High Temperature Gas to Liquid Metal Foam and Wire Mesh Heat Exchangers

by:

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A thesis submitted in conformity with the requirements for the degree of Master of Applied Science
Department of Mechanical and Industrial Engineering
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Abstract

Metal foams and wire meshes are open cell structures with low weight and density, high permeability and high thermal conductivity which make them attractive for a wide range of industrial applications involving fluid flow and heat transfer. In this study, the effect of natural convection, radiation and heat transfer enhancement of metal foams and wire meshes of 10 and 40 PPI (pores per inch) heat exchangers were examined and compared for different heat exchanger orientation, coolant flow rate and atmosphere temperature.

Thermal spray coating processes were also used in development of a new class of high temperature stainless steel heat exchangers. Stainless steel wire mesh heat exchangers were prototyped by connecting the tube to the wire mesh using wire arc thermal spray coating. Thermal spray coating provided efficient connections between the wire mesh and the tubes’ outer surface, and has potential to replace expensive brazing or other metal connection techniques.
Acknowledgments

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Finally, I would like to thank my family and friends for always being so supportive and encouraging, and would like to dedicate this thesis to my parents, brother:

Masieh, Nasrin and Mojtaba

Thank you for everything,
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Chapter 1
Introduction

1.1 Introduction

Over 14.5 billion cubic feet of gas are burned and vented daily from oil and gas operations worldwide, the equivalent of 2.5 million barrels of oil each day [1]. Gas flares are used to eliminate waste gases such as methane, which are not feasible to use or transport. For combustion to be complete the air-fuel ratio (AFR) must be stoichiometric, meaning enough air is provided to completely burn all the fuel. The products of complete combustion (water vapor and carbon dioxide) are not nearly as detrimental to air quality and do not contribute as much to global warming as when hydrocarbon gases are vented. In theory the heat of combustion can be recovered from the combustion gases using heat exchangers for different commercial or industrial processes.

There have been advancements in increasing the efficiency of combustion and reaching stoichiometric AFR made by several companies around the world; however, few companies have tried to recover the heat of that combustion. Questor Technology Inc. (Calgary, Alberta) designs incinerator columns that can burn the waste gases cleanly with 99.99% efficiency, but as of yet there is no technology or recovery system to use that heat [1]. By positioning a heat exchanger (HEX) box on top of the hot gas stack, as shown on Figure 1.1, the heat of combustion can be captured. However, it is difficult to manufacture heat exchangers that can withstand high combustion temperatures and have a high enough efficiency to make them commercially viable. Waste gases exit flares at temperatures of over 1000°C, which exceeds the operation range of most high thermal conductivity materials such as copper and aluminum that are typically used to fabricate heat exchangers. New, high efficiency heat exchanger design can compensate for the low thermal conductivity of materials such as stainless steel and Inconel that can withstand high temperatures.
In this study different designs were investigated in order to enhance the efficiency of tube heat exchangers by using different porous metallic materials that acted as external fins, increasing the surface area for heat transfer and therefore the efficiency of the heat exchangers.

Figure 1.1: Gas to Liquid heat exchanger use to extract heat from the gas flare (a) Red arrows indicate the inlet and outlet hot gas of the gas flare and blue arrows show the direction of the coolant in and out of the system. (b) Coolant flow inside the heat exchanger.
1.2 Literature Review

1.2.1 Fundamentals of heat exchangers

Heat exchangers are designed to transfer heat between two fluids that are at different temperatures while keeping them separate from each other. Design of heat exchangers differs based on application but it usually involves convection in both fluids, conduction in the separation wall and in some cases radiation. The flow direction of the fluids can be parallel or counter flow. In the case of a concentric heat exchanger consisting of two concentric pipes of different diameter, as shown in Figure 1.2 (a) or in the case of this study crossflow as shown in Figure 1.2 (b).

Figure 1.2: Concentric tube (double-pipe) heat exchanger (a) Parallel flow. (b) Counterflow.
Compact heat exchangers are widely used due to their large heat transfer surface area to volume ratio, which is called area density. This large surface area is usually obtained by attaching fins to the wall separating both fluids, with the fluid with lower heat transfer coefficient flowing on the fin side of the wall. Investigators have been studying the performance of metal foams and wire meshes for several years as a candidate to replace fins. Brazing has been used for many years to connect fins to the heat exchanger surface. In this study we are investigating new methods to move from the traditional brazing to more cost effective method, and to enhance the connection between the porous structures; foams, wire meshes, etc. and the substrate in order to increase the heat transfer performance of the heat exchanger.

1.2.2 Introduction to Metal Foams

Researchers have studied metal foams extensively due to their excellent mechanical, electrical, acoustic and structural properties [31]. Their low density and weight, high specific strength and permeability and the ability to manufacture them out of high thermal conductive metal makes them suited for structural components and applications involving heat transfer and fluid flow such as heat exchangers, evaporators and burners [8, 32]. Open-cell metallic foam structure consist of interconnected struts, and are characterized by their porosity, pore density, pore size and fiber diameter, Figure 1.3 illustrates 40 pores per inch (PPI) (Figure 1.3 (a)) and 10 PPI pore density (Figure 1.3 (b)) open-cell copper foams.

As pore size increases from 40 PPI (Figure 1.3 (a)) to 10 PPI (Figure 1.3 (b)), the air flow resistance decreases and air penetration through the sample increases which results in the increase in permeability of the foam sample [19]. The permeability and air flow resistance of the foam plays an important role in heat transfer characteristic of metal foams in a case of natural convection. The heat transfer coefficient also increases with permeability due to higher air velocities through the foam [19].
Figure 1.3: Illustration of copper metal foams. (a) 40 PPI pore density. (b) 10 PPI pore density.

Figure 1.4: SEM image analysis. (a) Top view of a 40 PPI metal foam. (b) Cross section of an individual hollow metallic foam strut.
Numerous manufacturing techniques have been developed to produce open cell metallic foams with interconnected struts as shown in Figure 1.4 (a). Chemical Vapor Deposition (CVD) and Electro-Deposition are two metal deposition processes in which the metal is deposited on an open cell polymer foam. After the deposition process when the polymer is fully coated with metal, the polymer is removed by placing it in a high temperature oven where it burns off; leaving behind a porous metallic foam. The metallic struts formed by this technique are hollow, as seen in Fig. 1.4 (b). An overview of metallic foam manufacturing processes can be found in a review by Banhart [46].

The conduction path in metal foams is not direct due to the randomness of the struts which results in a low effective thermal conductivity [34], typically only 2% to 6% of the base metal value [35]. Metal foams are widely used due to their high surface area to volume ratio [8]. Porosity, pore density, pore size and fiber diameter of the open-cell metal foams are important characteristics which affect the heat transfer performance of the porous structure.

Foam structure is defined by several parameters, including:

**Porosity**

Porosity, \( \varepsilon \), is defined as the void volume in the porous sample divided by its total volume. As the porosity of a sample increases, the amount of solid material of that sample decreases, which decreases the strength of that sample.

**Pore Density**

Pore density is measured in pores per inch (PPI) by counting the number of pores per linear inch. In this study 40 and 10 PPI copper foams were investigated.

**Pore Size**

Investigators have developed an approximate shape for individual pores of the foam and use a dodecahedron to represent each unit cell [40]. The equivalent diameter of one of the faces of the dodecahedron unit cell is called pore size \( d_p \), and since the size of unit cells through the sample are not uniform, statistical average of the local values are used to show the value of pore size [8]. The pore size and porosity are independent of each other [41].
Fiber Diameter

Fiber diameter $d_f$ is the equivalent diameter of each strut (fiber) which can be measured by analyzing the cross-section of the strut and is a function of porosity [8, 41, 42].

1.2.3 Introduction to Wire Meshes

Wire meshes are commercially available in a wide range of materials and there are also numerous wire orientations available in the market. Metallic wire meshes are porous materials consisting of an array of metals forming square, rectangular or circular pore patterns. Wire mesh screens are manufactured in a variety of pore sizes, wire diameters and wire types. Wire meshes are also categorized based on the type of connection between wires as welded, woven, crimped or molded. Woven wire meshes are manufactured in different pore densities as show in Figure 1.5. Investigators have studied the fluid-flow and heat transfer performance of wire meshes for the flow inside a channel, where they all indicated the importance of the orientation and the geometry of the wire mesh on its heat transfer performance [4, 13, 30]. In this study the heat transfer of the wire mesh in both vertical and horizontal orientations is compared to that of metallic foam. Tube heat exchangers were modified using wire meshes in order to enhance the heat transfer performance of the system by increasing its external surface area. Wire meshes enhance the heat transfer of the heat exchanger in a manner similar to solid plate fins, but due to their porosity, the pressure drop across them is significantly lower.

Figure 1.5: Woven copper wire mesh screens. (a) 10 PPI. (b) 40 PPI.
1.2.4 **Metal Foams and Wire Meshes as Heat Exchangers**

Investigators have been studying and analyzing wire meshes and metal foams in order to characterize their fluid flow and heat transfer behavior for several decades [4, 13]. Most studies were performed for temperatures below 200°C, where the effect of radiation can be neglected. The researchers concentrated their study on forced convection and studied the pressure drop across the porous structure and analyzed the heat transfer performance. Only a few researchers Joen et al. [10] fabricated gas to liquid heat exchangers using porous structure but they tested them in low temperature environments and did not deal with the complexity of the heat transfer calculation in the presence of radiation. In this section a brief summary of the relevant research regarding radiation, natural convection and gas to liquid heat exchangers in metal foams and wire meshes is presented.

**Gas to Liquid heat exchangers and effect of topology on the heat transfer performance**

Assaad et al. [9] created a new class of heat exchangers using wire mesh. They stacked and sintered stainless steel woven wire meshes together and created wire mesh bricks. They machined the bricks and cut them into thin wafers that could be combined to create porous structures. In order to contain the working fluid inside this porous structure they deposited metal coatings on the outer surface of the wafers using pulsed gas dynamic spraying (PGDS). They claimed, based on their burst and tensile tests, that the fabricated compact wire mesh heat exchanger could withstand internal pressures as high as 19.1 MPa.

Joen et al. [10] fabricated a single row heat exchanger consisting of aluminum metal foam covering aluminum tubes. They placed their samples inside a wind tunnel and tested various parameters including Reynolds number, tube spacing, foam height and the type of foam. They discovered that increasing the foam height reduces the exterior convection resistance while increasing the pressure drop. They also tested brazed and unbrazed samples which proved the importance of bonding and concluded that more research is needed to develop efficient and cost-effective connection (brazing) techniques to better connect the tube to the foam to provide solid metallic bonds.
Sypeck [11, 12] studied wrought metallic sandwich structures with open cell truss cores. He designed and fabricated aluminum wrought metallic structures using deformation, assembly and joining process of crimped aluminum alloy weaves and perforated aluminum alloy sheets. For the manufacturing process he used brazing in a vacuum furnace. He achieved a good connection between the outer wall and wrought metals using brazing.

Kim et al. (2004) [13] experimentally investigated the heat transfer and pressure drop for compact aluminum lattice-frame materials. They discovered that the pressure drop is strongly dependent upon the orientation of the structure. The comparison between the sandwich structure and empty channel showed that the heat sink can remove heat approximately 7 times more than an empty channel. They concluded that aluminum lattice-frame materials are ideal candidates for applications requiring heat transfer and mechanical load carrying capabilities.

Khayargoli et al. [14] investigated the effect of the microstructure of nickel and nickel-chromium alloys metal foams on flow parameters. They found that the permeability increases as the pore size increases which was due to increases in drag forces on the flowing fluid.

Tian et al. [4] studied fluid flow and heat-transfer during forced convection through cellular copper lattice structures. To find the maximum heat transfer performance of the weave copper meshes they tested several configurations. They discovered that unlike open-cell metal foams and packed beds, the friction factor of the bonded wire screen apart from being a function of porosity is also a function of orientation. They concluded that “wire-screen meshes compete favorably with the best available heat dissipation media”[4]. The overall thermal efficiency index of the copper textiles-based media (meshes) was found to be approximately 3 times higher than that of copper foam due to its high pressure drop.

Paek et al. [16] experimentally investigated thermo physical properties of different porosity aluminum foams. They measured the effective thermal conductivity and the permeability of the foam and found that effective thermal conductivity increases as the porosity decreases. Also at a fixed porosity, as the surface area in a given volume increases, flow resistance and
pressure drop increase due to a decreased permeability. They correlated the friction factor with the permeability-based Reynolds number.

Salimi Jazi et al. [7] and Tsolas [8] used wire arc thermal spraying process and deposited an Inconel 625 skin on copper and nickel foams to contain air flowing inside the metallic foam. Figure 1.6 illustrates the Inconel 625 skin which was applied on the 10 PPI nickel foam. Recently Assaad et al. [9] fabricated wire mesh heat exchanger by using pulsed gas dynamic spraying technique. They deposited metallic powders on the outer surface of metal wire mesh wafer and created compact heat exchangers. These research groups created a protective layer around their porous structure in order to contain the working fluid instead of brazing a metal sheet to the porous substrate. This porous structure increases the surface area of the wall which resulted in enhancement of the performance of the heat exchanger.

**Natural Convection and Thermal Radiation**

Zhao, Tassou and Lu [17] developed an analytical model to characterize the radiative transport processes in open-cell metal foams with an idealized morphology. Their model was in good agreement with measurements using a guarded-hot-plate apparatus for steel alloy foams. Their results showed that the contribution of reflectance to radiation can be as high as 50% and should not be neglected. They also found that the radiative conductance increases linearly with the increase in foam cell size.

![Image of thermal sprayed skin deposited on the 10 PPI metal foam](image-url)

**Figure 1.6:** Thermal sprayed skin deposited on the 10 PPI metal foam
Hetsroni, Gurevich and Rozendblit [18] investigated the effect of natural convection in metallic foam strips. The energy generated in the porous strips was assumed to be equal to heat losses by convection, radiation and conduction. They achieved 18-20 times increase in heat transfer when they compared the natural convection in 20 PPI metallic foam in both vertical and horizontal plates relative to flat plates with the same overall dimensions.

Bhattacharya and Mahajan [19] experimentally investigated natural convection in aluminum foams with varying pore density. They found that for a given pore size, heat transfer rate increases with porosity which they suggested is due to the dominant role of conduction in heat transfer. They also found that when keeping the porosity constant, the heat transfer rate was lower for higher pore density. In their study, foam samples were placed on the hot plate with the maximum temperature of 75°C.

Zhao, Lu and Hodson [20] experimentally measured radiative transfer in FeCrAIY (a steel based high temperature alloy) foam where they measured spectral transmittance and reflectance at different wavelength ranges over the range of 300-800 K and determined the spectral extinction coefficient and foam emissivity and extinction coefficient. The foam was considered to be semitransparent, capable of emitting, absorbing and scattering thermal radiation. They discovered when the temperature increased, the total extinction coefficient increased but the total reflectance decreased. They showed that at fixed porosity, the radiative conductivity of the foam increases with increasing temperature and cell size.

Zhao, Lu, Hodson and Jackson [21] examined the temperature dependence of effective thermal conductivity of steel alloy foams for temperatures between 200-800 K, under both vacuum and atmospheric conditions. They discovered that the transport of heat is dominated by thermal radiation and effective thermal conductivity increases at high temperatures. They also compared the effective thermal conductivity calculated at pressure varying from atmospheric to vacuum conditions and established the importance of natural convection since the effective thermal conductivity at atmospheric pressure was twice that of in a vacuum.
Phanikumar and Mahajan [22] used aluminum foam samples with varying pore sizes and porosity and examined their flow and heat transfer characterization when heated from below. The heat transfer rate for a given Rayleigh number decreased as pore density increased from 5 to 40 PPI, where resistance to flow diminished as pore density decreased and air penetration through the sample increased. They created a numerical model using the assumption of local thermal non-equilibrium (LTNE) by using two-equation model for energy where they successfully predict heat transfer in aluminum metal foam. Their LTNE model was in better agreement with the experimental results than the local thermal equilibrium model.

Lu et al. [30] reviewed the thermal characteristics of metallic sandwich structures with porous cores, based on numerical simulations and experimental results for cooling of the wall of a heated channel. They combined data showing the influence of topology on the Nusselt number, Reynolds number and friction factor, as displayed in Figure 1.7. Wire meshes, which are classified as “Textile (Cu)”, have higher Nusselt numbers (Figure 1.7 (a)) and lower friction factors than metal foams. Tian et al (2004) [4] investigated the fluid-flow and heat-transfer characteristics of cellular copper lattice structures and analyzed several configurations to identify the optimum orientation to obtain maximum heat transfer. After comparing wire-screen meshes, metal foams, packed beds and louvered fins, they concluded that “the overall thermal efficiency index of the copper textiles-bases media is approximately three times larger than that of stochastic copper foams”. This was due to the fact that the pressure drop across periodic wire-screen was much lower than metallic foams.
Figure 1.7: A compilation of results for: (a) friction factor as a function of the Reynolds number at the panel level and (b) the Nusselt number as a function of the Reynolds number. [30]
1.3 Research Objectives

The objectives of this research are:

- To build water-air heat exchangers using metal foams or wire meshes to increase the external heat transfer area.
- To measure the increase in heat transfer due to the use of metal foams and meshes for several different heat exchanger designs, while varying the water flow rate.
- Determine the effect of porosity and orientation of foams and meshes on heat transfer.
- Develop correlations to predict the heat transfer from the heat exchangers.
- To reduce high temperatures oxidation of open cell copper foams and meshes by developing a technique to apply uniform nickel coatings.
- To build a gas to liquid heat exchanger using twin-wire arc thermal spray coating techniques to connect wire mesh to tubes.

Parameters Varied:

- Metal foam material: Copper \( (k_{Cu} = 401.0 \text{ W/mK}) \)
- Wire mesh material: Copper\( (k_{Cu} = 401.0 \text{ W/mK}) \) and 304 Stainless Steel \( (k_{SS} = 17.0 \text{ W/mK}) \)
- Metal foam pore size: 40PPI and 10PPI
- Wire mesh pore size: 40PPI, 10PPI and 4PPI
- Coolant flow velocities: 0.0127 – 0.0317 kg/s (0.2 – 0.5 GPM)
- Furnace operation temperature: 300 - 800 °C
1.4 Organization of Thesis

The first chapter starts with a general introduction to this research followed by a literature review of open cell metals and heat exchangers.

Chapter 2 introduces different methods of metal coatings and explains the process of nickel coating of porous copper foams and meshes for high temperature heat exchange applications.

Chapter 3 explains the reasons for using heat exchangers, gives a brief summary of different types of heat exchangers and their performance. It describes in detail the manufacturing process of each open cell metal heat exchanger and how to calculate their surface area.

Chapter 4 describes the experimental apparatus that was used to test each heat exchanger which includes the coolant water flow and the high temperature furnace.

Chapter 5 describes the heat transfer enhancement that was measured for each prototype. In this study the effect of natural convection and radiation for metal foams and wire meshes are compared. First, the energy extracted from the furnace using water is calculated which is used to calculate the performance of each heat exchanger. Second, the extracted energy is used to calculate the convection heat transfer coefficient, which was used to develop an overall Nusselt number correlation. It was shown that as pore density increases, less air penetrates through the porous metals which will reduce the heat transfer performance of the heat exchanger.

Chapter 6 describes the fabrication of wire mesh stainless steel heat exchangers using wire arc thermal spray coating for high temperature applications. Different orientations of wire mesh placement over coolant water tubes were examined. Heat transfer enhancement was compared by developing an overall Nusselt number correlation for each heat exchanger. The results were promising and it was shown that thermal spray methods can be used to connect the fin to the tube.

Chapter 7 presents conclusions from the present research.
Chapter 2
Porous Materials for High Temperature Applications

2.1 Introduction

Heat transfer through a heat exchanger depends on its geometry and thermal properties. In Section 1.2.1 different geometries and types were discussed, and this chapter will concentrate on the effect of thermal property performance of the heat exchanger.

The amount of heat that can pass through a material depends on the thermal conductivity $k$ of that material, and to increase the efficiency and the performance of the heat exchangers, the selected material needs to have high thermal conductivity. It must be commercially available, affordable and be able to operate in high temperature environments. Open-cell metal foams are currently commercially available in nickel, copper and aluminum, and manufacturers are trying to increase the range of materials by developing new manufacturing processes. On the other hand, since wire meshes have been made for many years, they are commercially available in many materials. Aluminum’s high thermal conductivity ($k = 237$ W/m. °C) makes it a very good candidate for heat transfer applications, but the fact that its melting temperature is only 660°C makes it unsuitable for use at high temperatures. Copper has a high thermal conductivity ($k = 401$ W/m. °C) with high melting temperature (1084°C) but its maximum operating temperature is limited by its tendency to oxidize rapidly when heated.

The oxidation temperatures of different, commercially available materials were reviewed. Metal oxides have low thermal conductivity and if deposited on a heat transfer surface, reduce the performance of the heat exchanger. Figure 2.1 shows the maximum service temperature of some metals as a function of their thermal conductivities. Copper and copper alloys have high thermal conductivity but their operation temperature is very low compared to stainless steel and nickel alloys.
Figure 2.1: Maximum service temperature vs. thermal conductivity for various materials (CES EduPack 2006, Granta Design Limited, Cambridge, UK, 2006.)
In the case of copper, formation of an oxide layer on the surface at temperatures above approximately 150°C limits its operation temperature. There are two different classes of oxide films; protective and non-protective. Protective films stop oxidation since the rate of corrosion of the protective layer is very low, whereas non-protective films are porous and contain cracks.

Copper oxidizes at room temperature forming a film of cuprous oxide (Cu$_2$O) by direct reaction of copper with oxygen [23]. The oxidation reaction can be written as:

$$4 \text{Cu} + \text{O}_2 \rightarrow 2 \text{Cu}_2\text{O}$$

The oxidation reactions start at room temperature and as the temperature increases, the oxide film is overlaid by CuO film and since the CuO layer is nonprotective, it also creates Cu$_2$O [24].

$$2\text{CuO} \rightarrow \text{Cu}_2\text{O} + \frac{1}{2}\text{O}_2$$

The copper oxidizes much faster than nickel and its rate of oxidation changes from logarithmic to parabolic as the temperature increases [25].

Nickel, as seen in Figure 2.1, has a higher operation temperature than copper, and nickel oxide is a protective film which protects the layer below it. The thermal conductivity of nickel is also much higher ($k = 91\text{W/m. }^\circ\text{C}$) than stainless steel ($k = 17\text{W/m. }^\circ\text{C}$) which makes it a better candidate for heat exchanger applications.

To use the outstanding material and heat transfer properties of both copper and nickel, copper was used as the base material and nickel was coated on the copper to stop the oxidation at high temperature applications as shown in Figure 2.2. The coating was uniform around hollow copper foam struts and it covered the outer surface of copper foam struts completely.
Meal foams have a porous structure and the coating must be uniform throughout the length of the sample. The uniformity of the applied coating was investigated for both electroplating and electroless plating. Both material deposition techniques and procedures are explained in this chapter.

### 2.2 Electroplating

Electroplating is a coating process that uses electrodeposition techniques to apply coatings. The object to be coated is given a negative charge (cathode), and submerged into a solution that contains a salt of the deposited metal is shown in Figure 2.3. The metallic ions of the salt solution, which are positively charged, are moved by an electric field to produce a coating on the surface of the cathode [26].
The maximum allowable surface area of an object that can be coated by electroplating is limited by the current that can be supplied. Due to the structure of the foam, which has a high surface area to volume ratio, a large current is required for electroplating. Different solution baths are available for the nickel electroplating process; Watts nickel and nickel sulfamate as shown in Table 2.1. In this study Watts nickel bath was chosen with the electroless decompositions chosen and condition listed in the Table 2.1.

To test the coating quality of the electroplating process on the foam struts, two 40 PPI pore density copper samples were machined with identical dimensions 18 mm × 11 mm × 10 mm ($L \times W \times H$). The prepared bath solution consisted of 300 g/L nickel sulfate; 45 g/L nickel chloride and 45 g/L boric acid. The temperature of the solution was kept fixed at 55°C by placing it inside it on a heater during the electroplating process as shown in Figure 2.4 (b). In this study copper foam (cathode) was placed inside the solution, in the middle of a titanium basket which held the nickel solution as shown in Figure 2.4 (a).
positive terminal of the supply was connected to the titanium basket and the negative terminal connected with a set of copper wires to the 40 PPI copper samples. To remove any oxide, both samples were cleaned using 10% sulfuric acid. The first sample was electroplated with 6 A/dm$^2$ cathode current density, and the other one with 10 A/dm$^2$ cathode current density to observe the variation in the coating thickness based on the cathode current.

Table 2.1: Nickel electroplating solutions [27]

<table>
<thead>
<tr>
<th>Electrolyte composition(a), g/L</th>
<th>Watts nickel</th>
<th>Nickel sultamate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nickel sulfate, NiSO$_4$·6H$_2$O</td>
<td>225-400</td>
<td>---</td>
</tr>
<tr>
<td>Nickel sulfamate, Ni(SO$_3$NH$_2$)$_2$</td>
<td></td>
<td>300-450</td>
</tr>
<tr>
<td>Nickel chloride, NiCl$_2$·6H$_2$O</td>
<td>30-60</td>
<td>0-30</td>
</tr>
<tr>
<td>Boric acid, H$_3$BO$_3$</td>
<td>30-45</td>
<td>30-45</td>
</tr>
</tbody>
</table>

**Operation conditions**

<table>
<thead>
<tr>
<th>Operation conditions</th>
<th>Watts nickel</th>
<th>Nickel sultamate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature, °C</td>
<td>44-66</td>
<td>32-60</td>
</tr>
<tr>
<td>Cathode current density, A/dm$^2$</td>
<td>3-11</td>
<td>0.5-30</td>
</tr>
<tr>
<td>Anodes</td>
<td>Nickel</td>
<td>Nickel</td>
</tr>
</tbody>
</table>
The required current was calculated by first finding the surface area of the foam and then multiplying it by the cathode current density to calculate the applied current as shown in Equation 2.1 and 2.2 respectively. The required current for 6 – 10 A/dm$^2$ were calculated to be 3A and 6A respectively. The metal foam surface area calculation using the occupied volume is shown in Chapter 5. The overall occupied volume by the metal foam samples was 2064 mm$^3$.

Surface area of the 40PPI copper foam

$$2.414 \frac{mm^2}{mm^3} \times 2064 \ mm^3 = 4984 \ mm^2 \approx 50 \ cm^2$$  \hspace{1cm} (2.1)

The required electroplating current using 6 A/dm$^2$ cathode current density,

$$50 \ cm^2 \times 6 \ A/(cm^2) = 3A$$  \hspace{1cm} (2.2)

Each sample was kept inside the bath for 30 minutes. The coating process was successful and both samples were nickel coated, as shown in Figure 2.5. Nickel clusters were created.

Figure 2.4: Electroplating tools. (a) Titanium basket used to hold the nickel. (b) Bath solution used for electroplating
around the corners and for 6A current, the clusters were thicker and more frequent than for the 3A current.

Figure 2.5: Electroplated samples using 3A and 6A current from left to right respectively.

To check the uniformity of the coating over foam struts, samples were cut and the coatings were analyzed along the cross section. It was found that the coating thickness was not uniform throughout the foam cross section. For the 3A current sample, most of the foam struts in the middle of the sample were not coated properly and in some cases were not coated at all, as shown in Figure 2.6. The red lines in Figure 2.6 represent the left and right corners of the cut cross section of the foam sample, the coatings were much thicker for the struts closer to the outer surface of the samples and in the form of clusters on both corners. The nickel coatings were much thicker for 6A compared to the 3A current but inconsistency of the coatings was still a problem. In the case of 6A current, all the struts in the middle of the sample were coated but the clusters on both side were much larger when compared with the sample which was coated using 3A current. For larger foam samples the applied current is much larger, which requires a power supply with greater capacity.
The electroplating process is not an effective method of applying nickel coatings on the porous structure of the foam due to the inconsistency of the coating layer, the surface area limitation due the need for high current and also the formation of nickel clusters around the corners. There is a need for a method that can deliver a more uniform coating through the sample without the sample size limitation.

Figure 2.6: Ni Electroplating of Copper foam (3A Current)
2.3 Electroless Plating

Electroless plating is a chemical deposition technique which requires no external source of current [28]. The electroless nickel plating bath consists of a reducing agent, a source of nickel ions, a stabilizer and a complexing agent. The nickel is deposited when hydrogen is released by a reducing agent and oxidized, producing a negative charge on the surface of the part causing deposition of nickel [29]. In this study nickel sulfate was used as the source of nickel, samples were plated at Eastend Plating Co Ltd. To test the coating quality of the electless plating process on the foam struts, a 40PPI copper foam sample was machined with dimensions 18 mm × 11 mm × 10 mm (L × W × H). Hydrochloric acid was used to dissolve the copper oxides from the sample before the plating process.

To check the uniformity of the coating over foam struts, the samples were cut and the coating was analyzed along the cross section of the cut surface. Six random struts were analyzed under the microscope, and it was shown that the coating thicknesses were uniform throughout the foam cross section and all the struts were completely coated as shown in Figure 2.7. Unlike with electroplating, on electroless plated samples nickel clusters did not form on the corner of the samples.

Energy-dispersive X-ray spectroscopy (EDS) equipment on the scanning electron microscope (SEM) (TM3000, Hitachi High-Technologies Canada Incorporated, Toronto, ON) was used to create X-ray spectrums from the scanned area of the SEM on the non-epoxy covered sample. EDS allows the SEM user to identify different atomic number elements and their distribution over the scanned area to better characterize a surface.
The strut surface of the nickel-coated copper foam samples were analyzed using EDS. The samples were kept at room temperature and were not exposed to any high temperature environment prior to the test. Random struts were checked throughout the coated foam samples to ensure the accuracy of the results. The atomic percentages verified the existence of the nickel on the outer surface of the foam, as shown in Figure 2.8.

Figure 2.7: Nickel-coated copper foam struts using electroless plating technique.

Figure 2.8: Nickel on the outer surface of the foam.
Figure 2.8: Nickel and oxygen atomic percentage for a nickel-coated copper foam surface
The atomic percentage of the oxygen was also examined to insure the absence of the oxygen which would indicate the presence of oxides on the surface. The oxygen content throughout the sample was zero since the oxidizing temperature is approximately 200 ºC, as shown in Figure 2.8. The electroless plating technique was used to produce nickel coatings on all fabricated foam and wire mesh copper heat exchangers after the coating composition was checked using EDS.

### 2.4 Nickel oxide formation

In this study, as will be discussed in Chapter 3, heat exchangers were fabricated where the foam struts (fins) were not connected to the coolant tube with soldering paste. Since the foam struts were not cooled using the coolant water running inside the heat exchanger tube, the foam temperature was much higher than the samples with solder bonding between the strut and the foam. As a result nickel oxidized and created the colorful patterns shown in Figure 2.9. Silver nickel spots on the sample indicate the good connection between foam struts and the tube which resulted in high heat transfer and lower foam temperature. The oxidized samples where analyzed using EDS and the results are presented in Figure 2.10. High oxide content was measured through samples which indicated the presence of nickel oxide on the surface of the foam struts.
Figure 2.9: Oxide pattern after the oxidation. (a)(d) 40 PPI foam. (b)(c) 10 PPI foam.
Figure 2.10: Nickel and oxygen atomic percentage for a nickel-coated copper foam surface after the sample was exposed to the high temperature
Chapter 3
Fabrication of Heat Exchangers

3.1 Heat Exchanger Apparatus

Ten open-cell metal heat exchangers were fabricated for this study; four of which used metal foams and the other six used wire meshes. Each heat exchanger consisted of a copper tube for the coolant to pass through and circulated inside the furnace. The overall length of the copper tube, 482 mm, was held fixed for all heat exchangers. The tube was bent into a U shape using a tube bender until both sides were parallel as shown in Figure 3.1. Sn-alloy soldering paste (Loctite RP15, Henkel AG and Company, Düsseldorf, GER) was used to permanently connect the tube to the porous structure which is explained in Sections 3.2 and 3.3. Appendix A shows a schematic of the metal foam heat exchanger.

Four metal foam heat exchangers were fabricated, two using 10 PPI and the other two using 40 PPI metal foams (Dalian Thrive Mining Co. Ltd, Dalian, China). Foam pieces were cut using an electric-discharge machine to the dimensions of 152 mm × 44 mm × 20 mm. Two 6 mm diameter holes were bored through the length of each foam piece to insert the copper tube.

The volume occupied by the metal foam was calculated by subtracting the overall occupied volume by the volume of the holes which were bored using EDM and equaled 127000 mm$^3$. The surface areas for both heat exchangers were calculated by multiplying the occupied volume of metal foams by the surface area density of that open-cell metal. The surface area of the 40PPI copper foam was 306574 mm$^2$ and the 10PPI copper foam was 46482 mm$^2$. These surface areas were used in Chapter 5 to calculate the heat transfer performance of the heat exchangers.
Table 3.1: Heat exchanger parameters of the fabricated prototypes

<table>
<thead>
<tr>
<th>HEX #</th>
<th>Tube OD mm</th>
<th>Porous metal</th>
<th>Wire mesh PPI</th>
<th>Overall tube length mm</th>
<th>Mesh wire diameter mm</th>
<th>Sn - soldering paste</th>
<th>Porous metal surface area mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6</td>
<td>Foam</td>
<td>40</td>
<td>483</td>
<td>N/A</td>
<td>No</td>
<td>29497</td>
</tr>
<tr>
<td>2</td>
<td>6</td>
<td>Foam</td>
<td>10</td>
<td>483</td>
<td>N/A</td>
<td>No</td>
<td>46477</td>
</tr>
<tr>
<td>3</td>
<td>6</td>
<td>Foam</td>
<td>40</td>
<td>483</td>
<td>N/A</td>
<td>Yes</td>
<td>29497</td>
</tr>
<tr>
<td>4</td>
<td>6</td>
<td>Foam</td>
<td>10</td>
<td>483</td>
<td>N/A</td>
<td>Yes</td>
<td>46477</td>
</tr>
<tr>
<td>5</td>
<td>6</td>
<td>Wire mesh</td>
<td>40</td>
<td>483</td>
<td>0.3</td>
<td>Yes</td>
<td>17097</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>Wire mesh</td>
<td>10</td>
<td>483</td>
<td>0.6</td>
<td>Yes</td>
<td>10697</td>
</tr>
<tr>
<td>7</td>
<td>6</td>
<td>Wire mesh</td>
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<td>483</td>
<td>0.3</td>
<td>Yes</td>
<td>34194</td>
</tr>
<tr>
<td>8</td>
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<td>Wire mesh</td>
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<td>483</td>
<td>0.6</td>
<td>Yes</td>
<td>21394</td>
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<tr>
<td>9</td>
<td>6</td>
<td>Wire mesh</td>
<td>40</td>
<td>483</td>
<td>0.3</td>
<td>Yes</td>
<td>29497</td>
</tr>
<tr>
<td>10</td>
<td>6</td>
<td>Wire mesh</td>
<td>10</td>
<td>483</td>
<td>0.6</td>
<td>Yes</td>
<td>46477</td>
</tr>
</tbody>
</table>
3.2 Metal Foams

3.2.1 Bonding Process of Metal Foams to the Heat Exchanger

The coolant (water) runs inside the copper tube as shown in Figure 3.2, which is positioned inside the copper foam. Effective bonding of foam struts to the outer surface of the heat exchanger’s tube plays an important role in the efficiency of metal foams in heat exchanger applications. The conduction heat transfer between the tube and foam occurs through the foam struts and the tubes’ outer surface. To increase the heat transfer performance of the heat exchanger, the strength of the connection between these two must be maximized. The thermal resistance between the foam and the tube ($R_{\text{bond}}$) controls heat transfer. A connection method was developed which provided a uniform connection between the foam and the tube along the length of the heat exchanger. Using a Sn-alloy soldering paste, 90 percent of the foam struts were bonded to the outer surface of the copper tube.

To ensure that maximum connection is achieved throughout the length of the heat exchanger between struts and the tube, different tolerances between the bored hole diameter inside the foam and the tube outer diameter were examined. Eight copper tubes were machined to 203 lengths; four samples with 6 mm outer diameter (OD) and the other four samples with 9 mm OD. Eight copper metal foams with pore density of 40 PPI (Dalian Thrive Mining Co. Ltd, Dalian, China) were also cut using an electric-discharge machine to the dimensions of $152 \text{ mm} \times 44 \text{ mm} \times 20 \text{ mm}$ ($L \times W \times H$). Two 6 mm diameter holes were bored through the length of the foam for positioning of the copper tube through the porous foam.

To achieve a good connection, the tube’s outer surface needs to be roughened by sand blasting and also be free from oxide prior to tin deposition. At first all 6 and 9 mm diameter copper tubes were sand blasted to enhance the mechanical interlock bonding between the coating and the surface as shown in Figure 3.1 ($b$). Rosin, as a chemical cleaning agent (flux), was applied prior to soldering to remove the oxides from the surface and to ease the flow of the solder on the surface. A thin layer of tin was soldered on the copper tube to
further enhance the probability of a uniform connection between the foam struts and the tube as shown in Figure 3.1 (c).

![Image of foam struts and tubes](image)

Figure 3.1: 6 mm and 9 mm copper tubes. (a) Plain. (b) Sand blasted. (c) With a tin coating layer

The bored hole diameters inside the foam samples were 5 mm, 6 mm, 7 mm and 8 mm for the 6 mm diameter tubes and 8 mm, 9 mm, 10 mm and 11 mm for the 9 mm diameter tubes. Sn-alloy soldering paste (Loctite RP15, Henkel AG and Company, Düsseldorf, GER) was applied into the inner surface of the hole (machined into the foam) and the outer surface of the tin-coated tube before the tube was inserted in the metal foam. Soldered tubes were then positioned inside the foam and placed in an oven to melt the solder and bond the foams to the tubes. Samples were cut using a diamond cutter (ME-B221-1201 Brilliant 221, Clemex Technologies Inc., Longueuil, QC) on five locations with 25 mm spacing between them to analyze the effectiveness of the connections, as shown in Figure 3.2.
Figure 3.2: Illustration of a Tin connection between the tube and foam struts

It was observed that the connections were more uniform and more struts were connected to the tube’s outer surface when the tolerance between the tube’s OD and the bored hole diameter was zero. Using a Sn-alloy soldering paste and zero tolerance, 85 to 95 percent of the foam struts were bonded to the outer surface of the copper tube throughout the length of the foam. Using the described connection method, 10 PPI and 40 PPI copper metal foam heat exchangers were fabricated; Figure 3.3 illustrates a 40 PPI heat exchanger.
In order to stop the oxidation of the foam at high temperatures, uniform 30 μm layer of nickel was applied using electroless plating on all four metal foam copper heat exchangers; Figure 3.4 illustrates a 40 PPI nickel-coated copper metal foam heat exchanger where the foam struts are fully connected to the tube.
3.3 Wire Mesh Heat Exchangers

Six tube heat exchangers were fabricated using copper wire mesh; three of which used 10 PPI and the other three used 40 PPI wire meshes. Similar to metal foam heat exchangers, each fabricated wire mesh heat exchanger consists of a copper tube whose overall length is 483 mm and held fixed for all heat exchangers. The tubes were bent using a 6 mm tube bender until both sides were parallel. First, four heat exchangers were fabricated by positioning wire mesh screens on top of the copper tube, as shown in Figure 3.12. The screen dimensions were identical to those of metal foam heat exchangers 152 mm × 44 mm ($L \times W$) in order to analyze and compare the efficiency and performance of the extended surface area of both porous metals for the same occupied area.

The first two samples were fabricated using only one screen of wire mesh; the first sample was fabricated using 10 PPI woven wire mesh with 0.3 mm wire diameter and the second sample using 40 PPI mesh with 0.6 mm wire diameter, as shown in Figure 3.5 (a). The second two samples were fabricated using the same wire mesh diameters as the first two samples but instead of one screen, two screens were used, as shown in Figure 3.5 (b).

Prior to using Sn-alloy soldering paste (Loctite RP15, Henkel AG and Company, Düsseldorf, GER) to permanently connect the tube to the porous structure, the copper tubes were sand blasted and flux was then applied to remove oxides. A thin layer of tin was also soldered on the copper tube to give a uniform connection between the wire mesh and the tube, the process is explained in section 3.2.2.
Sn-alloy soldering paste was carefully applied on the samples, which were then placed in an oven to melt the solder and bond the screens to the tubes, as shown in Figure 3.6.
In order to stop the oxidation of the wire mesh heat exchanger at high temperature, a uniform 30 μm layer of nickel coating was applied using electroless plating on all four copper heat exchangers. Figure 3.7 illustrates a single screen 10 PPI nickel-coated copper wire mesh heat exchanger (a) and a double screen 40 PPI nickel-coated copper wire mesh heat exchanger (b) where the wire meshes are fully connected to the tube.

![Figure 3.7](image1.png) (a)

![Figure 3.7](image2.png) (b)

Figure 3.7: Heat exchangers after electroless Nickel plating (a) Single screen 10 PPI wire mesh heat exchanger. (b) Double screen 40 PPI wire mesh heat exchanger

To better compare the performance of the metal foam to the wire mesh, there was a need for a wire mesh heat exchanger with the same extended surface area as the metal foam. 10 PPI and 40 PPI heat exchangers were fabricated using parallel wire screens, as shown in Figure 3.7. Each heat exchanger consists of a calculated number of wire meshes with a length equal to the width of the metal foam samples, 44 mm, and a width equal to the thickness of the foam sample, 20 mm.

Sn-alloy soldering paste was applied where wire meshes are connected to the tube for each screen to enhance the connection between both substrates, as shown in Figure 3.8. Both
fabricated 10 PPI and 40 PPI samples were then placed in an oven to melt the solder and bond the screens to the tubes.

Figure 3.8: Vertical screen wire mesh heat exchangers. (a) 40 PPI mesh. (b) 10 PPI mesh.
Required number of screens and spacing

Since the surface area of the heat exchanger using vertical screens must match the overall surface area of the foam heat exchangers, the surface area of the foam was divided by the surface area of a single machined wire mesh screen to calculate the required number of screens, Equation 3.1.

\[
\frac{\text{Surface area of the foam}}{\text{Surface area of one screen}} = \text{Required number of screens} \tag{3.1}
\]

\[
\frac{46477 \text{ mm}^2}{1355 \text{ mm}^2} = 34 \text{ screens}
\]

In total, 34 screens were machined using 10 PPI copper wire mesh and 139 using 40 PPI copper wire mesh with dimensions of 44 mm and 20 mm \((L \times W)\).

To calculate the spacing between screens, the overall thickness of all screens was calculated using Equation 3.2. The thickness of a single woven wire mesh screen was defined as the summation of the diameter of two wires of that specific mesh [35]

Overall screen thicknesses for 40 PPI wire screens:

\[
\text{Number of Screens} \times \text{Thickness of each screen} = \text{Overall thickness of screens} \tag{3.2}
\]

\[
34 \text{ screens} \times (0.6 \text{ mm} + 0.6 \text{ mm}) = 41 \text{ mm}
\]

The calculated spacing between screens were 3 mm and 0.6 mm for 10 PPI and 40 PPI wire mesh screens respectively using Equation 3.3:

\[
\frac{\text{Allowable length} - \text{Overall screen thicknesses}}{\# \text{ of screens} - 1} = \text{Space in-between screens} \tag{3.3}
\]

\[
\frac{152 \text{ mm} - 43 \text{ mm}}{(34 - 1)} = 3 \text{ mm}
\]
The screens were positioned with the required spacing on the copper tube as shown in Figure 3.8. A uniform 30 µm layer of nickel coating was applied using electroless plating on both 40 PPI and 10 PPI wire mesh copper heat exchangers. Figure 3.9 illustrates a picture of a 10 PPI nickel-coated copper wire mesh heat exchanger where the wire meshes are fully connected to the tube.

Figure 3.9: 10 PPI vertical screen wire mesh, nickel-coated copper heat exchanger
Chapter 4
Experimental Apparatus and Procedure

4.1 Introduction

The experimental apparatus fabricated to test open-cell metal heat exchangers consists of two circulatory water loops and a high temperature furnace. A schematic representation of the experimental setup is shown in Figure 4.1. For the experimental set-up heat transfer performance for each heat exchanger was tested where distilled water passes through the tube nickel coated copper tube to extract heat from the furnace depending on the geometry of the heat exchanger. The high temperature furnace (Thermolyne F46128CM, Thermolyne Thermo Scientific) was chosen to create a steady high temperature environment to test the fabricated heat exchangers.

4.2 Coolant Water System

Distilled water was used as the coolant for testing the heat exchangers. Referring to Figure 4.1, the bottom line (1) feeds the water from the distilled water reservoir into the transfer pump (Model 360, Pony Pump Corporation, Los Angeles, CA), from which it pumps the water to the T junction. Two 6 mm bonnet needle valves were attached to both ends of the T junction to control the flow downstream to the heat exchanger (2) and back into the main distilled water reservoir (4). Line (3) feeds the water to a damper (Model DOT-3E1800, Swagelok Corporation, Mississauga, ON) to stop any fluctuation and from the pressure gage (Model 63-3109, Matheson Corporation, Montgomeryville, PA.) to a rotameter (7200 Series, King Instrument Co, Garden Grove, CA ) with a flow range of 0.013 kg/s (0.2 GPM) to 0.032 kg/s (0.5 GPM). The water then continues onto the inlet of the heat exchanger, where the heat exchanger was positioned inside the furnace. The heated water exits the heat exchanger but before it is sent back to the main water reservoir it is fed to another heat
exchanger where it was cooled using a secondary water loop. The secondary water loop consists of a copper coil and a drum to extract the heat from the main water loop to stop any boiling inside the tubes and to maintain a steady inlet water temperature to the heat exchanger. At the inlet and outlet of the heat exchanger, T junctions were placed to attach the required thermocouples to the cycle and measure the inlet and outlet water temperature for each heat exchanger. 13 mm type K pipe plug thermocouple probes with the standard error of ±0.1 °C were fixed to the T junction. The probes were placed such that when the water passes through the fitting, the probe comes in to contact with it.

Five thermocouples were attached to the surface of the heat exchanger to monitor and record the local surface temperature of the open cell metals as shown in Figure 4.1. Type-K thermocouples, with 0.6 mm diameter junctions were attached to the top surface of the porous structures of each heat exchanger at designated places with 38 mm spacing between each thermocouple. To ensure a good connection between thermocouples and surfaces, a high thermally conductive paste (Omegatherm 201, Omega Company, Stamford, CT) was applied where each thermocouple sat on the surface. Finally, all thermocouples were fixed using high-temperature cement (CC High Temperature, Omega Company, Stamford, CT) which was applied on top of the thermocouple. The same procedure was used to attach thermocouples to the surface of the nickel-coated tube to record the tube surface temperature.

A National Instruments Data Acquisition (DAQ) unit was used to record the thermocouple voltages. The DAQ was connected directly to a computer which was equipped with LabVIEW Signal Express v.3.0 (National Instruments Corporation, Austin, TX).

In order to calculate the overall uncertainty associated with both the DAQ system and the thermocouple, the method developed by Wheeler and Ganji was used [45]. The uncertainty of DAQ and thermocouple ± 1°C were measured separately and then combined using the same method. Appendix D shows the uncertainty calculation.
Figure 4.1: Schematic of the heat exchanger apparatus showing the instrumentation and, both water fluid loop configurations.
4.3 High Temperature furnace

The purpose of the study was to investigate the efficiency of the fabricated heat exchangers in high temperature environments. The furnace (Thermolyne F46128CM, Thermolyne Thermo Scientific) was chosen due to its high maximum operation temperature of 1700°C and geometry. Since the door of the furnace was a solid block without any openings, it was removed and replaced by an Aluminum oxide insulation sheet which was cut to the required size for the experiment.

Figure 4.2: Schematic of the furnace apparatus with the heat exchanger positioned inside.
dimensions. Two holes were drilled into the aluminum oxide insulation door for the inlet and outlet tube of the heat exchanger as shown in Figure 4.2. Inside the furnace, 6 heating elements and the furnace thermocouple are positioned as shown in Figure 4.3, where they are all connected to the furnace controller. Two insulation walls were fabricated using the same material as the insulation door, in order to stop direct radiation from the heating elements to the heat exchangers’ porous structure. Holes were drilled into both insulation walls to direct the hot air to the main chamber, where the heat exchanger was placed. Refer to Appendix C for schematics of the furnace with the heat exchanger positioned inside of it.

The heat exchanger is placed horizontally inside the furnace as shown in Figure 4.3 (b), where it was exposed to hot air.

Figure 4.3: Assembly drawing of a heat exchange inside the furnace. (a) Top view. (b) Side view.
The operating temperature of the furnace in the study was 300°C to 800°C which was controlled by the furnace thermocouples. Four type-K thermocouples were also mounted on the aluminum oxide insulation walls of the furnace to monitor the wall temperature for different furnace temperatures. At a steady state, when the furnace air temperature was stable, the wall temperature readings were always within 5 to 10% of the set furnace temperature.
Chapter 5
Heat Transfer Performance

5.1 Heat Exchanger Performance

The performances of the fabricated heat exchangers were investigated for four different water flow rates 0.013 – 0.032 kg/s (0.2 - 0.5 GPM) and at six different temperatures ranging from 300°C to 800°C. To ensure that a steady state was reached during the experiment, the water cycle was operated for 15 minutes for each furnace temperature and flow rate. The experiments were performed at a steady state and readings were taken when the thermocouple outputs had stabilized. The performance of all heat exchangers were compared based on three factors: the total heat transfer to the coolant water; radiation and convection heat transfer of the heat exchanger; and variation of Nusselt and Rayleigh number.

Figure 5.1 illustrates the temperature difference between the inlet and the outlet coolant temperature of the 40 PPI metal foam heat exchanger for 4 different flow rates, and as a function of the furnace temperature. The maximum temperature rise of 12°C was obtained between the inlet and outlet coolant flow of the heat exchanger for a flow rate of 0.013 kg/s with 800°C furnace temperature. It was observed that as the coolant flow increased from 0.013 kg/s to 0.032 kg/s, the temperature rise decreased. This decrease in the temperature can be explained using the energy balance Equation 5.1. The amount of heat extracted by each heat exchanger was approximately the same for different coolant flow rates, and depended largely on the furnace temperature. Therefore, as the coolant flow increased, the temperature rise decreased.

\[
Q = \dot{m}_{\text{coolant}} \times C_{p_{\text{coolant}}} \times (T_{in} - T_{out}) \tag{5.1}
\]

The temperature rises of the heat exchangers were compared for 800°C furnace temperature and the results are shown in Figures 5.2 - 5.4, where the y-axis represents the temperature rise and the x-axis is the mass flow rate of the coolant. Refer to Appendix B for the
temperature difference between the inlet and outlet coolant flow of all eleven heat exchangers. All fabricated heat exchangers resulted in higher temperature rise than the bare tube heat exchanger. The extended surfaces of wire mesh and foam enhanced heat transfer from the hot furnace environment to the cooling water running inside the heat exchanger tubes.

![Graph showing coolant temperature rise for 40 PPI metal foam heat exchanger for various flow rates, where furnace temperature increased from 300°C to 800°C. The experimental uncertainty values are ± 1.0°C in the temperature measurement.](image)

Figure 5.1: Coolant temperature rise for 40 PPI metal foam heat exchanger for various flow rates, where furnace temperature increased from 300°C to 800°C. The experimental uncertainty values are ± 1.0°C in the temperature measurement.

Two different types of wire mesh heat exchangers were fabricated in this study, with vertical and horizontal wire mesh panels as shown in Figure 3.7 & 3.8. In the case of horizontal wire mesh screens, all fabricated wire mesh screen heat exchangers outperformed tube heat exchanger, which indicated the effectiveness of porous fins as heat transfer enhancers as shown in Figure 5.2. When comparing the performance of the wire mesh screens together, 2 screen 10 PPI mesh outperformed the other heat exchangers. For both 10 PPI and 40 PPI horizontal heat exchangers, having two screens of wire mesh was better than one screen. Air penetrated much more easily through 10 PPI than 40 PPI wire mesh, since
the pore sizes were much larger. High pore density wire mesh caused extra resistance to air flow which reduced natural convection heat transfer. For 0.013 kg/s coolant flow rate, the heat transfer performance increased by a factor of 2 when adding a single 10 PPI screen mesh to the tube heat exchanger, and by a factor of 3 when the second screen was added.

The vertical wire mesh screens also outperformed the tube heat exchange and enhanced heat transfer as shown in Figure 5.3. The temperature rise recorded for the 40 PPI vertical wire mesh heat exchanger was higher than 10 PPI wire mesh, where heat transfer performance increased 2.2 times for 10 PPI screen mesh heat exchanger and 2.8 times for 40 PPI wire mesh heat exchanger. It was seen earlier from the results for the horizontal screen heat exchangers that air penetrates better through the pores of 10 PPI than the 40 PPI wire mesh. In a case of vertical screen heat exchangers air managed to flow through and in-between

Figure 5.2: Coolant temperature rise comparison for screen heat exchangers for various flow rates, where furnace temperature is fixed at 800°C. The experimental uncertainty values are ± 1°C in the temperature measurement.
screens for both wire meshes. Optimizing the spacing between wire meshes may produce even greater heat transfer, but that was not investigated here.

![Figure 5.3: Coolant temperature rise comparison for vertical wire mesh heat exchangers for various flow rates, where furnace temperature is fixed at 800°C. The experimental uncertainty values are ± 1°C in the temperature measurement.](image)

Both radiation and convection play a significant role in heat transfer from the ambient air to the foam. Zhao, Lu and Hodson [20] showed that at fixed porosity, the radiative conductivity of the foam increases with increasing temperature and cell size. The radiative properties of metallic foam are defined based on their transmittance, extinction coefficient, reflectance and emissivity. The total transmittance $\tau$ of a foam sample, which is the fraction of thermal radiation at wavelength $\lambda$ incident on the surface that penetrates it,

$$\tau(L) = \frac{\int_0^\infty I_\lambda(L) d\lambda}{\int_0^\infty I_\lambda(0) d\lambda} = \frac{\int_0^\infty I_\lambda(0) \tau_\lambda d\lambda}{\int_0^\infty I_\lambda(0) d\lambda}$$  \hspace{1cm} (5.2)

where $I_\lambda$ is thermal radiation intensity, $\tau_\lambda$ is transmittance and $L$ is the thickness of the foam. The transmittance of a foam sample increases as the radiation penetration through the foam increases which should result in a higher transmittance for 10 PPI foam compare to 40 PPI.
foam. The 40 PPI foam used for the fabrication of the heat exchanger was thick which the radiation from passing through the sample. In order to study the thermal radiation within the foam (independent of a foam sample thickness) and understand the decay rate of radiation intensity passing through the foam the extinction coefficient is used. Zhao, Lu and Hodson [20] defined the effective extinction coefficient as

\[ K = \frac{\ln(I(L)/I(0))}{L} = -\frac{\ln(\tau(L))}{L} \]  

(5.3)

Hsu and Howell [49] presented a formula of the effective extinction coefficient \( K \) (m \(^{-1}\)) of foam

\[ K = \frac{3(1 - \xi)}{d_p} \]  

(5.4)

where \( d_p \) is the pore size and \( \xi \) is the porosity of the foam sample. The pore size and porosity of the metallic foams were calculated previously and the results are presented in Table 5.1 [8]. As the pore size increased from 40 PPI to 10 PPI foam, the foam radiation penetration thickness increased which resulted in more heat transferred via radiation to deeper locations of the foam sample. As the pore size increased the decay rate of radiation intensity passing through the foam \( K \) and decrease in \( K \). The total transmittance calculated using equation 5.3 was 0.4 and 0.004 for 10 PPI and 40 PPI foam respectively, which indicates the importance of pore density in transmittance. Total transmittance depends on the thickness, porosity and pore size of the foam sample which in the case of 40 PPI foam is close to zero.

Table 5.1: Radiation properties of 10 PPI and 40 PPI foam.

<table>
<thead>
<tr>
<th></th>
<th>10 PPI Foam</th>
<th>40 PPI Foam</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pore size (mm)</td>
<td>2.808</td>
<td>0.673</td>
</tr>
<tr>
<td>Porosity (%)</td>
<td>0.944</td>
<td>0.938</td>
</tr>
<tr>
<td>Extinction Coefficient (m (^{-1}))</td>
<td>60</td>
<td>278</td>
</tr>
<tr>
<td>Total Transmittance (%)</td>
<td>0.5</td>
<td>0.004</td>
</tr>
</tbody>
</table>
Metal foams, like wire meshes, enhanced the performance of the bare tube heat exchanger with 10 PPI foam outperforming the 40 PPI foam as shown in Figure 5.4. Heat transfer performance increased 2.3 times for 10 PPI foam and 1.7 times for 40 PPI metal foam heat exchanger. Even though the surface area of the 40 PPI foam was greater than that an equivalent volume of 10 PPI foam, its heat transfer performance was lower.

![Figure 5.4: Coolant temperature rise comparison for metal foam heat exchangers for various flow rates, where furnace temperature is fixed at 800°C. The experimental uncertainty values are ± 1°C in the temperature measurement.](image)

To estimate the impact of the bonding resistance to the heat transfer performance, metallic foam heat exchangers were fabricated with and without brazing connection between the porous structure of the foam and the tube. The importance of the connection between the tube and foam is shown in Figure 5.5 where the 10 PPI foam heat exchanger fabricated using soldering paste performed 3.5 times better than the one without soldering paste, which shows the significant impact of the bonding method on heat transfer.
Figure 5.5: Coolant temperature rise comparison for metal foam heat exchangers for various flow rates, where furnace temperature is fixed at 800°C. The experimental uncertainty values are ± 1°C in the temperature measurement.

By comparing the performance of the best heat exchanger for each category, the 40 PPI vertical wire mesh heat exchanger was selected as the heat exchanger with maximum heat transfer as shown in Figure 5.6. The temperature rise was improved 2.8, 2.3 and 1.5 times using 40 PPI vertical wire mesh, 10 PPI foam and 2 screen 10 PPI horizontal wire mesh respectively. In terms of manufacturing process, fabricating the 40 PPI vertical wire mesh was the most time consuming process and the weight of the fabricated heat exchanger was 3 to 4 times higher than all of the other heat exchangers. As was previously mentioned, the performance of the 40 PPI vertical wire mesh heat exchanger could be improved if some of the screens were removed, which would result in greater air penetration between vertical screens.
Figure 5.6: Temperature rise comparison between heat exchangers for various flow rates, where furnace temperature is fixed at 800°C. The experimental uncertainty values are ± 1°C in the temperature measurement.

### 5.2 Heat Transfer Calculations

Heat was transferred from the furnace using fabricated metallic porous structure heat exchangers, which resulted in a temperature increase between water at the inlet and the outlet of the heat exchanger. Water enters the heat exchanger with inlet temperature $T_i$ and exits the heat exchanger at $T_o$, where the cross section of the tube is constant and the fluid is heated inside the tube as show in Figure 5.7. The boundary condition at the surface of the tube is usually approximated using constant surface temperature ($T_s =$ constant) or constant surface heat flux ($\dot{q}_s =$ constant) [32]. The constant air temperature inside the furnace was used as a boundary condition as shown in Figure 5.7. The first law of thermodynamic states that energy is conserved and therefore at steady state

$$\dot{E}_{in} = \dot{E}_{out}$$
Figure 5.7: Variation of the coolant temperature for the case of constant atmosphere temperature

\[ T_A = \text{Constant} \]

Figure 5.8: Heat transfer to a fluid flowing inside a tube.

At steady state the net rate of energy transfer from the furnace to the water inside the tube is

\[ \dot{Q} = \dot{m}c_p (T_i - T_e) \]

(5.6)
where $\dot{Q}$ is the net heat transfer to the coolant, $\dot{m}$ is the mass flow rate of the water and $C_p$ is the specific heat of the water as shown in Figure 5.8 [32]. Heat transfer to the coolant is by radiation and natural convection:

$$\dot{Q}_{\text{Total}} = \dot{Q}_{\text{Convection}} + \dot{Q}_{\text{Radiation}}$$

Heat transfer by natural convection is described by Newton’s law of cooling as

$$\dot{Q}_{\text{Convection}} = hA_S(T_s - T_\infty)$$

Where $h$ is convection heat transfer coefficient, $A_S$ is the heat transfer heated area, $T_s$ is the tube surface temperature and $T_\infty$ is the temperature of the surrounding air. The metal foams and wire meshes acted as fins to enhance the heat transfer performance of tube heat exchangers by increasing the surface area across which the convection occurs [15]. The heat transfer between the tube and the open-cell porous structure of the wire mesh or metal foam is by conduction. As the porosity of metallic foams increases the resistance to air flow decreases which results in increase of air penetration through the sample. The effective heat transfer coefficient also increases due to greater air penetration through the foam [19]. In the case of vertical wire mesh heat exchangers Fig. 3.9 wire mesh screens acted as fins and the spacing between them controlled the air penetration in-between the porous structures. There exists an optimum minimum fin spacing that maximizes the natural convection heat transfer of the heat exchanger. If the fins are closely packed as in a case of 40 PPI wire mesh Fig 3.8 (a), the heat exchanger has a higher surface area for heat transfer but lower heat transfer coefficient. In the case of widely spaced fins, the heat transfer coefficient is higher but the surface area is less [32].

Radiation heat transfer between surfaces depends on their temperature, orientation and radiative properties of those surfaces. For radiation between two surfaces

$$\dot{Q}_{12} = \frac{\sigma (T_1^4 - T_2^4)}{\frac{1}{A_1\varepsilon_1} + \frac{1}{A_2\varepsilon_2}}$$

(5.9)
where $A$ is surface area, $T$ is the surface temperatures in Kelvin, $\varepsilon$ is emissivity, and the subscripts 1 and 2 refers the heat exchanger and the furnace respectively. The Stefan-Boltzman constant $\sigma = 5.67 \times 10^{-8}$ W/m$^2$K$^4$. The shape factor $F_{12}$ is the fraction of the radiation leaving surface 1 (the heat exchanger) that strikes surface 2 (the oven walls) directly. In this study since the heat exchanger is placed inside the furnace, surface 2 completely surrounds surface 1 and therefore $F_{12} = 1$. Assuming $\varepsilon_1 = 0.15$ [47] and $\varepsilon_2 = 0.2$ [48], equation 5.9 simplifies to

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{0.85 \frac{A_1}{0.15A_1} + \frac{1}{A_1} + \frac{0.8}{0.2A_2}}$$

(5.10)

where $A_2 = 135000$ mm$^2$ was calculated using 180 mm $\times$ 90 mm $\times$ 190 mm as dimensions of the furnace. $A_1$ is the sum of the surface area of the tube inside the oven and the overall surface area of the porous structure and varies for each heat exchanger. The total area of the tube inside the furnace 7350 mm$^2$ was calculated using the overall length of the tube inside the furnace of 368 mm. To describe the surface radiation of the porous geometry for the 40 PPI foam and 40 PPI vertical wire mesh heat exchangers, the emissivity of 0.6 was estimated using equation 5.11. The effective emissivity of the foam was calculated using

$$\varepsilon_{eff} = \frac{\int_{0}^{\infty} I_\lambda(0)\varepsilon_{\lambda,eff}d\lambda}{\int_{0}^{\infty} I_\lambda(0)d\lambda} = 1 - \rho_{eff}$$

(5.11)

where $\rho_{eff}$ is the total reflectance [20]. The total reflectance was estimated using an average between the experimental data for total reflectivity of Ni-Cr (Equation 5.12)[50] and zirconia (Equation 5.13) [51, 52].

$$\rho = -1.51 + \delta(14.22 + \delta(-40.54 + \delta[54.82 + \delta(-35.5 + 8.81\delta)]))$$

(5.12)

$$\rho = 0.475 - 1.926 \delta + 5.107 \delta^2 - 4.634 \delta^3 + 1.826 \delta^4 - 0.27 \delta^5$$

(5.13)

$\delta$ is defined by $T/1000$ (K).

In the high pore density metal foams the emissivity is independent of sample thickness and occurs mainly at layers close to the outer surface of the foam. The radiation cannot reflect
outside the foam sample and is reflected and scattered within the foam [20]. The change in emissivity values can significantly affect the heat transfer performance of the system which indicates the importance of radiation at high temperature.

The heat extracted by the coolant is equal to the sum of the radiation and natural convection;

\[
\dot{m} c_p (T_i - T_c) = h A_S (T_S - T_x) + \frac{\sigma (T_i^4 - T_x^4)}{0.85 \frac{1}{A_i} + \frac{0.5}{0.15 A_i} + 0.5 A_2}
\]  \(5.14\)

The heat transfer coefficient \(h\) was calculated from equation 5.14

The enhancement of heat transfer due to the porous media \(\dot{Q}_{Fin}\) was calculated by subtracting the heat transfer of the plain tube heat exchanger from the total heat transfer of modified heat exchanger

\[
\dot{\dot{Q}}_{Fin} = \dot{Q}_{Total} - \dot{Q}_{Tube}
\]  \(5.15\)

Refer to Appendix C for the summary of the heat transfer enhancement of each porous media.

### 5.3 Heat Exchanger Surface Area

Extensive research has been done to calculate the surface area of metallic foams and to model each unit cell using dodecahedron or other idealized geometrics. Bhattacharya et al. [41], Lue et al. [43], Gani et al. [44] and Tsolas [8] have approximated the structure of the metal foam (see Figure 5.9) as cylinders oriented in such a way to produce cubic structures, as shown in Figure 5.10. Each cubic cell consists of 12 parallel and perpendicular cylinders with uniform spacing from each other.
Figure 5.9: SEM image analysis of a 40 PPI metal foam

Figure 5.10: Open-cell structure created using cylinders to represent a metal foam unit cell
They have derived an equation for the interfacial surface area which is in contact between the fluid and the solid phase, so called surface area density \( \alpha_{sf} \)

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

(5.17)

Tsolas (2010) [8] calculated the surface area density for 10 PPI and 40 PPI pore density for Copper and Nickel metal foam (Dalian Thrive Mining Co. Ltd, Dalian, China). Copper metal foams used in this study are from the same manufacturer, and same manufacturing processes were used to produce them, therefore the surface area density measurements are adopted from his research. The following numbers were used to calculate the surface area of the metal foams:

- 0.366 (mm\(^2\)/mm\(^3\)) - 10 PPI pore density copper
- 2.414 (mm\(^2\)/mm\(^3\)) - 40 PPI pore density copper.

To simplify the surface area calculation for horizontal wire mesh screens, woven wire meshes are modeled as a series of perpendicular cylinders. In a case of a single screen 10 PPI pore density copper wire mesh, there were 18 wires with 152 mm in length and 60 wires with 44 mm length which gave an overall wire length of 5365 mm. The calculated surface area was 10697 mm\(^2\) for a single 10 PPI wire mesh screen and 17097 mm\(^2\) for a single 40 PPI screen.

For the heat exchangers fabricated using vertical wire mesh screens (see Figure 3.8), the surface areas of the wire meshes were also calculated assuming they were a series of perpendicular cylinders. In the case of a single screen 10 PPI pore density copper wire mesh screen, there were 18 wires with 20 mm length and 8 wires with 44 mm length which gives an overall wire length of 715 mm. There were two holes in each screen, so after the substitution 675 mm was calculated to be the total length of the wire meshes for one screen. The surface area of single screen 10 PPI pore density wire mesh 1355 mm\(^2\) was calculated using the surface area of the cylinder. For a single screen 40 PPI woven mesh with a wire diameter of 0.3 mm where 69 wires of 20 mm and 32 wire of 44 mm long were used, the
surface area of each screen was calculated to be 2202 mm². The summary of overall surface area of the fabricated heat exchangers is presented in table 5.2.

Table 5.2: Overall surface area of the heat exchangers

<table>
<thead>
<tr>
<th>HEX #</th>
<th>Porous metal</th>
<th>Mesh pore density PPI</th>
<th>Area of the tube mm²</th>
<th>Area of the porous structure mm²</th>
<th>Over all area (A₁) mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>Foam</td>
<td>40</td>
<td>1270</td>
<td>21300</td>
<td>22600</td>
</tr>
<tr>
<td>4</td>
<td>Foam</td>
<td>10</td>
<td>7350</td>
<td>46500</td>
<td>5400</td>
</tr>
<tr>
<td>5</td>
<td>Wire mesh (1 screen)</td>
<td>40</td>
<td>7350</td>
<td>16800</td>
<td>24100</td>
</tr>
<tr>
<td>6</td>
<td>Wire mesh (1 screen)</td>
<td>10</td>
<td>7350</td>
<td>10700</td>
<td>18000</td>
</tr>
<tr>
<td>7</td>
<td>Wire mesh (2 screen)</td>
<td>40</td>
<td>7350</td>
<td>33500</td>
<td>40800</td>
</tr>
<tr>
<td>8</td>
<td>Wire mesh (2 screen)</td>
<td>10</td>
<td>7350</td>
<td>21400</td>
<td>28800</td>
</tr>
<tr>
<td>9</td>
<td>Wire mesh (Vertical)</td>
<td>40</td>
<td>1270</td>
<td>21300</td>
<td>22600</td>
</tr>
<tr>
<td>10</td>
<td>Wire mesh (Vertical)</td>
<td>10</td>
<td>7350</td>
<td>46500</td>
<td>5400</td>
</tr>
</tbody>
</table>

5.4 Discussion

An experimental investigation of heat transfer due to free convection and radiation for metallic foam and wire mesh heat exchangers at temperatures between 300°C to 800°C was carried out. It was observed that pore density, height, cell size, connection and emissivity of the porous metallic surfaces affected the performance of the fabricated heat exchangers.
After the temperature rise for different heat exchangers was measured, the overall rate of heat extraction was calculated using equation 5.6. The highest heat transfer rate achieved was 1185 W for a water flow rate of 0.025 kg/s (0.4 GPM) with an ambient temperature of 800°C using a 40 PPI mesh vertical heat exchanger. Refer to Appendix C for the summary of total heat transfer rate for all fabricated heat exchangers at 0.025 kg/s flow.

Figure 5.11: Total heat transfer rate comparison for screen heat exchangers at various temperatures for 0.025 kg/s coolant flow.

When comparing the performance of the wire mesh screens together, 2 screen 10 PPI mesh outperformed the other heat exchangers as shown in Figure 5.11. For both 10 PPI and 40 PPI horizontal heat exchangers, having two screens of wire mesh was better than one screen at high temperatures. The increase in performance of the 2-screen, 40 PPI mesh heat exchanger at elevated temperature is due to its radiation heat transfer rate and will be discussed later in this chapter. Air also penetrated much more easily through 10 PPI than 40 PPI wire mesh, since the pore sizes were much larger and that is why the 2-screen 10 PPI outperformed 40 PPI heat exchanger.
Figure 5.12: Total heat transfer rate comparison for vertical wire mesh heat exchangers at various furnace temperatures for 0.025 kg/s coolant flow.

Figure 5.13: Total heat transfer rate comparison for metal foam heat exchangers at various furnace temperatures for 0.025 kg/s coolant flow.
40 PPI vertical wire mesh heat exchanger outperformed the 10 PPI wire mesh as shown in Figure 5.12. It was seen earlier from the results for the horizontal screen heat exchangers that air penetrates better through the pores of 10 PPI than the 40 PPI wire mesh. In the case of vertical screen heat exchangers air managed to flow through and in-between screens for both wire meshes. At elevated temperatures 40 PPI vertical wire mesh heat exchanger performed better than 10 PPI which was analyzed when the total heat transfer rate was break into natural and radiation heat transfer rate.

In the case of Metal foam heat exchangers 10 PPI foam outperforming the 40 PPI foam as shown in Figure 5.13. Even though the surface area of the 40 PPI foam was greater than that an equivalent volume of 10 PPI foam, its heat transfer performance was lower.

The radiation component of the total heat transfer was calculated using equation 5.10. Natural convection heat transfer was calculated by subtracting the radiation heat transfer from the overall heat transfer of each heat exchanger using equation 5.14. By separating the radiation and natural convection from the overall heat transfer the importance of radiation heat transfer at high temperatures was observed.

The total heat flux, natural convection heat flux and radiation heat transfer and their variation based on pore density and the orientation of the porous media are discussed separately for each heat exchanger category in this section. Refer to Appendix C for the summary of the total heat flux and natural convection heat flux for each heat exchanger at various temperatures.

5.4.1 Horizontal wire mesh heat exchangers

The maximum total heat transfer for the horizontal screen heat exchangers was achieved using 2 screen 10 PPI wire mesh as shown in table 5.3. The effectiveness of the extended surface area of each fabricated heat exchanger was obtained by comparing their total heat flux.

As pore density increased from 10 PPI to 40 PPI total heat flux decreased, which indicated the relation of pore density to the heat transfer performance of the heat exchangers as shown
in Figure 5.14. Single screen heat exchangers performed better than double screen heat exchangers in the term of total heat flux. The additional screen added the same surface area as the first screen, but inhibited natural convection, which explains its reduced effectiveness. If the second screen was as effective as the first screen, both single and double screen heat exchangers would archive an identical total heat flux which was not the case. The second screen for both 10 PPI and 40 PPI enhanced the heat transfer as shown in the table 5.3 but they were not as efficient in terms of heat transfer performance as the single screen heat exchangers.

Two-screen 40 PPI wire mesh heat exchanger had the maximum radiation heat transfer of 360W at 800°C due to its high surface area as shown in Figure 5.15. The four horizontal screen wire mesh heat exchangers have the same emissivity, therefore surface area of the wire mesh played an important role in the radiation heat transfer. The radiation heat transfer increased by the factor of 1.3 and 2.6 at 800°C for 10 PPI mesh 1 screen and 2 screens respectively. Radiation heat transfer is an important factor in a high temperature environment and cannot be neglected.

Figure 5.14: Total heat flux comparison for screen heat exchangers at various temperatures
for 0.025 kg/s coolant flow.

Figure 5.15: Radiation heat transfer comparison for screen heat exchangers at various temperatures for 0.025 kg/s coolant flow.

When comparing the performance of the wire mesh screens together in terms of natural convection, 1 screen 10 PPI mesh outperformed the other heat exchangers as shown in figure 5.16. The single-screen 10 PPI wire mesh heat exchanger had the maximum natural convection heat flux of 21196 W/m² at 800°C due to its high permeability. As the pore density increases the natural convection heat flux also increase which indicated the importance of pore density in natural convection heat transfer. Both single and double
screen 10 PPI horizontal heat exchangers outperformed single and double screen 40 PPI wire mesh heat exchanger. Air penetrated much more easily through 10 PPI than 40 PPI wire mesh, since the pore sizes were much larger. High pore density wire mesh caused extra resistance to air flow which reduced natural convection heat transfer. The additional screen added the same extra surface area as the first, but inhibited natural convection, which explained its reduced effectiveness.

Figure 5.16: Convection heat flux comparison for screen heat exchangers at various temperatures for 0.025 kg/s coolant flow.
5.4.2 Vertical wire mesh heat exchangers

The comparison between the total heat flux of 10 PPI mesh vertical and 40 PPI wire mesh vertical illustrates the importance of air penetration in-between and through screens as shown in Figure 5.17. The 40 PPI wire mesh vertical heat exchanger had a larger surface area and higher total heat transfer compared to 10 PPI wire mesh heat exchanger but the screen fins were not effective. The higher total heat flux for 10 PPI wire mesh compared to 40 PPI wire mesh was due to the change in the fluid motion of air since the pore size of the wire mesh screens and the spacing between the screens differed. Wire mesh screens were positioned vertically in order to compare the performance of wire mesh to the foam of same dimension and surface area. For the 10 PPI wire mesh heat exchanger the spacing between each screens were large enough that air could travel in-between screens and through the heat exchanger but it was not the case for 40 PPI wire mesh heat exchanger. Also as the pore size of the wire mesh increased the air penetration through the screens also increased which enhanced the total heat flux as shown in the heat flux comparison of Figure 5.17.

For 40 PPI vertical wire mesh heat exchanger, the radiation heat transfer was dependent on the radiation penetration thickness. Due to the porous and compact geometry of the wire screens, the inner surface near the core of the 40PPI wire mesh heat exchanger was not directly exposed to radiation. Comparison between the heat transfer performances of both heat exchanger indicated that at 800°C, radiation heat transfer for 40 PPI vertical wire mesh heat exchanger (700 W) was higher than the radiation heat transfer of 10 vertical mesh heat exchanger (470 W). Even though the core section of 40 PPI heat exchanger was not directly exposed to radiation, it had a higher surface area than the 10 PPI wire mesh heat exchanger.

The radiation and natural convection heat transfer were investigated simultaneously in order to investigate the main source of heat transfer for both 10 PPI and 40 PPI wire mesh heat exchanger as shown in figure 5.17-5.18.
Figure 5.17: Total heat flux comparison for vertical wire mesh heat exchangers at various furnace temperatures for 0.025 kg/s coolant flow.

Figure 5.18: Radiation heat transfer comparison for vertical wire mesh heat exchangers at various furnace temperatures for 0.025 kg/s coolant flow.
The dominant effect of radiation heat transfer for 40 PPI vertical wire mesh heat exchangers is demonstrated in Figure 5.19. At 800°C furnace temperature radiation heat transfer (800 W) was 2 times more than natural convection heat transfer. At 300°C both natural convection and radiation were of the same order of magnitude but as the temperature increased, the radiation heat transfer increased more rapidly than the convection heat transfer.

On the other hand, for 10 PPI vertical wire mesh, convection dominated over radiation heat transfer as shown in Figure 5.20. The air freely penetrated in-between the screens and pores of wire meshes which resulted in higher natural convection heat transfer even at 800°C. As
the pore size increased the importance of natural convection in total heat transfer also increase.

Figure 5.20: Comparison of natural convection and radiation heat transfer for 10 PPI vertical wire mesh heat exchanger for various furnace temperature for 0.025 kg/s coolant flow.

The vertical wire mesh screens also had lower natural convection heat flux in comparison with the plain tube heat exchanger as shown in Figure 5.21. The natural convection heat flux for the 10 PPI vertical wire mesh heat exchanger was much higher than 40 PPI wire mesh: at 800°C it was 9925 W/m² for the 10 PPI and 1210 W/m² for the 40 PPI wire mesh heat exchanger. In the case of vertical screen heat exchangers air managed to flow through and in-between screens for both wire meshes. The results indicated the inefficiency of natural convection for the 40 PPI vertical wire mesh heat exchanger. Optimizing the spacing between wire meshes may produce even greater heat transfer, but that was not investigated here.
5.4.3 Metallic foam heat exchangers

The comparison between the total heat flux of 10 PPI and 40 PPI foam heat exchangers illustrated the importance of air penetration through the porous structure of foams as shown in Figure 5.22. When comparing metallic foam heat exchangers, the total heat flux of 10 PPI foam heat exchanger was higher than 40 PPI foam. The 40 PPI foam heat exchanger had higher surface area compared to 10 PPI foam heat exchanger but the effectiveness of the extended surface area of 40 PPI foam are much lower than 10 PPI foam. As the pore density of the foam increased, the air penetration through the foam decreased which explains the low heat transfer performance of the 40 PPI foam. In the case of radiation heat transfer, the
porous structure of 40 PPI foam heat exchanger performs similarly to the 40 PPI vertical mesh heat exchange where the radiation wasn’t able to completely penetrate through the foam structure.

Figure 5.22: Total heat flux comparison for vertical wire mesh heat exchangers at various furnace temperatures for 0.025 kg/s coolant flow.

In the case of convection heat flux 10 PPI metal foam outperformed the 40 PPI foam as shown in Figure 5.23. Natural convection heat flux at 800°C was 12798 W/m² for the 10 PPI foam and 547 W/m² for the 40 PPI metal foam heat exchanger. The 40 PPI foam, just like 40 PPI vertical wire mesh heat exchanger, did not enhance natural convection heat transfer and its large surface area was not effective in promoting natural convection enhancement.
5.4.4 Nondimensional parameters

The heat transfer performance of the fabricated heat exchangers were presented in the form of nondimensional parameters $Nu (L)$ and $Ra (L)$ defined as

$$Nu = \frac{h L}{k} \quad (5.16)$$

where $L$ is the characteristic length which is equal to 20 mm and $k$ is the thermal conductivity of the fluid. The Nusselt number was correlated as a function of the Rayleigh number, which is the product of the Grashof and Prandtl numbers:

$$Ra = Gr Pr = \frac{g \beta (T_s - T_x) L^3}{\alpha \nu} \quad (5.17)$$
where \( \nu \) is kinematic viscosity, \( \beta \) is the coefficient of volume expansion and \( \alpha \) is thermal diffusivity.

All of the heat exchangers were compared based on monitoring the variation of \( Nu \) and \( Ra \) as shown in Figure 5.25-27. Nusselt number increased while Raleigh number decreased as the ambient temperature increased for all the fabricated heat exchangers. The decrease in Raleigh number was due to the rapid increase in kinematic viscosity and thermal diffusivity of air as temperature increased from 300°C to 800°C as shown in Figure 5.24.

![Figure 5.24: kinematic viscosity and thermal diffusivity of air to at various temperatures.](image)

The denominator on the right hand side of Equation 5.17 increases much faster than the numerator as the temperature increases, which results in a decreased Rayleigh number. As the air temperature increased \( L, g \) and \( T_\infty \) stayed constant while all the other variables changed.

In the case of horizontal wire mesh heat exchangers, all the heat exchangers had higher Nusselt numbers in comparison with the plain tube heat exchanger except the 2 screen 40
PPI wire mesh heat exchanger as shown in Figure 5.25. For the 10 PPI wire mesh heat exchanger, adding the second screen to the heat exchanger enhanced the Nusselt number proving that the second screen was effective and did not reduce or block the natural convection heat transfer. The second screen was effective in that it not only enhanced the radiation heat transfer due to its extended surface area but it also enhanced the natural convection heat transfer of the system. The porous structure of the 40 PPI wire mesh stopped the air penetration through the heat exchanger and blocks the natural convection heat transfer which explains its low Nusselt number.

Figure 5.25: Nusselt number as a function of Rayleigh number for horizontal wire mesh heat exchangers
The performance of the heat exchanger was enhanced when the second screen was added to the system, see Fig. 5.2, while the Nusselt number decreased as shown in Figure 5.25. The second screen entirely reduced the natural convection heat transfer, but since it enhanced the radiation heat transfer, the 2 screen 40 PPI heat exchanger performed better than the 1 screen 40 PPI heat exchanger.

For the case of vertical wire mesh heat exchangers both, 10 PPI and 40 PPI heat exchangers resulted in higher Nusselt number compared to the plain tube heat exchanger as shown in Figure 5.26. The spacing between the screens of 10 PPI wire mesh heat exchangers was much more than 40 PPI heat exchanger and air penetrates much easier through 10 PPI wire mesh which resulted in higher Nusselt numbers.

Figure 5.26: Nusselt number as a function of Rayleigh number for vertical wire mesh heat exchangers
The calculated Nusselt number of the 10 PPI metal foam heat exchanger was much higher than 40 PPI metal foam as shown in Figure 5.27. As pore density increases, the air penetration through the foam decreases which resulted in the undesirable reduction of natural convection. Similar to 2 screen 40 PPI wire mesh heat exchanger, the 40 PPI metal foam resulted in lower Nusselt number in comparison with the plain tube heat exchanger which indicated the inefficiency of the foam as a natural convection enhancer for heat exchanger applications.

The enhancement in heat transfer for 40 PPI foam heat exchanger, see in Fig. 5.4, was mainly due to radiation heat transfer.

![Figure 5.27: Nusselt number as a function of Rayleigh number for metal foam heat exchangers](image)
Chapter 6
Wire-Arc Thermal-Sprayed Gas to Liquid Heat Exchanger

6.1 Introduction

Metal foams offer a method of enhancing heat transfer in heat exchangers due to their large surface area to volume ratio and high thermal conductivity. Researchers are trying to improve the mechanical properties and also increase the oxidation temperature of foams by alloying. Currently there are no metal foams in the market that operate at temperatures above 1000°C. As shown in section 2.4, nickel-coated copper foam heat exchangers oxidize if not cooled continuously during operation. Wire meshes may not have as large a surface area to volume ratio as foams but are available in a much wider variety of materials. In this study 304 stainless steel wire mesh was used as the porous structure for the stainless steel tube heat exchanger since it is much more resistant to oxidation.

Brazing and welding have been used for many years to joint metallic substrates together. Brazing is expensive since it requires a vacuum furnace to heat the substrate. In this study spot welding and tungsten inert gas (TIG) welding were investigated to connect wire mesh to the tubes’ outer surface area. In both cases the results were not satisfactory and the wire was not connect to the tube. For successful welding both parts should melt at the same time which was not the case. The wire mesh melts and evaporates much faster than the tube as seen in Figure 6.1. Better welded connections were achieved using larger wire mesh diameters but welded wire meshes with large wire diameter are only commercially available with small pore density, 1 PPI which was not suitable for this study. There is a need for more efficient and economical method of connecting the wire mesh to the tube since heat transfer depends on a good bond between the two.
wire arc spray coating is a technique to deposit metallic coatings on substrates as shown in Figure 6.2. In this technique two electrically conductive wires are fed into a spray gun, which then generates an arc by applying a voltage between their tips. The arc melts both wires and a jet of compressed air is used to atomize the liquid metal and accelerate molten particles toward the substrate to be coated. The accelerated particles solidify after hitting the substrate and form a coating. Other thermal spray techniques such as high velocity oxy-fuel spraying and plasma spraying are available commercially to create dense coatings. In this study wire arc spray was used to provide an intimate bond between the wire mesh and the tube for heat exchanger applications due to its low cost and ability to produce thick coatings.

Figure 6.1: 25 mm diameter tube welded to 0.7 mm thickness wire mesh.
6.2 Fabrication of the Heat exchanger

6.2.1 Thermal Spray Coating Deposition

There are different types of wire meshes currently available in the market, among those are woven and welded wire meshes. In this study welded wires were investigated, since wires are metallurgically bonded to each other rather than just overlapping, and therefore welded wires have higher thermal conductivity than woven mesh.

The following points are important in order to achieve a good connection while connecting the mesh to the tube using wire arc:

1. Diameter of the tube
• As the diameter of the tube increases, its surface area also increases; therefore more mesh can be bent or wrapped around the tube giving higher contact area between both surfaces)

2. Thickness of the wire mesh
• As the thickness of the wire mesh increases, it causes the mesh to become stiffer and much harder to bend, but easier to weld.

3. The surface of the tube
• The surface of the tube should be well sand blasted to enhance mechanical interlocking between the coating and the tube. The joint between them is not metallurgical and is strongly dependent on mechanical bonding. To achieve a good connection, the tube outer surface and the wire mesh needs to be roughened by sand blasting and also be free from oxide.

4. Length of the samples
• Lift-up is usually seen during the spraying process. It occurs when one side of the sample is connected, but on the other side the mesh is lifted from the tube after the spraying process. When using small samples shorter than 4”, there is usually no lift-up, but as the length of the tube increases lift up also increases. The cause of lift-up could be the thermal expansion of the substrate due to temperature gradients during the thermal spraying process.

5. Clearance between samples
• Both tube and the wire mesh must be in close contact before spraying. As the clearance between them decreases the connection resulting from spraying has higher strength.

The most economical and simplest way to fasten the mesh to the tubes prior to spraying was found to be by wrapping wires around the mesh and the tube as shown in Figure 6.3 (a). Wire mesh and tube were sand blasted separately prior to the wire fastening process, and again after the fastening process.
The wire mesh was folded around the tube before it was tied on as shown in Figure 6.2 (b). For 9.52 mm (0.375 in.) tube diameter as shown in Figure 6.2, with a 4 PPI wire mesh, it was found that bending the mesh so that 7 wires were in contact with the tube gave the best results.

(a) Side view

(b) Front view

Figure 6.3: Mechanical bonding between 4 PPI wire mesh and the tube. (a) 3 Wire fasteners were use enhance the connection between both substrates. (b) 7 wires were connected to the tube.
Stainless steel was sprayed using wire arc on the sample shown in Figure 6.4 where a 4 PPI stainless steel wire mesh with 0.8 mm wire diameter was fastened on a 19 mm outer diameter, 152 mm long stainless steel tube. A dense coating was deposited on the sample as shown in Figure 6.4 (a). Figure 6.4 illustrates a cross section of the coated sample after the wire arc spraying process. The wires were surrounded by the sprayed stainless steel coating with strong mechanical bonding between the coating, tube and the wire mesh. Magnified images of the wire and coating are shown in Figure 6.5.

Figure 6.4: Thermal sprayed sample (a) Top view. (b) Cross section view.
The sand blasting process was effective since the coating was well connected to the wire mesh and the coating was not lifted from the wire mesh surface. Applying thermal spray coating underneath the wire mesh, where it sits on the tube, was found to be a challenge and void spaces were left in-between after the coating process. Stainless steel coating was well connected to the surface of the tube as shown in Figure 6.6 and Figure 6.7 for the two surfaces marked as A and B in Figure 6.4 (b).
Figure 6.6: Connection between stainless steel thermal sprayed coating and the tube, for the marked surface A of Figure 6.4 (b)

Figure 6.7: Connection between stainless steel thermal sprayed coating and the tube, for the marked surface B of Figure 6.4 (b)
By using wire arc spraying technique it was possible to get a dense stainless steel coating on the stainless steel tube and connect both parts, tube to the wire mesh. A complete list of all tested samples can be finding in Appendix D.

### 6.2.2 Wire Arc Thermal Spray Coated Heat Exchangers

Six stainless steel (SS) metal heat exchangers (HEX) were fabricated using thermal spray technique and for all case 304 SS was chosen for the tube and the wire mesh. In all five cases the length, width and pore density (4 PPI) of the mesh was the same but different connection and orientation methods were examined to study and investigate the optimum connection method for enhancing the heat transfer of the HEX. For all the fabricated heat exchangers, 4 PPI stainless steel wire mesh sheets were cut to the dimensions of 152 mm (6 in.) × 76 mm (3 in.) \((L \times W)\). Stainless steel tubes with overall length of 584 mm and an outer diameter of 6 mm were used for the fabrication of the heat exchangers. Each tube was completely bent using a 6 mm tube bender until both sides were parallel with 22 mm spacing between them as shown in Figure 6.9.

A summary of the fabricated stainless steel heat exchangers is presented in Table 6.1. For the first two heat exchangers, 3 wires of the wire mesh were fastening on the tube using a stainless steel wire fastener. Both heat exchangers were fabricated using 4 PPI pore density welded wire mesh with the same wire orientation as Fig 6.8 \((a)\) but different wire diameter. For the third heat exchanger the wire mesh orientation was changed in order to compare different wire orientations and determine which orientation results in higher heat transfer as shown on Figure 6.8 \((b)\). For the fourth and fifth heat exchangers wire meshes were not bent around the tube to increase the contact area between both parts; instead the sheets were placed horizontally on the tube. Heat exchanger number 6 & 7 were plain tube heat exchanger whereas heat exchanger number 6 was also thermal sprayed and a thin layer of stainless steel coating was applied on it.
Table 6.1: The orientation of the wire mesh on a single tube for all heat exchangers

<table>
<thead>
<tr>
<th>HEX #</th>
<th>Wire Mesh Diameter (mm)</th>
<th>Schematics</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>0.8</td>
<td><img src="image1" alt="Schematics" /></td>
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<td>(Transverse)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.7</td>
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<tr>
<td>(Transverse)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>0.7</td>
<td><img src="image3" alt="Schematics" /></td>
</tr>
<tr>
<td>(Longitudinal)</td>
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<td></td>
</tr>
<tr>
<td>4</td>
<td>0.7</td>
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<tr>
<td>(Transverse)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>0.7</td>
<td><img src="image5" alt="Schematics" /></td>
</tr>
<tr>
<td>(Transverse)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>N/A</td>
<td><img src="image6" alt="Schematics" /> Stainless steel coated</td>
</tr>
<tr>
<td>7</td>
<td>N/A</td>
<td><img src="image7" alt="Schematics" /></td>
</tr>
</tbody>
</table>
Figure 6.8: Wire mesh orientations on the tube. (a) Transverse. (b) Longitudinal.
The following is an outline of the procedure for bonding wire mesh to tubes:

1. Before a thermal sprayed skin can be applied on both surfaces, both parts need to be sand blasted before and after they are fastened together using the wire.

2. A stainless steel metallic skin (Metcoloy 2 alloy wire, Sultzer Metco, Westbury, NY) is deposited on the sand blasted tube and wire mesh using a ValuArc (Sulzer Metco Inc., Westbury, NY) electric wire arc spray process. Table 6.2 lists the spray process parameters utilized for each material.

Table 6.2: Wire Arc thermal spray parameters for deposition of stainless steel coating

<table>
<thead>
<tr>
<th>Gun</th>
<th>ValuArc</th>
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<tr>
<td>Wire Feed Rate (m/min)</td>
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</tr>
<tr>
<td>Voltage (V)</td>
<td>31</td>
</tr>
<tr>
<td>Inlet Pressure (psi)</td>
<td>85</td>
</tr>
<tr>
<td>Air Flow Rate (SCFM)</td>
<td>60</td>
</tr>
</tbody>
</table>

These parameters control the porosity, oxide content and several other material and mechanical properties of the sprayed coating.

Figure 6.9 illustrates a picture of a heat exchanger after the spraying process in which wires were fully connected to the tube surface along the length of the tube. This was HEX 3 in the Table 6.1 and the wire orientation is the same as Figure 6.7 (b).
For the heat exchanger 4 & 5 shown in Figure 6.9 (a) & (b) the wire mesh was not wrapped around the tube and the connection between both substrates was minimum. In these last two cases, more pores were exposed, meaning more extended surface area was used to increase the heat transfer of the HEX. The manufacturing processes of these two heat exchanger were much easier due to the fact that bending of the wire mesh was not necessary.

Figure 6.9: Sample heat exchanger fabricated and prepared using thermal spray coating
Figure 6.10: Sample heat exchanger fabricated and prepared using thermal spray coating
Since the stainless steel tube of the fabricated heat exchangers was long and could not fit inside the lab furnace, a modification was made to the design and the tube was bent 90° as shown in Figure 6.11.

![Stainless steel heat exchanger after the modification](image)

Figure 6.11: Stainless steel heat exchanger after the modification

### 6.3 Heat Transfer Performance

The performance of the fabricated stainless steel heat exchangers were investigated for four different water flow rates 0.013 – 0.032 kg/s (0.2 - 0.5 GPM) and at seven different temperatures ranging from 300°C to 900°C. To ensure that steady state was reached during the experiment, the water cycle was operated for 15 minutes for each furnace temperature and flow rate. The experiments were performed at steady state and readings were taken when the thermocouple outputs had stabilized. The performance of all heat exchangers were compared base on; the temperature rise between the inlet and the outlet flow, and the total heat flux.

All fabricated stainless steel wire mesh heat exchangers outperformed the tube heat exchanger, which indicated the effectiveness of porous fins as heat transfer enhancers. When comparing the effect of wire mesh wire diameter to the heat transfer by comparing HEX 1 and HEX 2, HEX 1 performed slightly better due to it thicker wire mesh as shown in Figure 6.12.
The HEX 1-3 as indicated in the table 6.1 outperformed all the other heat exchangers which show the importance of good connection between the wire mesh and the coolant tube. Since the mesh struts were in complete contact with the tube the heat transfer performance was enhance. The Transverse orientation performed better than longitudinal orientation which indicated the importance of the wire position of the tube as shown in the Figure 6.13.

Figure 6.12: Coolant temperature rise for HEX 1, 2 & 7 for various flow rates, where furnace temperature increased from 300°C to 900°C. The experimental uncertainty values are ± 1.0°C in the temperature measurement.
Figure 6.13: Coolant temperature rise for HEX 2, 3 & 7 for various flow rates, where furnace temperature increased from 300°C to 900°C. The experimental uncertainty values are ± 1.0°C in the temperature measurement.

For stainless steel heat exchangers, having two screens of wire mesh was better than one screen. For 0.013 kg/s coolant flow rate, the heat transfer performance increased 34 percent when adding a single 4 PPI screen stainless steel mesh to the tube heat exchanger, 62 percent when the second screen was added as shown in Figure 6.14. The additional screen added the same extra surface area as the first, but inhibited natural convection, which explained its reduced effectiveness. The manufacturing process of these two heat exchangers was much easier than all the other stainless steel heat exchangers since there was no need to wrap the wire mesh around the tube.
Figure 6.14: Coolant temperature rise for HEX 4, 5 & 7 for various flow rates, where furnace temperature increased from 300°C to 900°C. The experimental uncertainty values are ± 1.0°C in the temperature measurement.

Figure 6.15: Coolant temperature rise for HEX 6 & 7 for various flow rates, where furnace temperature increased from 300°C to 900°C. The experimental uncertainty values are ± 1.0°C in the temperature measurement.
The comparison between the coated and uncoated tube heat exchangers indicated that the coating layer did not affect the heat transfer performance of the tube heat exchanger as shown in Figure 6.15.

The effectiveness of the extended surface area of each fabricated heat exchanger was obtained by comparing their total heat flux (W/m²), calculated by dividing the heat transfer rate by the surface area. The total heat flux for the first three heat exchangers were similar and the maximum total heat flux 5743 W/m² was achieved at 900°C for HEX 2 as shown in Figure 6.16.

Figure 6.16: Total heat flux comparison for HEX 1, 2 & 7 for various temperatures
It was found in Chapter 5 that the single screen heat exchangers performed better than double screen heat exchangers in terms of total heat flux. The extended surface area provided by the second screen was not as efficient as the first screen as shown in Figure 5.11. The 10 PPI and 40 PPI pore density of the wire mesh used in Chapter 5 were higher than the 4 PPI stainless steel wire mesh used in this chapter. The addition of the second wire mesh for HEX 5 did not reduce the total heat flux as shown in Figure 6.17. The pore density of the wire mesh was only 4 PPI, which did not stop the air penetration through the second wire mesh screen. Both heat exchangers had similar total heat flux, which indicates that the second wire mesh screen was as effective as the first screen. If the wire mesh pore size is not large enough, the disturbance to the air flow could affect the efficiency of downstream fins and reduce the effectiveness of the extended surface area.

Figure 6.17: Total heat flux comparison for HEX 4 & 5 for various temperatures.
Chapter 7
Conclusions

This research was undertaken to study and compare heat transfer enhancement due to natural convection and radiation to metal foams and wire meshes of 10 and 40PPI for different heat exchanger orientations, coolant flow rate and surrounding air temperature. The following conclusions resulted from this study.

1. Electroless metal coating technique can be successfully used to apply nickel coatings on copper foams in order to reduce corrosion and increase operating temperature.

2. Copper foam and wire mesh can be bonded to copper tubes using brazing to make air-to-water heat exchangers.

3. The performance of heat exchangers was compared by measuring the energy extracted by the coolant from the heated air surrounding each heat exchanger. It was determined that permeability of the foam or mesh attached to the heat exchanger tubes and air penetration plays an important role in natural convection through the porous structure. As pore density increased, less air penetrated through the porous structure, which reduced the performance of the heat exchanger.

4. Fabricated heat exchangers were analyzed based on their radiation heat transfer rate. It was observed that the radiation heat transfer rate can be the dominant factor for heat transfer in some heat exchangers at elevated temperatures. The effect of radiation heat transfer rate increases dramatically as the temperature increases and cannot be ignored.

5. Nusselt number correlations as a function of Rayleigh numbers were developed for each heat exchanger to describe their heat transfer performance. Higher Nusselt numbers were achieved for low pore density heat exchangers. It was also observed that as the temperature increases, the Nusselt number increases but Rayleigh number decreases.
6. Stainless steel heat exchangers were fabricated by bonding stainless steel wire mesh to stainless steel tubes using wire arc thermal spray coating techniques. The heat exchangers were tested inside the furnace and their heat transfer performance analyzed. Significant increases in heat transfer were obtained by attaching wire mesh.

7. In the case of wire mesh heat exchangers, if the wire mesh pore size is not large enough, the disturbance to the air flow could affect the efficiency of downstream fins and reduce the effectiveness of the extended surface area.
References


Figure A.1: Heat exchanger drawing.
Figure A.2: Front view of the complete assembly of the heat exchanger inside the furnace.
Figure A.3: Top view of the complete assembly of the heat exchanger inside the furnace.
Figure A.4: Axillary view of the final assembly.
Appendix B

Temperature Data

Table B.1: Temperature results for 1 Screen 10 PPI wire mesh heat exchanger

<table>
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<th>Coolant flow rate</th>
<th>Temperature rise at different furnace temperature (°C)</th>
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<td>0.013 kg/s (0.2 GPM)</td>
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Heat exchangers surface temperature at different furnace temperature for 0.025 kg/s (0.4 GPM) flow (°C)

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Table B.2: Temperature results for 2 Screen 10 PPI wire mesh heat exchanger

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<tr>
<td>0.032 kg/s (0.5 GPM)</td>
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Heat exchangers surface temperature at different furnace temperature for 0.025 kg/s (0.4 GPM) flow (°C)

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### Table B.3: Temperature results for 1 Screen 40 PPI wire mesh heat exchanger

<table>
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<th>500 °C</th>
<th>600 °C</th>
<th>700 °C</th>
<th>800 °C</th>
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<tbody>
<tr>
<td>0.032 kg/s (0.5 GPM)</td>
<td>0.7</td>
<td>0.8</td>
<td>1.2</td>
<td>2.0</td>
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<tr>
<td>0.025 kg/s (0.4 GPM)</td>
<td>0.8</td>
<td>1.0</td>
<td>1.6</td>
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</tr>
<tr>
<td>0.020 kg/s (0.3 GPM)</td>
<td>0.9</td>
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<td>6.0</td>
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</tr>
<tr>
<td>0.013 kg/s (0.2 GPM)</td>
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<td>2.5</td>
<td>4.2</td>
<td>5.7</td>
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### Heat exchangers surface temperature at different furnace temperature for 0.025 kg/s (0.4 GPM) flow (°C)

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<th>Locations</th>
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<td>112</td>
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Table B.4: Temperature results for 2 Screen 40 PPI wire mesh heat exchanger

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<tr>
<td>0.025 kg/s (0.4 GPM)</td>
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<tr>
<td>0.020 kg/s (0.3 GPM)</td>
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Heat exchangers surface temperature at different furnace temperature for 0.025 kg/s (0.4 GPM) flow (°C)

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Table B.5: Temperature results for 10 PPI wire mesh vertical heat exchanger

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<td>500 °C</td>
<td>600 °C</td>
<td>700 °C</td>
<td>800 °C</td>
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</tr>
<tr>
<td>0.032 kg/s (0.5 GPM)</td>
<td>1.0</td>
<td>1.4</td>
<td>2.3</td>
<td>3.4</td>
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<tr>
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<td>9.2</td>
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<tr>
<td>0.013 kg/s (0.2 GPM)</td>
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Heat exchangers surface temperature at different furnace temperature for 0.025 kg/s (0.4 GPM) flow (°C)

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<td>115</td>
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Table B.7: Temperature results for 10 PPI metal foam heat exchanger

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<td>600 °C</td>
<td>700 °C</td>
<td>800 °C</td>
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<td>0.7</td>
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<td>3.5</td>
<td>5.5</td>
<td>8.0</td>
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<td>3.2</td>
<td>5.1</td>
<td>7.6</td>
<td>10.3</td>
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<td>0.020 kg/s (0.3 GPM)</td>
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<td>2.3</td>
<td>4.2</td>
<td>6.5</td>
<td>9.9</td>
<td>12.7</td>
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<tr>
<td>0.013 kg/s (0.2 GPM)</td>
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<td>2.9</td>
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<td>8.1</td>
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<table>
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<tr>
<th>Locations</th>
<th>Heat exchangers surface temperature at different furnace temperature for 0.025 kg/s (0.4 GPM) flow (°C)</th>
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<td>45</td>
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Table B.8: Temperature results for 40 PPI metal foam heat exchanger

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<td>300 °C</td>
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<tr>
<td>0.032 kg/s (0.5 GPM)</td>
<td>0.8</td>
</tr>
<tr>
<td>0.025 kg/s (0.4 GPM)</td>
<td>0.9</td>
</tr>
<tr>
<td>0.020 kg/s (0.3 GPM)</td>
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<tr>
<td>0.013 kg/s (0.2 GPM)</td>
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Heat exchangers surface temperature at different furnace temperature for 0.025 kg/s (0.4 GPM) flow (°C)

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<th>500 °C</th>
<th>600 °C</th>
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<td>247</td>
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<td>439</td>
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<td>2</td>
<td>69</td>
<td>103</td>
<td>163</td>
<td>229</td>
<td>310</td>
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<td>140</td>
<td>197</td>
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<td>370</td>
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<tr>
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<td>94</td>
<td>134</td>
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Table B.9: Temperature results for 10 PPI metal foam heat exchanger with no Sn paste

<table>
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<th>Coolant flow rate (kg/s, GPM)</th>
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<th>400 °C</th>
<th>500 °C</th>
<th>600 °C</th>
<th>700 °C</th>
<th>800 °C</th>
</tr>
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<tbody>
<tr>
<td>0.032 kg/s (0.5 GPM)</td>
<td>0.7</td>
<td>0.9</td>
<td>1.3</td>
<td>2.0</td>
<td>2.7</td>
<td>3.8</td>
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<tr>
<td>0.025 kg/s (0.4 GPM)</td>
<td>0.9</td>
<td>1.2</td>
<td>1.6</td>
<td>2.6</td>
<td>3.7</td>
<td>5.0</td>
</tr>
<tr>
<td>0.020 kg/s (0.3 GPM)</td>
<td>1.1</td>
<td>1.4</td>
<td>2.0</td>
<td>3.3</td>
<td>4.6</td>
<td>6.2</td>
</tr>
<tr>
<td>0.013 kg/s (0.2 GPM)</td>
<td>1.3</td>
<td>1.6</td>
<td>2.5</td>
<td>4.0</td>
<td>5.7</td>
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Table B.10: Temperature results for 40 PPI metal foam heat exchanger with no Sn paste

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<th>500 °C</th>
<th>600 °C</th>
<th>700 °C</th>
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<tr>
<td>0.032 kg/s (0.5 GPM)</td>
<td>0.6</td>
<td>0.9</td>
<td>1.4</td>
<td>1.9</td>
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<td>4.6</td>
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<td>1.9</td>
<td>2.7</td>
<td>4.3</td>
<td>5.6</td>
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<tr>
<td>0.020 kg/s (0.3 GPM)</td>
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<td>2.3</td>
<td>3.7</td>
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<td>7.1</td>
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<tr>
<td>0.013 kg/s (0.2 GPM)</td>
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<td>1.6</td>
<td>2.7</td>
<td>4.6</td>
<td>6.8</td>
<td>8.5</td>
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## Appendix C

### Heat transfer Data

Table C.1: Total heat transfer for all fabricated heat exchangers at 0.025 kg/s flow

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<th>500 °C</th>
<th>600 °C</th>
<th>700 °C</th>
<th>800 °C</th>
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</thead>
<tbody>
<tr>
<td>Tube</td>
<td>37</td>
<td>48</td>
<td>95</td>
<td>154</td>
<td>238</td>
<td>339</td>
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<tr>
<td>10 PPI mesh 1 screen</td>
<td>88</td>
<td>122</td>
<td>207</td>
<td>286</td>
<td>424</td>
<td>572</td>
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<td>10 PPI mesh 2 screen</td>
<td>95</td>
<td>132</td>
<td>244</td>
<td>371</td>
<td>551</td>
<td>752</td>
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<td>169</td>
<td>270</td>
<td>387</td>
<td>530</td>
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<td>159</td>
<td>281</td>
<td>434</td>
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<td>763</td>
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<td>561</td>
<td>842</td>
<td>1185</td>
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Table C.2: Natural convection heat flux for different heat exchangers at various temperatures

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Table C.3: Summary of the heat transfer enhancement due to porous media

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<th>Heat exchanger</th>
<th>300 °C</th>
<th>400 °C</th>
<th>500 °C</th>
<th>600 °C</th>
<th>700 °C</th>
<th>800 °C</th>
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<td>203</td>
<td>311</td>
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<td>378</td>
<td>561</td>
<td>762</td>
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## Appendix D

### Thermal Spray Data

Table D.1: Thermal sprayed samples

<table>
<thead>
<tr>
<th>Sample</th>
<th>Tube OD mm</th>
<th>Material</th>
<th>Wire mesh OD mm</th>
<th>Pore density PPI</th>
<th>Wire type</th>
<th>Welded</th>
<th>Wire fastened</th>
<th>Sprayed</th>
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<td></td>
<td>Copper</td>
<td>0.6 mm</td>
<td>10</td>
<td>Woven</td>
<td></td>
<td></td>
<td>X</td>
</tr>
<tr>
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<td></td>
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<td>1.6 mm</td>
<td>2</td>
<td>Welded</td>
<td></td>
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<td>4</td>
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<td>1.6 mm</td>
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</tr>
<tr>
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</table>
Appendix E

Uncertainty

This method combines the bias and precision errors to define a realistic estimate. The DAQ uncertainty based on a bias error of ± 0.7°C and precision error of ± 0.1°C is

$$
\delta e_{DAQ} = \sqrt{e^2_{bias} + e^2_{precision}}
$$

$$
\delta e_{DAQ} = \sqrt{(0.7°C)^2 + (0.1°C)^2}
$$

$$
\delta e_{DAQ} = \pm 0.7°C
$$

The thermocouple uncertainty based on a bias error of ± 0.7°C and precision error of ± 0.4°C is

$$
\delta e_{DAQ} = \sqrt{e^2_{bias} + e^2_{precision}}
$$

$$
\delta e_{DAQ} = \sqrt{(0.7°C)^2 + (0.4°C)^2}
$$

$$
\delta e_{DAQ} = \pm 0.8°C
$$

The overall uncertainty is

$$
\delta e_{DAQ} = \sqrt{e^2_{bias} + e^2_{precision}}
$$

$$
\delta e_{DAQ} = \sqrt{(0.8°C)^2 + (0.7°C)^2}
$$

$$
\delta e_{DAQ} = \pm 1°C
$$