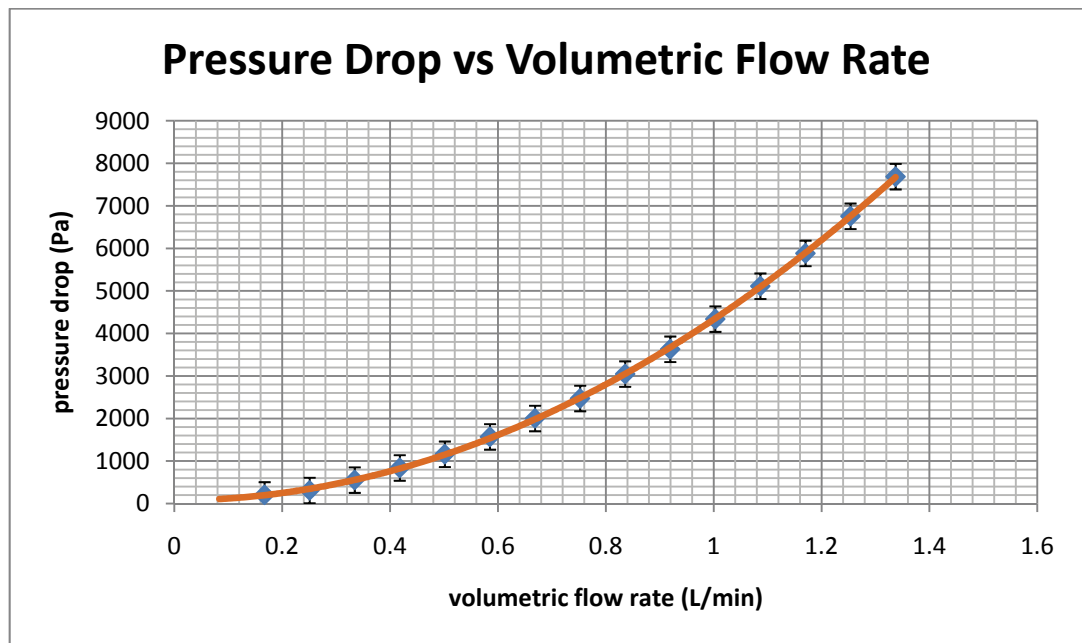
Figure 5.21 Metal foam 20-02 pressure drop result

Table 5.12 Metal foam 20-02 pressure drop results

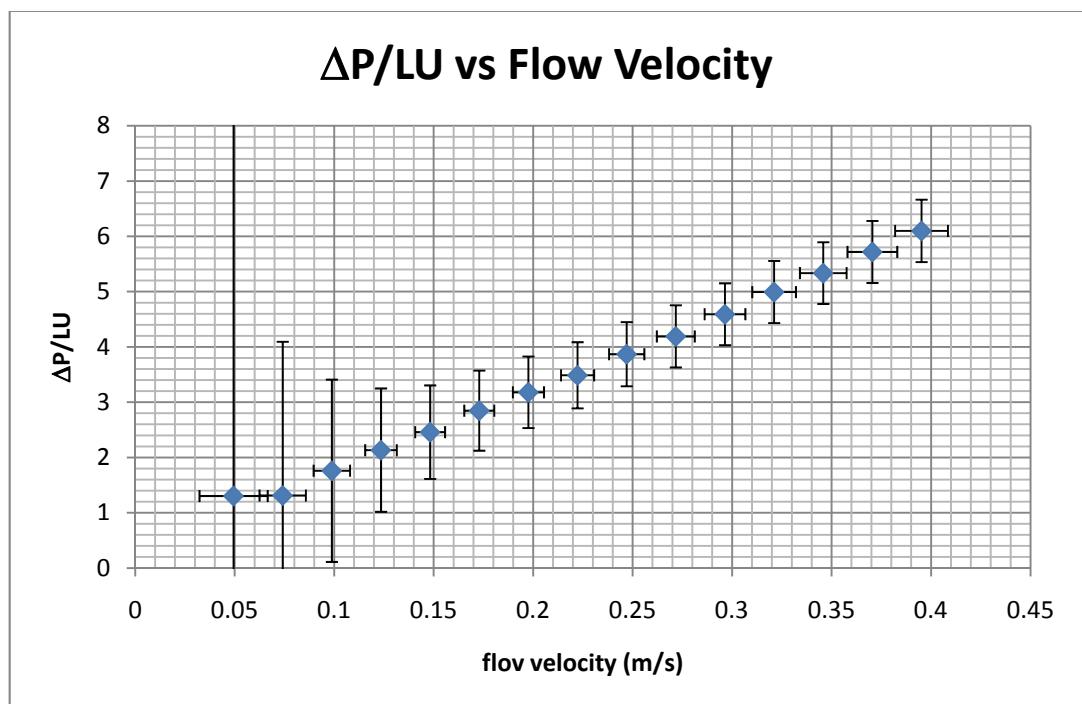
Mass flow rate (kg/h)	Measured values			Uncertainties	
	Re_k	Δp (Pa)	$\Delta p/L$ (Pa/m)	Δp (Pa)	$\Delta p/L$
10	5.7	116.2	3643	74.5	74.6
15	8.6	206.5	6473	41.9	42.0
20	11.5	433.1	13577	20.0	20.1
25	14.4	639	20031	13.6	13.7
30	17.3	885	27743	9.8	10.0
35	20.1	1149.6	36038	7.5	7.8
40	23.0	1493.7	46824	5.8	6.1
45	25.9	1784.5	55940	4.9	5.2
50	28.8	2046.1	64141	4.2	4.6
55	31.3	2411.9	75608	3.6	4.0
60	34.5	2828.9	88680	3.1	3.6
65	37.4	3246.1	101759	2.7	3.2
70	40.3	3655.6	114596	2.4	3.0
75	43.2	4007.3	125621	2.2	2.8

Table 5.13 Metal foam 20-03 pressure drop results

Mass flow rate (kg/h)	Measured values			Uncertainties	
	Re_k	Δp (Pa)	$\Delta p/L$ (Pa/m)	Δp (Pa)	$\Delta p/L$
10	10.6	205.3	6436	42.2	42.2
15	15.9	310	9178	27.9	28.0
20	21.2	554.4	17379	15.6	15.73
25	26.5	839.9	26329	10.3	10.47
30	31.8	1161.8	36420	7.5	7.7
35	37.1	1569.9	49213	5.5	5.8
40	42.4	2003.6	62809	4.3	4.7
45	47.7	2472.5	77508	3.5	3.9
50	53.0	3047.7	95539	2.8	3.4
55	58.3	3630.9	113821	2.4	3.0
60	63.6	4339.9	136047	2.0	2.7
65	69.0	5113.2	160288	1.7	2.5
70	74.2	5884.1	184455	1.5	2.3
75	79.5	6757	211818	1.3	2.2
80	84.8	767.8	240997	1.1	2.1

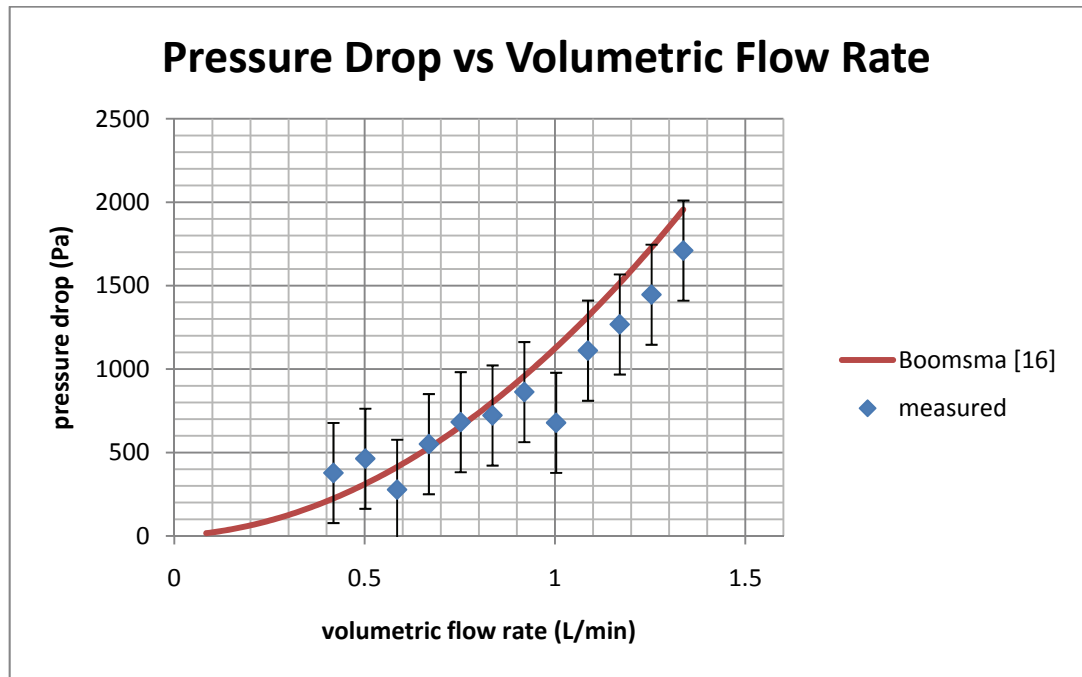


a. Pressure drop vs. volumetric flow rate graph

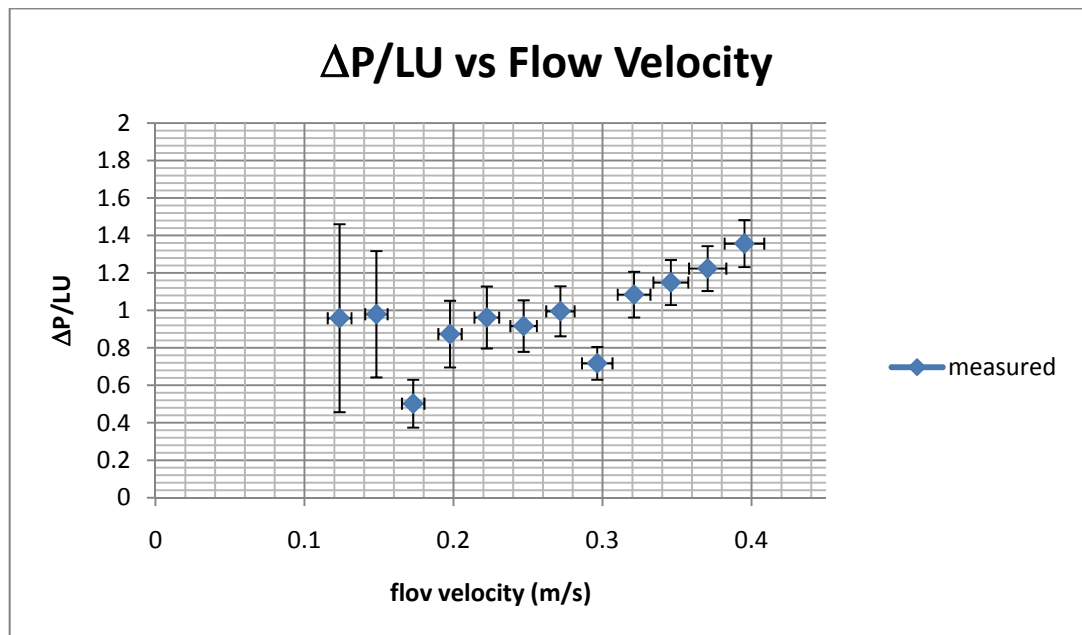


b. $\Delta p/LU$ vs. flow rate graph

Figure 5.22 Metal foam 20-03 pressure drop result



a. Pressure drop vs. volumetric flow rate comparison graph

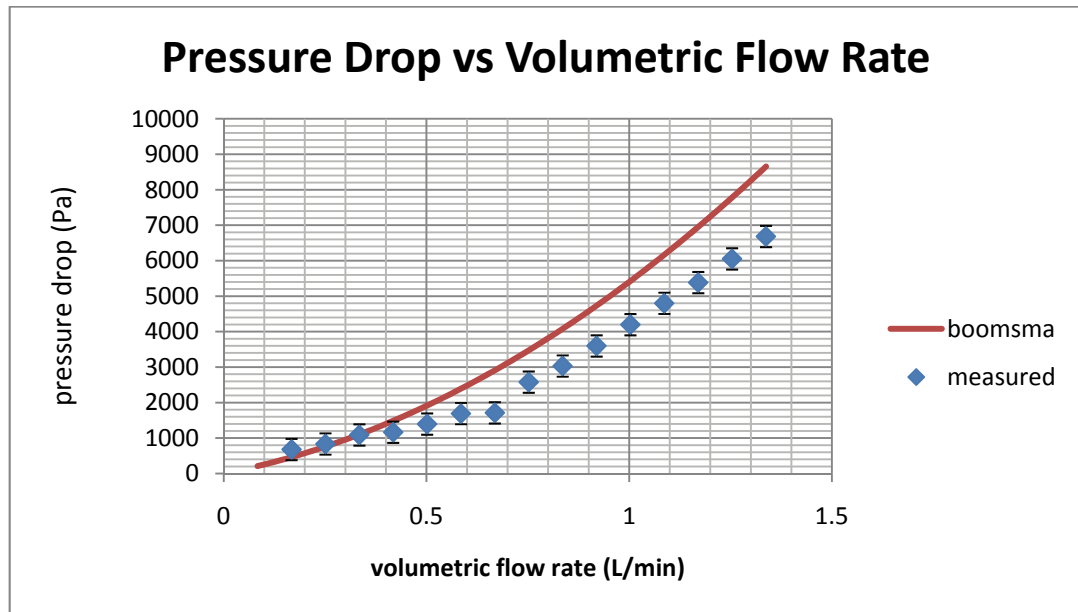


b. $\Delta p/LU$ vs. flow rate comparison graph

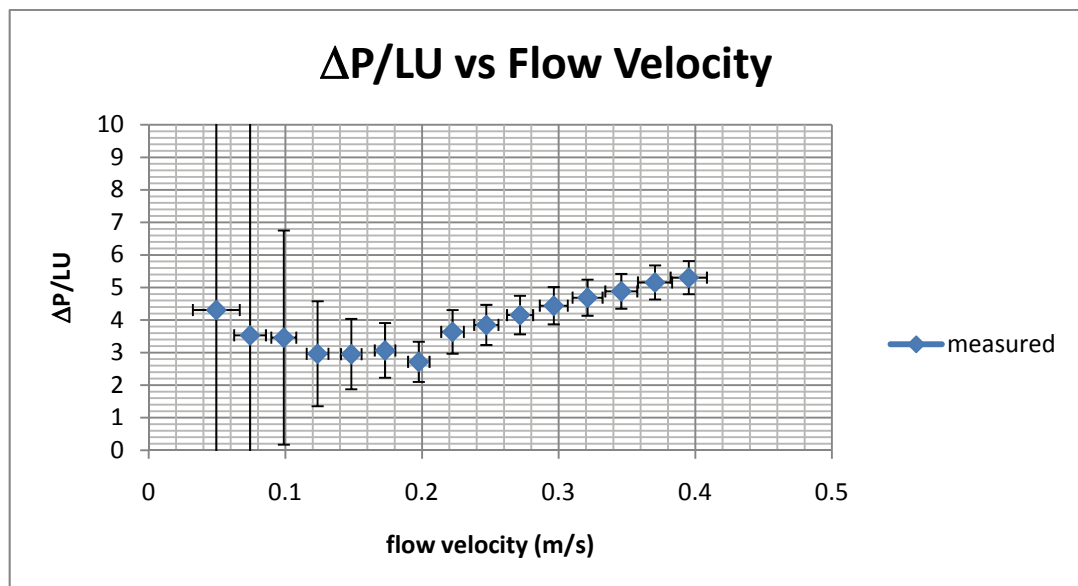
Figure 5.23 Metal foam 40-01 pressure drop result

Table 5.14 Metal foam 40-01 pressure drop results

	Calculated by using Boomsma's K and C for 40PPI		Measured values			Uncertainties	
Mass flow rate (kg/h)	Δp (Pa)	$\Delta p/L$ (Pa/m)	Re_k	Δp (Pa)	$\Delta p/L$ (Pa/m)	Δp (Pa)	$\Delta p/L$
25	223.6	7010	22.5	377.7	11840	22.9	23.0
30	310.5	9733	27.0	463.2	14520	18.7	18.8
35	411.4	12897	31.5	277	8683	31.3	31.3
40	526.4	16500	36.0	550.7	17263	15.7	15.8
45	655.4	20544	10.5	682.4	21392	12.7	12.8
50	798.4	25029	45.0	722.4	22646	12.0	12.1
55	955.5	29953	49.4	862.7	27044	10.0	10.2
60	1126.7	35318	53.9	678.4	21266	12.8	12.9
65	1311.8	41124	58.4	1110.8	34821	7.8	8.0
70	1511.1	47369	62.9	1267.8	39743	6.8	7.1
75	1724.4	54055	67.4	1446.3	45339	6.0	6.3
80	1951.7	61182	71.9	1710.6	53624	5.1	5.4



a. Pressure drop vs. volumetric flow rate comparison graph



b. $\Delta p/LU$ vs. flow rate comparison graph

Figure 5.24 Metal foam 40-02 pressure drop result

Table 5.15 Metal foam 40-02 pressure drop results

Mass flow rate (kg/h)	Calculated by using Boomsma's K and C for 92-2		Measured values			Uncertainties	
	Δp (Pa)	$\Delta p/L$ (Pa/m)	Re_κ	Δp (Pa)	$\Delta p/L$ (Pa/m)	Δp (Pa)	$\Delta p/L$
10	461.5	14466	4.1	678.9	21282	12.8	12.9
15	758.7	23783	6.2	834.6	26163	10.4	10.5
20	1100.2	34488	8.3	1090.6	34188	7.9	8.1
25	1486.0	46583	10.3	1167.8	36608	7.4	7.6
30	1916.1	60067	12.4	1396.3	43771	6.2	6.5
35	2390.6	74940	14.5	1691.8	53034	5.1	5.4
40	2909.4	91203	16.5	1712.3	53677	5.1	5.4
45	3472.4	108853	18.6	2578.8	80840	3.4	3.8
50	4079.8	127894	20.6	3033.1	95081	2.9	3.4
55	4731.5	148324	22.7	3599.8	112846	2.4	3.0
60	5427.5	170142	24.8	4197.8	131593	2.1	2.7
65	6167.9	193350	26.8	4800.2	150477	1.8	2.6
70	6952.5	217946	28.9	5384.7	168799	1.6	2.4
75	7781.5	243933	30.1	6052.2	189724	1.4	2.3
80	8654.7	271308	33.0	6685.1	209564	1.3	2.2

As seen in Figure 5.15a, quadratic relationship especially for higher flow velocities is obvious. For lower flow rates, Darcian flow becomes more dominant and pressure drop is linear. Boomsma also compressed aluminum metal foam of %92 porosity by three times. Although same compression factors were applied, final porosities are different and testing flow velocity range is not the same. But in spite of these facts similar results were observed.

Permeability values given in Table 5.6 are expected to decrease with increasing compression factor. To have more healthy results, denser pressure measurements should be taken especially at lower flow velocities.

Reynolds number based on permeability was given in Chapter 4 as:

$$Re = \frac{\rho U K^{1/2}}{\mu} \quad (5.29)$$

Table 5.16 Foam 40-03 pressure drop results

Mass flow rate (kg/h)	Calculated by using Boomsma's K and C for 92-3		Measured values			Uncertainties	
	Δp (Pa)	$\Delta p/L$ (Pa/m)	Re_K	Δp (Pa)	$\Delta p/L$ (Pa/m)	Δp (Pa)	$\Delta p/L$
10	733,4	22992	7,0	514,3	16122	16,8	16,9
15	1204,1	37745	0,6	676,1	21194	12,8	12,9
20	1743,9	54669	14,1	917,7	28768	9,4	9,6
25	2353,1	73765	17,6	1302,6	40834	6,6	6,9
30	3031,5	95031	21,1	1809,7	56730	4,8	5,1
35	3779,2	118469	24,6	2375,9	74480	3,6	4,1
40	4596,1	144074	28,1	3023,4	94777	2,9	3,4
45	5482,3	171859	31,2	3781,4	118539	2,3	2,9
50	6437,7	201810	35,2	4842,5	151803	1,8	2,5
55	7462,5	233933	38,7	5777,8	181122	1,5	2,4
60	8556,4	268227	42,2	6775,3	212391	1,3	2,2
65	9719,7	304692	45,7	7838,3	245715	1,1	2,1
70	10952,2	343329	49,3	9100,4	285279	1,0	2,0
75	12120,7	379958	52,4	10378,8	325354	0,8	2,0

Graph in Figure 5.15b was plotted to observe transition from linear Darcy regime closely. Transition appears nearly at 0.1 m/s for foam 40-03. At this velocity metal foam leaves the linear Darcy regime and enters the form dominated pressure drop regime characterized by the addition of the quadratic term to the linear Darcy Law [16].

5.9 Metal foam heat transfer results and discussion

Nusselt number values for metal foam heat sinks were also calculated. Nusselt number was given in Chapter 4 as follows:

$$Nu = \frac{qd_{hyd}}{k_f A_{con}(T_{pl} - T_{c,in})} = \frac{\dot{m} c d_{hyd}(T_{c,out} - T_{c,in})}{k_f A_{con}(T_{pl} - T_{c,in})} \quad (5.30)$$

Since hydraulic diameter of a porous medium cannot be calculated open channel dimensions were considered. As a result, hydraulic diameters of open cell metal foams are higher than microchannels. Thus, high values of Nusselt numbers were expected.

Nusselt number values versus volumetric flow rate of metal foam heat sinks are presented in Figure 5.16. Also Nusselt number and thermal resistances of metal foams tested are presented in Table 5.8. Compression factor was observed to be more dominant compared to porosity counts at Nusselt number values. Three times compressed foam 10-03, 20-03 and 40-03 have higher Nusselt numbers. Two times compressed foam 40-02 has comparable thermal performance with foam 10-03. It can be said that compression greatly enhances heat transfer performance of metal foams and they can be used as high performance heat sinks only by compression.

Table 5.17 Nusselt number and thermal resistances of metal foams

	Mass flow rate (kg/h)	Volumetric flow rate (l/min)	Nusselt Number	Thermal Resistance (°C/W)	Nu Uncertainty (%)	Thermal Resistance Uncertainty (%)
Foam 10-01	15	0,25	25,03	1,35	10,90	3,08
	30	0,50	26,01	1,30	10,78	4,99
	45	0,75	26,62	1,27	10,76	7,48
	55	0,92	27,19	1,25	10,75	9,11
Foam 10-03	15	0,25	40,19	0,84	10,95	3,23
	30	0,50	42,20	0,80	10,83	5,30
	45	0,75	45,93	0,74	10,82	7,44
	56	0,94	46,85	0,72	10,81	9,13
Foam 20-01	15	0,25	25,79	1,31	10,91	3,26
	30	0,50	27,44	1,23	10,79	5,37
	45	0,75	27,43	1,23	10,77	8,22
	60	1,00	27,83	1,22	10,76	11,07
Foam 20-02	15	0,25	36,41	0,93	10,94	3,24
	30	0,50	39,73	0,85	10,82	5,17
	45	0,75	42,18	0,80	10,80	7,43
Foam 20-03	15	0,25	46,20	0,73	10,97	3,22
	30	0,50	51,65	0,66	10,87	5,10
	45	0,77	54,58	0,62	10,86	7,72
Foam 40-01	15	0,25	28,83	1,17	10,91	3,12
	30	0,50	31,59	1,07	10,79	4,95
	45	0,75	31,33	1,08	10,77	7,69
	60	1,00	33,35	1,01	10,77	9,88
Foam 40-02	15	0,25	39,99	0,85	10,94	3,14
	30	0,50	43,89	0,77	10,82	4,79
	45	0,75	46,03	0,74	10,80	6,89
Foam 40-03	15	0,25	47,38	0,71	10,96	3,13
	30	0,50	54,46	0,62	10,87	4,99
	45	0,75	59,43	0,57	10,86	7,07
	54	0,90	62,73	0,54	10,89	9,04

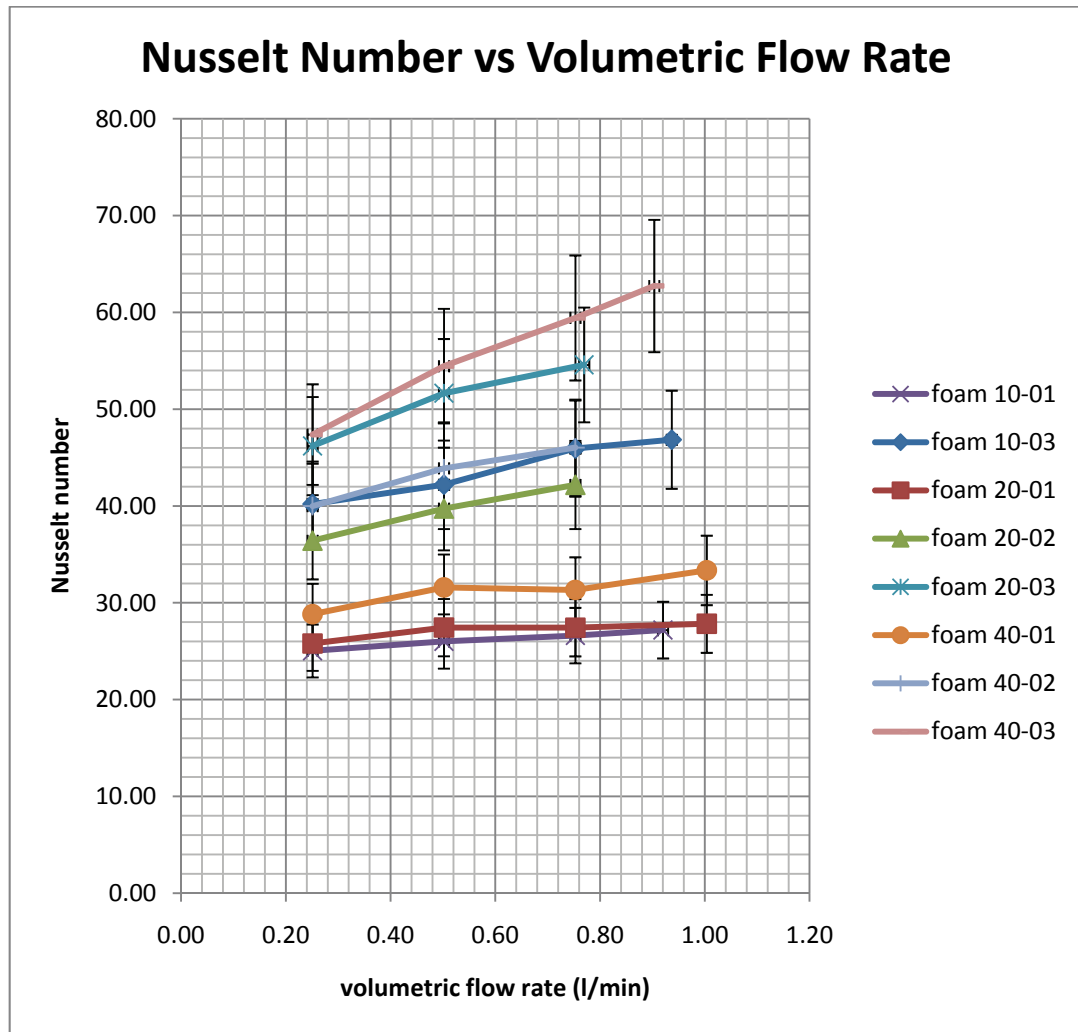
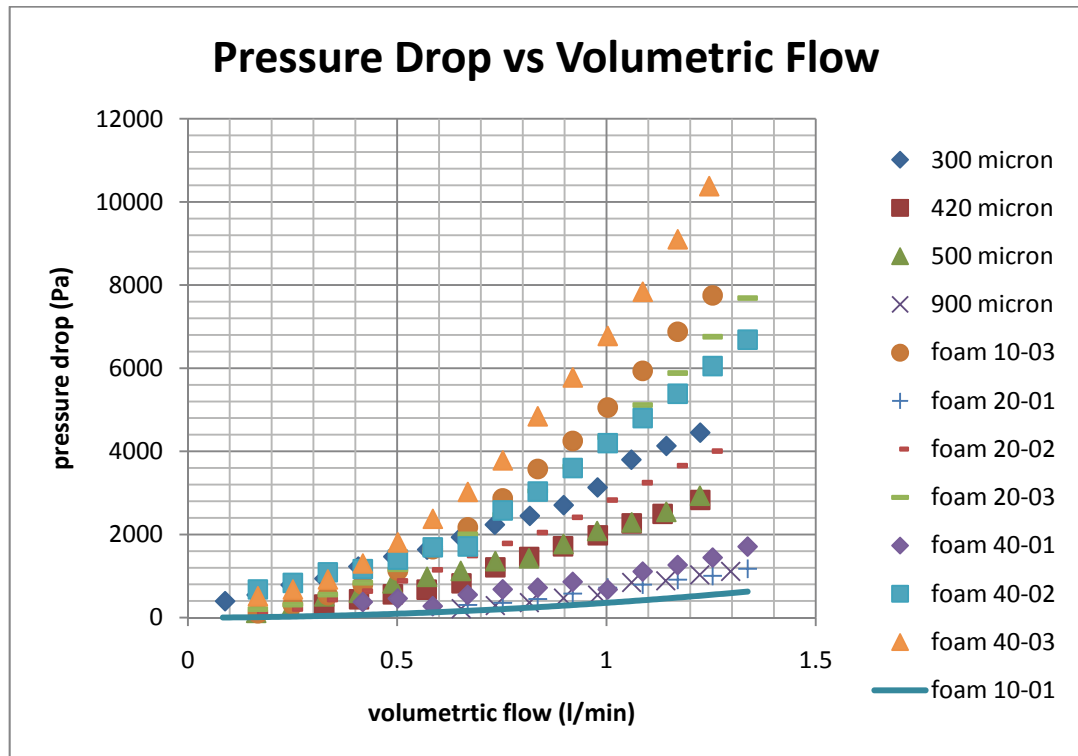


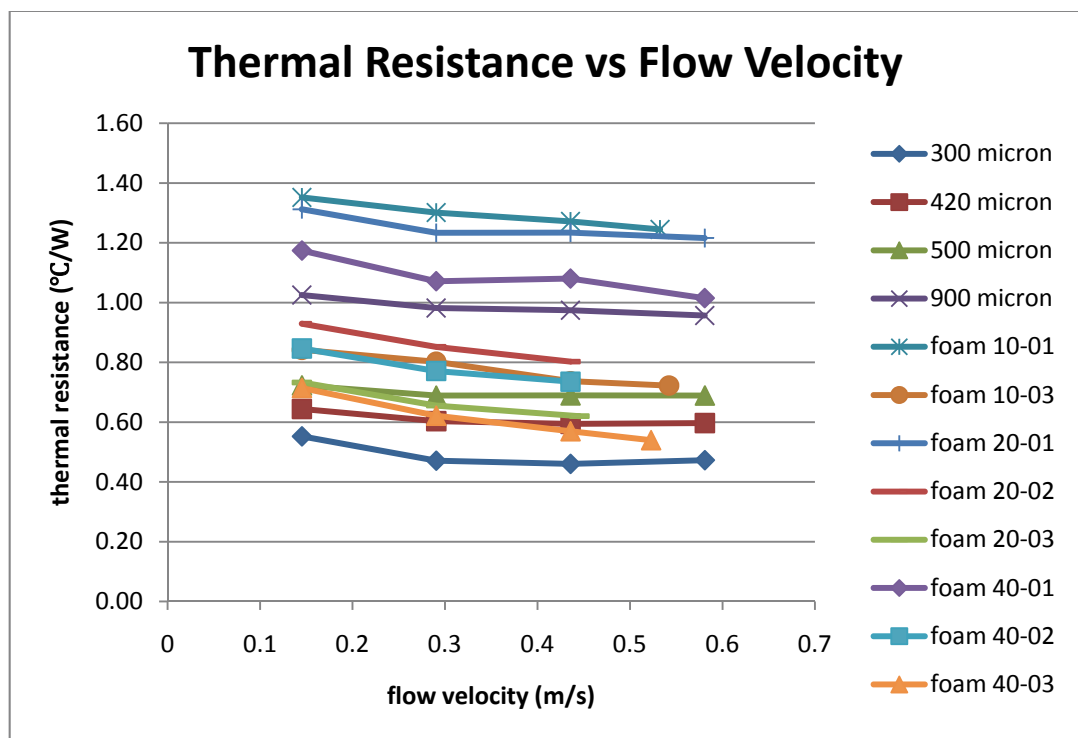
Figure 5.25 Nusselt number versus volumetric flow rate of metal foam samples

5.10 Heat exchanger performance comparison

The main purpose of this thesis is to see heat exchanger performances and make a comparison between metal foam and microchannel heat sinks since they are both mentioned as compact high performance heat sinks in previous studies. To make performance comparison, three graphs that includes all heat sinks produced for this study are presented in Figures 5.17a, 5.17b and 5.18.



a. Pressure drop vs. volumetric flow rate graph of heat sinks



b. Thermal resistance vs. flow velocity graph of heat sinks

Figure 5.26 Pressure drop and thermal resistance comparison graphs

In Figure 5.17a, variation of pressure drop with volumetric flow rate is presented. A direct observation is that, especially compressed metal foams have higher pressure drop values than microchannels and uncompressed metal foams have lowest values.

When thermal resistance behaviors of heat sinks for various flow velocities is considered, it can be seen in Figure 5.17b that 300 micron channel has lowest thermal resistance and metal foams become comparable when 300 micron channel was not included in comparison. It is also obvious that uncompressed foams have lowest performance since they have highest thermal resistance values.

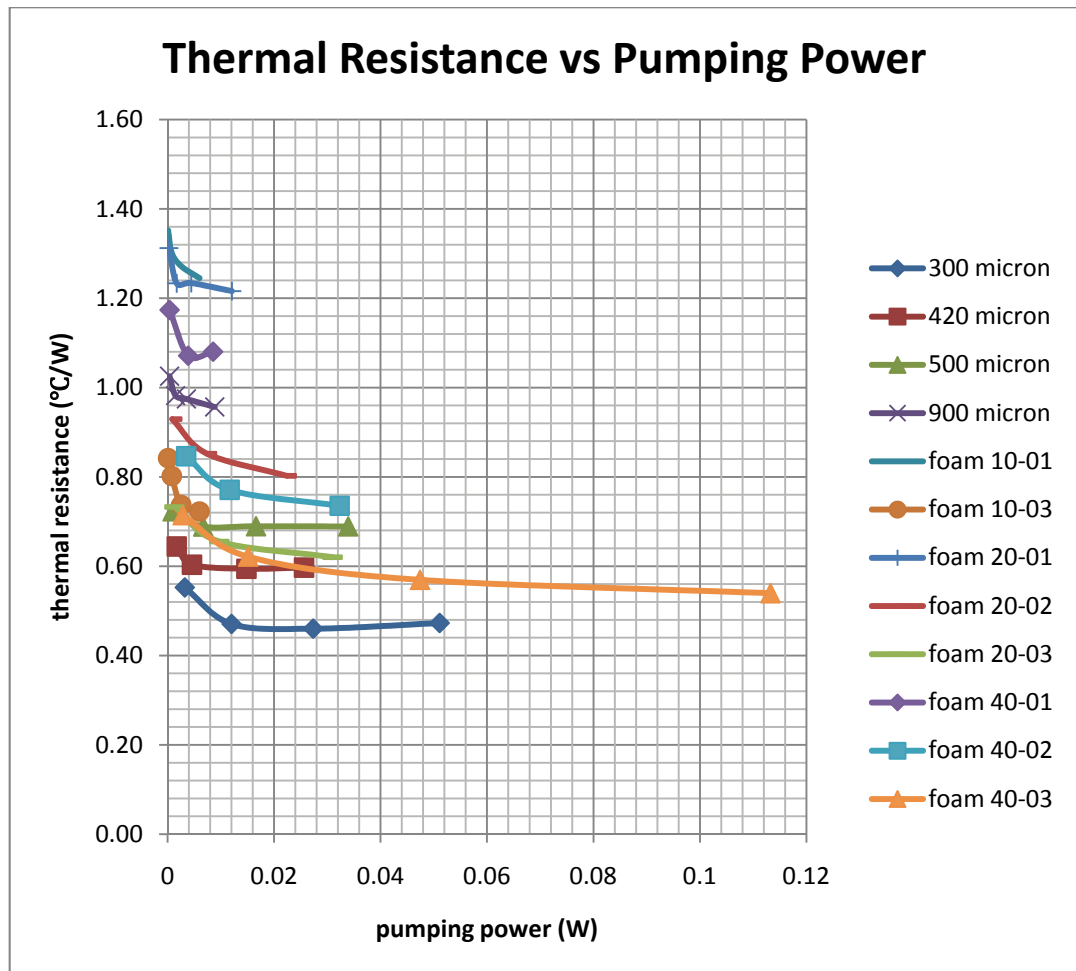


Figure 5.27 Pressure drop and thermal resistance comparison graphs

When designing a heat exchanger, heat transfer performance of heat exchanger should be weighed against power required to flow coolant through. Nusselt number and Colburn factor cannot be used as thermal performance comparison tool since they have different range of values. Another most common means to measure the heat convection effectiveness is the thermal resistance.

In a good design, both thermal resistance and pumping power are expected to be low. Figure 5.18 have shown that 300 micron and 420 micron channel have highest heat exchanger performances. For higher pumping powers, thermal resistance of foam 40-03 becomes comparable to 420 micron channel. One can observe that as pumping power increases thermal resistance of microchannels have a tendency to become constant, while thermal resistance of metal foams continues to decrease steadily. This means that, for higher pumping powers thermal resistance versus pumping power curves of foam 40-03 and 300 micron channel intersects. This behavior was also observed by Zhang et al. [58]. However, employing a heat sink at higher pumping power is not feasible.

CHAPTER 6

CONCLUSIONS

This investigation aimed to compare thermal and fluid characteristics of metal foam and microchannel heat sinks. Since high performance heat sinks are required in R/T modules of phased-array antenna, this study will guide future designs and help the reader to understand fluid flow and forced heat convection both in microchannels and metal foams.

During the study, both microchannel and metal foam heat sinks were produced and experimentally investigated. To have a variety, four different microchannels and 8 different metal foams were tested. Some physical properties of all tested heat sinks were given also in this chapter in Table 6.1.

Pressure drop behavior of microchannels can be said to be in agreement with conventional theory. As expected, Nusselt number values were lower than predicted by classical theory since heated area was smaller than finned area.

Both uncompressed and compressed foams were tested. 5 metal foams tested in this study had similar physical properties (same compression factor and initial porosities with different final porosities) to foams tested in Boomsma's work. Good agreement was observed although porosities and testing ranges differed slightly.

Table 6.1 Physical properties of tested heat sinks

Specimen name	Channel width	Channel height	Hydraulic diameter	Aspect ratio	Number of channels
300 micron	300 μm	4 mm	558 μm	13,33	24
420 micron	420 μm	4 mm	760 μm	9,52	20
500 micron	500 μm	4 mm	888 μm	8,00	18
900 micron	900 μm	4 mm	1469 μm	4,44	12
Physical Properties of Uncompressed Foam					
Specimen name	Foam	Pore diameter (mm)	Specific surface Area (m ² /m ³)	Measured porosity (%)	
foam 10-01	10 ppi	6.9	820	92.1	
foam 20-01	20 ppi	3.6	1700	92.0	
foam 40-01	40 ppi	2.3	2700	92.8	
Physical Properties of Compressed Foam					
Specimen name	Initial uncompressed	Compression factor (<i>M</i>)	Expected porosity	Measured porosity (%)	
foam 10-03	10 ppi	3	76	81,0	
foam 20-02	20 ppi	2	84	86,6	
foam 20-03	20 ppi	3	76	79,1	
foam 40-02	40 ppi	2	84	87,1	
foam 40-03	40 ppi	3	76	80,1	

Heat exchanger performances of metal foam and microchannels are directly compared. As a result, it has been observed that 300 micron channel has best performance and 420 micron channel has comparable characteristics with metal foam 40-03. Uncompressed metal foams did not have a comparable performance. Hence it has been concluded that if they will be employed as high performance heat sinks, metal foams should be compressed. Compressing more than a factor of 3 enhances the performance and actually, there should be an optimum

compression factor when both pressure drop and forced convection heat transfer performance are considered. Boomsma [16] reported a compression factor of 4 for 95% porosity aluminum open cell metal foam and 6 for 92% porosity metal foam in his study. It means that optimum compression factor also varies with initial porosity values.

Besides thermal and hydraulic performances of heat exchangers, manufacturability and production cost plays an important role when comparing heat sinks with similar performances. Although 300 micron channel has higher heat exchanger performance, production cost is dramatically higher than cost of metal foam heat sink production. Additionally, when serial production is considered, since every channel should be machined on metal base plate via wire EDM or micro sawing, production time is also higher for microchannels. In that sense, metal foams become comparable to microchannels if pressure drops are acceptable.

Using metal foams as heat exchangers might be practical however additional experimental and numerical studies with wider range of porosities and compression factors are required. As mentioned in Chapter 3 heat exchanger performance of metal foam heat sinks is scalable. This means that compressing metal foams more, obviously will result in a better heat transfer performance while pressure drop will become higher. So, there should be an optimum compression factor as also concluded by Boomsma [16]. More numerical study especially for compressed metal foams should be conducted since for every design of a metal foam heat exchanger, establishing an experimental setup is time consuming and expensive.

During this study important knowledge concerning fluid flow and heat transfer through microchannels and open cell aluminum metal foams and their application as heat exchangers has been gained. Also, building an experimental setup to obtain needed information has been learned and during experimental study numerous problems have been overcome. Some suggestions regarding

experimental study either for metal foams or microchannels can be made as follows:

SUGGESTIONS FOR FUTURE WORK:

- Using differential pressure transducer instead of vacuum pressure transducer with narrower measurement ranges solves leveling problem of pressure transducer and doubling problem in error.
- Employing test specimens with special inlet and exit manifold design that causes less minor losses would be beneficial to directly observe pressure drop through observed channels or foam.
- Producing bigger microchannel heat sink that has more channels and metal foam heat sink that has a wider channel for experimentation to observe differences in temperature and pressure drop clearly. Doing that will result in more pressure drop. Therefore uncertainties can become less important when compared to measure pressure drop values.
- Using higher heater capacity with bigger heat conduction area in order to see larger differences between exit and inlet cooling fluid temperature and to reduce the importance of error due to temperature sensors.
- Designing heat sink assembly just for experimentation that reduces thermal transition interfaces.

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APPENDIX A

TECHNICAL DRAWINGS OF TEST SPECIMENS

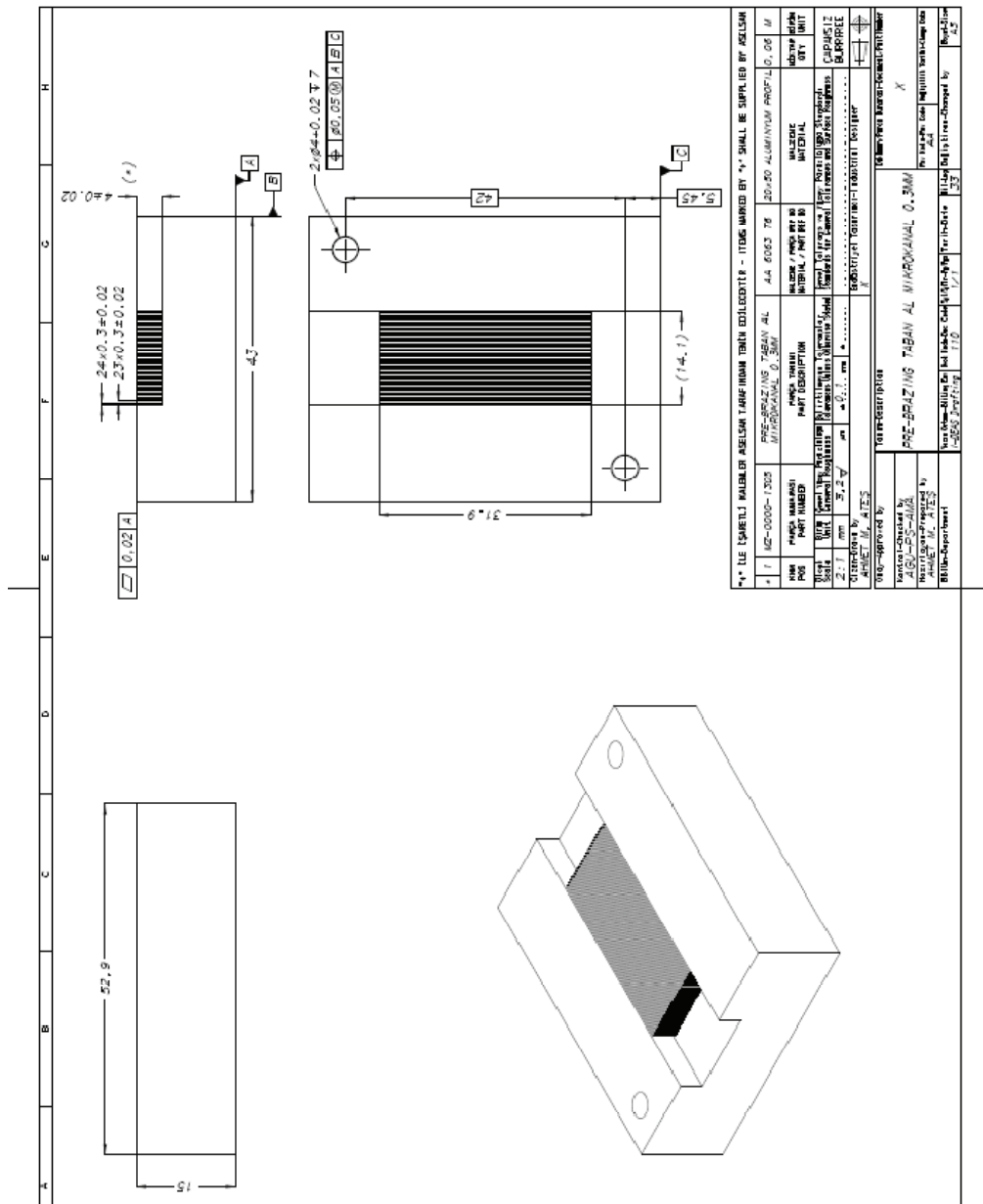
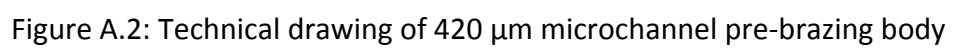
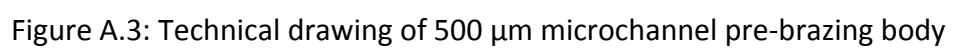
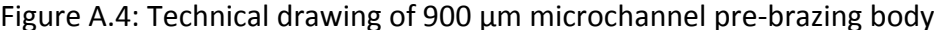


Figure A.1: Technical drawing of 300 μm microchannel pre-brazing body







APPENDIX B

SAMPLE UNCERTAINTY CALCULATION

Sample uncertainty calculation of 300 micron channel thermal resistance at a flow rate of 1lpm can be calculated as below.

Uncertainty formulation was derived in chapter 5 as follows:

$$\frac{U_{R_{th}}}{R_{th}} = \sqrt{\left(\frac{U_{T_{pl}}}{T_{pl}}\right)^2 + \left(\frac{-U_{T_{c,in}}}{T_{c,in}}\right)^2 + \left(\frac{-U_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{-U_c}{c}\right)^2 + \left(\frac{-U_{T_{c,out}}}{T_{c,out} - T_{c,in}}\right)^2 + \left(\frac{U_{T_{c,in}}}{T_{c,out} - T_{c,in}}\right)^2}$$

Uncertainty of value that can be measured with an error $X + e$ and $X - e$ was defined as:

$$U = e/\sqrt{3}$$

Therefore, uncertainties of plate temperature, inlet and exit coolant temperature, specific heat and mass flow rate can be conducted.

Error of J-type thermocouple that was employed was $\pm 1^\circ\text{C}$.

$$U_{T_{pl}} = 1/\sqrt{3}$$

Error of RTD temperature sensors that was used to measure coolant temperatures at inlet and exit was $\pm 0.2^\circ\text{C}$.

$$U_{T_{c,in}} = U_{T_{c,out}} = 0.2/\sqrt{3}$$

Error in measured mass flow rate was $\pm 0,5\text{kg/h}$.

$$U_{\dot{m}} = 0.5/\sqrt{3}$$

Error of specific heat due to change in temperature of water can be found subtracting the specific heat values of water at related temperatures: (4072.1-4070.2).

$$U_c = (4072.1 - 4070.2)/\sqrt{3}$$

Measured values at 1lpm are:

$$T_{pl} = 53.62^\circ\text{C}$$

$$T_{c,in} = 26.35^\circ\text{C}$$

$$T_{c,out} = 27.20^\circ\text{C}$$

$$\dot{m} = 60\text{kg/h}$$

$$c = 4071.6\text{J/kg.K}$$

$$\frac{U_{R_{th}}}{R_{th}}$$

$$= \sqrt{\left(\frac{\left(\frac{1}{\sqrt{3}}\right)^2}{53.62}\right)^2 + \left(\frac{\left(\frac{-0.2}{\sqrt{3}}\right)^2}{26.35}\right)^2 + \left(\frac{\left(\frac{-0.5}{\sqrt{3}}\right)^2}{60}\right)^2 + \left(\frac{\left(\frac{-(4072.1 - 4070.2)}{\sqrt{3}}\right)^2}{4071.6}\right)^2 + \left(\frac{\left(\frac{-0.2}{\sqrt{3}}\right)^2}{27.20 - 26.35}\right)^2 + \left(\frac{\left(\frac{0.2}{\sqrt{3}}\right)^2}{27.20 - 26.35}\right)^2}$$

$$= 0.096805$$

APPENDIX C

SAMPLE ANALYTICAL PRESSURE DROP AND NUSSELT NUMBER CALCULATION FOR MICROCHANNELS

Sample analytical pressure drop and Nusselt number calculation of 300 micron at a flow rate of 1lpm was presented at this appendix.

Pressure drop formulation for developing laminar flow was given by equation 4.29.

$$\Delta p = \frac{2f_{app}\rho u_m^2 L}{d_{hyd}} + K_c \frac{\rho u_m^2}{2} + K_e \frac{\rho u_m^2}{2} + \left(\frac{A_c}{A_p}\right)^2 (2K_{90}) \frac{\rho u_m^2}{2}$$

Apparent friction factor was given as:

$$K(x) = (f_{app} - f) \frac{4x}{d_{hyd}}$$

For the channel length that is larger than the hydrodynamic entrance length, $L > L_h$, incremental pressure defect $K(x)$ becomes a constant value known as Hagenbach's factor $K(\infty)$. For rectangular channels it was given as:

$$K(\infty) = 0.6796 + 1.2197\alpha + 3.3089\alpha^2 - 9.5921\alpha^3 + 8.9089\alpha^4 - 2.9959\alpha^5$$

Entrance length of the hydrodynamic developing region, Reynolds number, flow velocity, Poiseuille number, hydraulic diameter, and channel aspect ratio was defined as:

$$L_H = d_{hyd} 0.05 Re$$

$$Re = \rho u_m \frac{d_{hyd}}{\mu}$$

$$u = \dot{m} / \rho A_{cs}$$

$$Po = f Re = 24(1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2357\alpha^5)$$

$$d_{hyd} = \frac{4ab}{2(a+b)} = \frac{2ab}{(a+b)}$$

$$\alpha = a/b$$

$$a = 0.0003m \text{ (channel width)}$$

$$b = 0.004m \text{ (channel height)}$$

$$\rho = 997.1kg/m^3 \text{ (density of water at } 25^\circ C)$$

$$\mu = 0.0008684 \frac{kg}{ms} \text{ (dynamic viscosity of water at } 25^\circ C)$$

$$L = 0.0319 m \text{ (channel length)}$$

$$A_{cs} = 24 * 0.0003 * 0.004 = 0.0000288 m^2 \text{ (total area of channels, 24 is number of total channels)}$$

$$u_m = \frac{\frac{60kg}{h * 3600s/h}}{\frac{997.1kg}{m^3} * 0.0000288m^2} = 0.5804m/s$$

$$d_{hyd} = \frac{2*0.0003*0.004}{(0.0003+0.004)} = 0.00055814m$$

$$Re = 997.1 * 0.5804 * \frac{0.00055814}{0.0008684} = 372$$

$$\begin{aligned} Po = fRe &= 24 \left(1 - 1.3553 * \left(\frac{0.0003}{0.004} \right) + 1.9467 \left(\frac{0.0003}{0.004} \right)^2 \right. \\ &\quad \left. - 1.7012 \left(\frac{0.0003}{0.004} \right)^3 + 0.9564 \left(\frac{0.0003}{0.004} \right)^4 - 0.2357 \left(\frac{0.0003}{0.004} \right)^5 \right) \\ &= 21.80675 \end{aligned}$$

$$f = \frac{21.80675}{372} = 0.0586$$

$L_H = 0.00055814 * 0.05 * 372 = 0.0104m < 0.0319$ Therefore, Hagenbach's factor should be used.

$$\begin{aligned} K(\infty) &= 0.6796 + 1.2197 \left(\frac{0.0003}{0.004} \right) + 3.3089 \left(\frac{0.0003}{0.004} \right)^2 - 9.5921 \left(\frac{0.0003}{0.004} \right)^3 \\ &\quad + 8.9089 \left(\frac{0.0003}{0.004} \right)^4 - 2.9959 \left(\frac{0.0003}{0.004} \right)^5 = 0.78592 \end{aligned}$$

$$f_{app} = \frac{K(\infty)d_{hyd}}{4L} + f = \frac{0.78592 * 0.00055814}{4 * 0.0319} + 0.0586 = 0.06206$$

Two sudden expansion and sudden contraction should be included in pressure drop calculations. Sudden expansion and contraction after and before microchannels and after and before heat sink body.

Sudden expansion and contraction coefficients can be read from figure 4.4.

$$K_{c1} = 0.7$$

$$K_{c2} = 0.98$$

$$K_{e1} = 0.05$$

$$K_{e2} = 0.8$$

$$K_{90} = 1.2 \text{ (Recommended value by Philips [17])}$$

$$A_c = 0.004 * 0.0141 = 5.64 * 10^{-5} m^2 \text{ (channel area)}$$

$$A_p = 0.0000785 m^2 \text{ (inlet and outlet plenum area before bending)}$$

$$\begin{aligned} \Delta p &= \frac{2 * 0.06206 * 997.1 * 0.5804^2 * 0.0319}{0.00055814} + (0.7 + 0.98) \frac{997.1 * 0.5804^2}{2} \\ &\quad + (0.05 + 0.8) \frac{997.1 * 0.5804^2}{2} + (0.05 + 0.8) \frac{997.1 * 0.5804^2}{2} \\ &= 3016.4 \text{ Pa} \end{aligned}$$

For Nusselt number calculations Table 4.1 and Table 4.2 can be used. For intermediate values of x^* and α Kandlikar [85] supplied curve-fitting equations as given below:

Table D.1: Curve-fit equations for Tables 4.1, and 4.2

Equations	a	b	c	d	e	f
1	8.2321	2.0263	1.2771	0.29805	2.2389	0.0065322
2	8.2313	1.9349	2.295	0.92381	7.928	0.0033937
3	36.736	2254	17559	66172	555480	1212.6
4	30.354	1875.4	13842	154970	783440	-8015.1
5	31.297	2131.3	14867	144550	622440	-13297
6	28.315	3049	27038	472520	1783300	-35714
7	6.7702	3.1702	0.4187	2.1555	2.76×10^{-6}	NA
8	9.1319	3.7531	0.48222	2.5622	5.16×10^{-6}	NA

Equations 1-6: $y = \frac{a+cx+ex^2}{1+bx+dx^2+fx^3}$

Equations 7-8: $y = a + bx + c(\ln x)^2 + d\ln x + ex^{-1,5}$

Fully developed Nusselt number for three-side heating, Table 4.2:

y-variable - Nusselt number for three-sided heating

x-variable - Equation 1: α_c

Fully developed Nusselt number for four-side heating, Table 4.2:

y-variable - Nusselt number for four-sided heating

x-variable - Equation 2: α_c

Thermal entry region Nusselt numbers for four-sided heating, Table 4.1:

y-variable - Nusselt number for four-sided heating in the entry region

x-variable - Equation 3: $\alpha_c = 0.1$

Equation 4: $\alpha_c = 0.25$

Equation 5: $\alpha_c = 0.333$

Equation 6: $\alpha_c = 0.5$

Equation 7: $\alpha_c = 1.0$

Equation 8: $\alpha_c = 10$

x^* for Table 4.1 first column variable was given as:

$$x^* = \frac{L/d_{hyd}}{RePr}$$

$L = 0.0319 \text{ m}$ (channel length)

$d_{hyd} = 0.00055814 \text{ m}$ (hydraulic diameter)

$Re = 372$ (for 1lpm water flow at 27°C)

$Pr = 5.4936$ (Prandtl number at 27°C)

$$x^* = \frac{0.0319/0,00055814}{372 * 5.4936} = 0.026736$$

And aspect ratio is less than $\alpha < 0.1$ for that values four side heating values as given in table 4.1 can be used. Use equation 3 for intermediate value.

$$Nu = y = \frac{36.736 + 17559 * 0.026736 + 555480 * 0.026736^2}{1 + 2254 * 0.026736 + 154970 * 0.026736^2 - 8015.1 * 0.026736^3} \\ = 8.318$$