

Powder Recycling for the Production of Pin Fin Heat Sinks by Cold Gas Dynamic Spray

By

Justin Perry

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Abstract

As a result of the rise in processing power demands of today's personal computers, water cooled pin fin heat sinks are increasingly being employed for the cooling of graphical processing units. Currently, these high performance devices are manufactured through high-cost, high-waste processes. In recent years, a new solution has emerged using the cold gas dynamic spray process, in which pin fins are directly manufactured onto a baseplate by spraying metallic powder particles through a mask. This process allows for a high degree of adaptability to different graphics processing unit shapes and sizes not achievable by any other process to date. One drawback of this process is that, as substrate sensitivity to heat and mechanical residual stresses requires the use of reduced spray parameters, there is reduced deposition efficiency, resulting in a fair portion of the feedstock powder being wasted. This work aims to demonstrate the feasibility of using powder recycling to mitigate this issue and compares coatings sprayed with reclaimed powder to their counterparts sprayed with as-received powder. The work demonstrates that cold gas dynamic spray is a highly flexible and economically competitive process for the production of pin fin heat sinks when using powder recycling even when spray parameters result in reduced deposition efficiency. The benefits of pin fins on heat transfer properties of flat plates used for graphical processing units is briefly addressed and demonstrated.

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Nomenclature

Δm_{mask}	Mass added to mask, kg
$\Delta m_{substrate}$	Mass added to substrate, kg
ΔP_{fin}	Finned area pressure drop, Pa
ΔT	Temperature difference, K
ΔT_1	Inlet temperature difference, K
ΔT_2	Outlet temperature difference, K
ΔT_{lm}	Log-mean temperature difference, K
ε_f	Fin effectiveness
η	Pump efficiency
η_f	Fin efficiency
η_o	Surface efficiency
θ	Fin base angle, °
θ_b	Fin base to ambient temperature difference, K
τ_{eff}	Viscous dissipation value
μ	Dynamic viscosity, Pa·s
ρ	Density, kg/m ³
ρ_o	Stagnation density, kg/m ³
ρ_p	Particle density, kg/m ³
A	Nozzle area at a certain distance downstream, m ²
A^*	Nozzle throat area, m ²
A_b	Exposed base area, m ²
A_{block}	Area of the heater block surface, m ²
$A_{c,b}$	Area of the fin footprint, m ²
A_{fin}	Fin area, m ²
a_p	Particle acceleration, m/s ²
A_p	Particle area, m ²
A_{tot}	Total heat transfer area, m ²

A_{unfin}	Unfinned area, m ²
B	Transverse fin base width, m
B_s	Side fin base width, m
C_{Cu}	Cost of copper, \$/kg
C_D	Drag coefficient
$C_{electricity}$	Cost of electricity, \$/kW-hr
C_{labour}	Cost of labour, \$/hr
C_{N2}	Cost of nitrogen, \$/kg
C_{pN2}	Specific heat of nitrogen, kJ/kg·K
C_{V-H}	Constant correlating fin height to traverse velocity, mm ² /s
D_{nozzle}	Linear spray distance, m
D_{fin}	Fin diameter, m
D_h	Hydraulic diameter, m
d_p	Particle diameter, m
DE	Deposition efficiency
DE_{mask}	Mask deposition efficiency
$DE_{substrate}$	Substrate deposition efficiency
E	Energy, W
EC	Total electricity cost, \$
e_v	Required pumping power, W
F	Force, N
F_D	Drag force, N
g	Gravitational constant, m/s ²
GC	Total gas cost, \$
H	Fin height, m
h	Convection coefficient, W/m ² ·K
h_s	Sensible enthalpy, kJ/kg·K
I	Unit tensor
I_1	1 st order modified Bessel function

I_2	2 nd order modified Bessel function
J	Diffusion flux term
k	Specific heat ratio
k_{Cu}	Conductivity of copper, W/m·K
k_{eff}	Effective conductivity, W/m·K
k_f	Fluid conductivity, W/m·K
k_m	Conductivity of the fin material, W/m·K
k_t	Turbulent conductivity, W/m·K
L	Length of the finned area, m
L_{dT}	Length between block temperature readings, m
LC	Total labour cost, \$
m	Fin parameter, m ⁻¹
M	Mach number
$m_{Cu,feeder}$	Mass of copper in powder feeder, kg
$\dot{m}_{N2}$	Mass flowrate of nitrogen, kg/s
m_p	Particle mass, kg
$m_{reclaimed}$	Mass of powder reclaimed, kg
$m_{sprayed}$	Mass sprayed, kg
$m_{undeposited}$	Mass of powder undeposited, kg
$\dot{m}_{water}$	Mass flowrate of water, kg/s
n	Surface normal direction
N	Number of fins
$N_{fins,W}$	Number of fins widthwise
P	Pressure, Pa
P^*	Pressure at the nozzle throat, Pa
P_{flow}	Perimeter of the flow area, m
P_o	Stagnation pressure, Pa
PC	Total powder cost, \$
PC_{mask}	Total powder cost lost on the mask, \$

$PC_{substrate}$	Total powder cost lost on the substrate, \$
$PC_{undeposited}$	Total powder cost undeposited, \$
PFR	Powder feed rate, kg/s
q	Heat rate, W
q''	Heat flux, W/m ²
$\dot{Q}_{Cu}$	Heat rate through copper, W
q_f	Heat rate through fin, W
q_{max}	Maximum fin heat rate, W
$q_{max,t}$	Maximum heat rate through fin array, W
Q_{N2}	Total heat into nitrogen, J
$\dot{Q}_{N2}$	Heat rate into nitrogen, W
q_t	Total heat rate from fin array, W
$\dot{Q}_{water}$	Heat rate into water, W
R	Ideal gas constant, J/mol-K
Re	Reclamation efficiency
R_{Cu}	Resistance of copper, K/W
R_{eq}	Equivalent resistance, K/W
R_{fin}	Resistance of fins, K/W
R_{nofin}	Resistance without fins, K/W
R_m	Resistance of the medium, K/W
R_{TC}	Thermal contact resistance, K/W
R_{unfin}	Resistance of unfinned area, K/W
Re	Reynolds number
S	Distance between fins, m
S_m	Mass source term, kg/s
S_h	Heat source term, W
T	Temperature, K
T^*	Temperature at nozzle throat, K
T_1	First temperature, K

T_2	Second temperature, K
T_b	Fin base temperature, K
T_{block}	Temperature of the block, K
$\bar{T}_{bottom}$	Temperature read at bottom the bottom thermocouple, K
t_{box}	Time in box, s
T_{gas}	Gas temperature, K
T_m	Mean flow temperature, K
$T_{N2,1}$	Inlet temperature of nitrogen, K
$T_{N2,2}$	Outlet temperature of nitrogen, K
T_o	Stagnation temperature, K
T_s	Surface temperature, K
t_{spray}	Time of spray, s
$\bar{T}_{top}$	Temperature read at bottom the top thermocouple, K
$T_{water,in}$	Water inlet temperature, K
$T_{water,out}$	Water outlet temperature, K
T_{∞}	Ambient temperature, K
UA	Thermal conductance, W/K
v	Velocity, m/s
V_{gas}	Gas velocity, m/s
$\dot{V}_{flow}$	Volumetric flow rate, m ³ /s
V_{max}	Maximum flow velocity, m/s
V_p	Particle velocity, m/s
V_T	Traverse velocity, m/s
W	Width of the finned area, m
Y	Mass fraction

1 INTRODUCTION

1.1 Background

With the surge of modern high performance computer applications, the demand for more powerful computer hardware components has drastically increased in the consumer market. No longer is this high-tech hardware reserved for large corporations, as every day personal computers (PC) are being fitted with powerful graphics processing units (GPU) in order to cope with the requirements of running high end games, as well as the advent of virtual reality [1]. This increased demand in performance comes hand in hand with a desire to maintain or reduce the overall size of PCs, leading to a reduction in the GPU footprint. As such, more power is being dissipated as heat over smaller areas, and in tighter confinements. Therefore, efficient cooling and thermal management of these parts has become a significant challenge [2], [3].

Heat sinks have been used in PCs for decades, and usually utilize forced convection as the main method of heat transfer, with fans being used to generate an airflow over the heat sink. More recently, it has become necessary to use water-cooled heat sinks, as the ability for air to absorb heat has become insufficient in many cases [4]. Furthermore, air-cooling systems are less compact than their water-cooled counterparts. Current heat sink manufacturing processes are either material inefficient or costly in terms of change over whenever the heat sink footprint is altered [5]. The former issue is especially prevalent in the machining of heat sinks where more than half of the bulk material can be wasted, while the latter is further compounded considering the frequency with which new GPUs are emerging, as heat sinks need to be adapted to new GPU footprints on almost a

monthly basis [6], [7]. This can be problematic when restrictive manufacturing processes such as die-casting and extrusion are employed. Since these processes rely on a hard-cased pattern, the whole process must be modified to accommodate any change in product design, which offers poor flexibility in a rapidly changing industry and drastically increases the production costs.

Another factor that needs to be accounted for is demand variability, as not all GPUs are sold in the same quantities. High-end GPUs may sell only thousands of units, while lower end models may sell millions. This inconsistency in demand leads to a variation in the number of heat sinks that must be produced for any one GPU. A challenge thus exists in identifying a heat sink manufacturing process that can accommodate various production volumes [8] while also remaining flexible to frequent product design changes.

Cold gas dynamic spraying (CGDS) is a well-established metallic coating process and has recently evolved into a novel additive manufacturing (3D printing) technology [9]–[13]. Through this technique, metallic powder particles are accelerated to high velocities in an inert supersonic gas stream. Upon impact with the substrate, the particles undergo rapid plastic deformation and bond to the surface to create coatings or simple structures. Recently, this process has been used to produce pin fin heat sinks for compact air-air heat exchangers [14]–[20]. This is accomplished by inserting a wire mesh between the CGDS nozzle and the metallic substrate. In so doing, an array of pyramidal pin fins can be created with geometric parameters dependent on the mesh characteristics and spray parameters.

This process presents many advantages when compared to other competing manufacturing processes, such as a low cost of operation, and limitless spray pattern

variability. However, a drawback of the CGDS process is that it can impart high residual stresses to the substrate. If the latter is sensitive to heat and mechanical residual stresses, the use of reduced spray parameters is required (lower temperature and pressure). Another consideration is that the use of certain feedstock materials in the CGDS process can cause clogging at high spray temperatures. This again limits the range of spray parameters that can be used for production. An unfortunate drawback of this is that a large portion of the powder being sprayed does not deposit on the substrate and is consequently wasted, reducing the process viability. Consequently, in order to mitigate the loss of powder when reduced spray parameters must be used, the possibility of reclaiming the un-deposited powder so that it can be recycled for use in the manufacturing of subsequent heat sinks must be explored.

1.2 Research Objectives

The motivation for this work was to explore the possibility of improving the economic viability of producing pin fin heat sinks by CGDS by exploring the possibility and benefits of using powder recycling. The goal would be to reduce the overall process cost by reclaiming any un-deposited feedstock powder from the spraying process, and reusing it in subsequent sprays. This study was done in collaboration with Ironside Engineering Inc. and the developed process could be used for the first time in retail production. In order to achieve this goal, the following steps were undertaken:

1. Copper pin fin arrays were manufactured by CGDS and a cost analysis model was developed to determine the least costly parameters for pin fin production.

2. The un-deposited powder from the production of the pin fin array heat sinks was reclaimed and analyzed in comparison to the as-received copper powder for a variety of characteristics.
3. The reclaimed powder was used to produce coatings to ascertain its sprayability. These coatings were analyzed for morphological and mechanical differences from as-received coatings.
4. The cost analysis model was then updated to include the savings associated to powder recycling.
5. The coatings and pin fin arrays were annealed in an effort to maximize their thermal conductivity, and subsequently compared to the non-annealed examples.
6. The pin fin arrays sprayed with as-received powder were tested in a dedicated rig to determine their thermal performance characteristics. These were also compared to their annealed counterparts and un-finned plates.
7. A computational fluid dynamics (CFD) model was developed in ANSYS Fluent in order to visualize and better understand the temperature distribution through the water and pin fin array heat sink in the heat transfer test rig, with the main goal being to support certain assumptions made in heat testing.

1.3 Thesis Outline

This thesis is organized into 6 chapters. Chapter 1 presents an introduction to the motivating factors for this research, as well as a brief overview of the methodology that was employed to accomplish the goals of this work.

Chapter 2 presents an up to date summary of the literature relevant to this work. The first section gives an overview of modern heat sink technologies and the underlying heat

transfer concepts that are used in their design. The second section provides an outline of additive manufacturing processes, as well as their advantages and shortfalls when it comes to the production of pin fin arrays. The third and final section summarizes the CGDS process and expands on how it can be used as an additive manufacturing technique.

Chapter 3 outlines the specific objectives of this work. This chapter provides a clear workflow with the goal of demonstrating the importance of each step in accomplishing the final goal of reducing the cost of producing pin fin array heat sinks by CGDS.

In Chapter 4, the experimental procedures and methods of characterization are enumerated and explained. This includes detailed information on the experimental materials, experimental equipment, characterization equipment, and characterization techniques used in this study to satisfy the research objectives. In this chapter the CFD model is also defined.

Chapter 5 presents the results and discussion on the final pin fin array heat sinks produced and the viability of powder recycling, as well as the results of the heat testing. This also includes the plans for testing, the CGDS parameters, the sample preparation steps, and the methods of data collection used in the heat transfer test rig, as well as the conclusions of the CFD model.

Conclusions drawn from Chapter 5 can be found in Chapter 6. Therein can also be found recommendations with regards to future work that should be undertaken to further the outlined research objectives. A complete list of references and appendices can be found hereafter.

2 LITERATURE REVIEW

This chapter provides an in depth overview of key concepts and previous research related to this work. The first section presents the important concepts in heat transfer and thermal management which are necessary to understand the use of pin fin heat sinks. In the second section, additive manufacturing processes are outlined, as well as the use of thermal spray processes for this purpose. In the final section, the CGDS process will be examined, along with relevant gas dynamics theory, and how this process can be applied to the manufacturing of pin fin heat sinks.

2.1 Heat Sinks

2.1.1 Background

Heat sinks are used in many applications with the goal of providing enhanced heat dissipation from thermally sensitive parts. In the case of electronics, heat is generated as electrical current passes through electrically resistive materials, including metallic wires, soldered connections or other electrical components. In a computer, the central processing unit (CPU) and the graphical processing unit (GPU), are composed of densely packed circuitry through which the majority of the machine's computations are performed. As such, these components draw considerable amounts of energy from the computer's power supply, and therefore generate the most heat. Extensive heat release has been shown to reduce the performance of the chips: higher temperatures lead to higher electrical resistances that slow down processing speeds [21]. Furthermore, excessive heat can cause damage to the chips through the deterioration of plastic components and even failure of soldered connections.

The purpose of a heat sink is to aid the transfer of heat from the aforementioned high temperature electrical or mechanical parts to a coolant fluid, usually air or water, thereby allowing the part to operate at sustainable temperatures. It accomplishes this by taking advantage of the heat transfer principles of extended surfaces. The term extended surface refers to the fact that these devices feature a baseplate with a series of protruding/extending shapes, called fins. The use of such extended surfaces increases the area through which convection may occur (as seen in Figure 2.1), thereby enhancing the overall heat transfer capability of the piece. As such, heat is drawn from the base into the fin material by conduction, and is then dissipated from the fin boundaries into the surrounding fluid by convection.

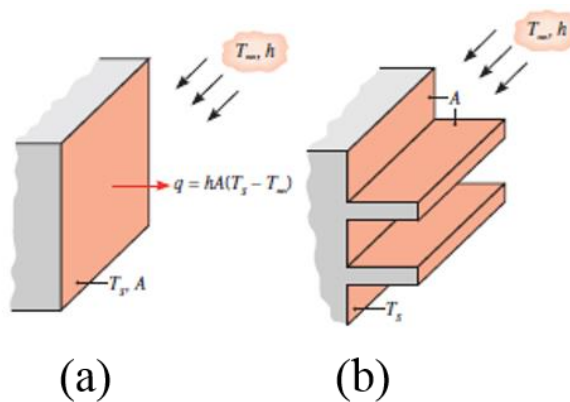


Figure 2.1: Comparison of the area (A) available to convective heat transfer between (a) a flat plate and (b) a plate with extended surfaces [22] (© 2011 Wiley Books).

Heat sinks can feature a wide variety of different fin geometries and configurations. Selection of fin geometry/configuration often depends on design constraints, such as weight, space limitations and manufacturing costs. Nevertheless, some of the most important fin design considerations include heat transfer performance (directly correlated

to the convection coefficient resulting from the addition of the fins) and the pressure losses associated with the flow of fluid over the extended surface [22]. In a heat sink there will always be substantial pressure losses, as the addition of extended surfaces into the flow causes obstruction, raising the resistance to the flow of the fluid. However, these extended surfaces are necessary to have a high heat transfer performance. Conversely, when pressure losses become too high, fluid flow slows down or is stalled, drastically reducing the convection coefficient and therefore overall heat transfer performance. Thus, the key to heat sink design is to find a balance between the performance of the extended surface and the pressure losses.

2.1.2 Heat Sink Modelling

Heat sinks are generally modelled using the concept of thermal resistances. Using this, each mode of heat transfer in the fin array is considered as a thermal resistance. As such, the heat transfer can be reduced to the convective heat transfer between the fins and the surrounding fluid, and convection between the exposed portions of the baseplate and the surrounding fluid. A representation of the equivalent system is shown in Figure 2.2.

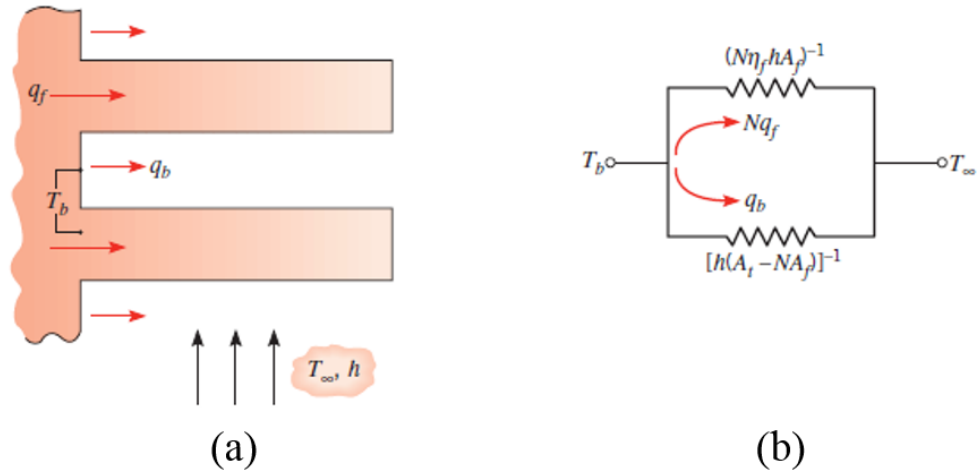


Figure 2.2: Representation of heat transfer through (a) the extended surfaces and (b) the equivalent thermal circuit [22] (© 2011 Wiley Books).

The following development is adapted from “Fundamentals of Heat and Mass Transfer” textbook by Incropera et al. [22] for the determination of heat sink thermal resistances.

The general equation for a thermal resistance is

$$R_m = \frac{T_1 - T_2}{q} \quad (2-1)$$

where R_m is the thermal resistance of the medium, T_1 and T_2 are the temperatures on either boundary of the resistance, and q is the heat rate through the resistance. Extending this principle to fins, the thermal resistance of a fin becomes

$$R_{fin} = \frac{\theta_b}{q_f} \quad (2-2)$$

where R_{fin} is the total resistance of the fin, q_f is the heat rate through the fin, and θ_b is defined as

$$\theta_b = T_b - T_\infty \quad (2-3)$$

which simply describes the difference in temperature between the fin base, T_b , and the surrounding fluid, T_∞ .

A good way to characterize the benefits of using fins is to determine the fin effectiveness (ε_f), which is defined as the heat transfer rate going through the fin compared to the heat transfer rate from the base if there was no fin (i.e. via convection directly from the base through the fin footprint area). Fin effectiveness can also be expressed in terms of thermal resistances:

$$\varepsilon_f = \frac{R_{nofin}}{R_{fin}} \quad (2-4)$$

where R_{nofin} represents the thermal resistance without the fin (simple convection from the base through the fin footprint area). This resistance is noted as

$$R_{nofin} = \frac{1}{hA_{c,b}} \quad (2-5)$$

where h is the convection coefficient over said area, and $A_{c,b}$ is the cross-sectional area of the fin at the base (fin footprint area). An effective fin should not have a higher resistance than that of the exposed base, and as such, fin effectiveness should always at least be higher than 2 for the heat sink to be worthwhile.

Fin efficiency is another useful way to represent a fin's thermal performance. The fin efficiency, η_f , can be written as

$$\eta_f = \frac{q_f}{q_{\max}} \quad (2-6)$$

where q_{max} is the heat rate from the fin under ideal conditions in which the fin is at a uniform temperature throughout its length, equal to the base temperature. As such, q_{max} may be rewritten as

$$q_{max} = hA_{fin}\theta_b \quad (2-7)$$

where A_{fin} is the total area of the fin surface. While fin efficiency serves as a measurement of a single fin's thermal performance, the overall surface efficiency may be used to characterize an array of fins, as in a heat sink. This can be written as

$$\eta_o = \frac{q_t}{q_{max,t}} \quad (2-8)$$

where $q_{max,t}$ is heat rate through the entire fin array if it were at a uniform temperature equal to that of the base, and q_t is the actual total heat rate from the array. For both the aforementioned heat rates, the surface area through which heat transfer takes place is noted A_t , which includes the area of the fin surfaces and the exposed portions of the baseplate. Therein, q_t can be determined by

$$q_t = N\eta_f hA_{fin}\theta_b + hA_b\theta_b \quad (2-9)$$

where N is the number of fins in the array, and A_b is the exposed surface area of the base.

The heat rate, $q_{max,t}$, can be written

$$q_{max,t} = hA_{tot}\theta_b \quad (2-10)$$

and the area A_{tot} is determined by equation (2-11).

$$A_{tot} = NA_{fin} + A_b \quad (2-11)$$

Furthermore, q_t may be rewritten as

$$q_t = h[N\eta_f A_{fin} + (A_{tot} - NA_{fin})]\theta_b \quad (2-12)$$

$$= hA_{tot} \left[1 - \frac{NA_{fin}}{A_{tot}} (1 - \eta_f) \right] \theta_b$$

which can be combined with equation (2-8) so that η_o may be rewritten as

$$\eta_o = 1 - \frac{NA_{fin}}{A_{tot}} (1 - \eta_f) \quad (2-13)$$

Rearranging equation (2-12) gives an expression for the thermal resistance of the entire fin array, R_{eq} , which can be expressed as

$$R_{eq} = \frac{\theta_b}{q_t} = \frac{1}{\eta_o h A_t} \quad (2-14)$$

The equivalent thermal circuit can be seen in Figure 2.3.

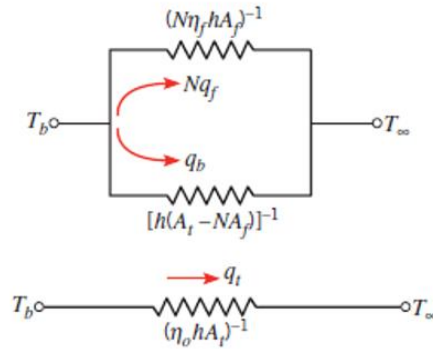


Figure 2.3: Representation of heat transfer through the equivalent thermal circuit showing the original thermal circuit (top) and the simplified version (bottom) [22] (© 2011 Wiley Books).

Fin efficiency depends on numerous parameters including fin material (conductivity), flow conditions (convection coefficient) and fin geometry. Various expressions can be obtained to evaluate the fin efficiency from these different parameters. Given that this

work deals specifically with pyramidal pin fins, only the fin efficiency for such fins is presented here. The geometry of these fins is shown in Figure 2.4.

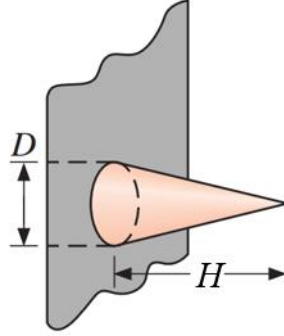


Figure 2.4: Pyramidal pin fin geometry [22] (© 2011 Wiley Books).

The value of fin efficiency was calculated using a relation found and developed in the work of Incropera et al. [22]:

$$\eta_f = \frac{2I_2(2mH)}{mHI_1(2mH)} \quad (2-15)$$

and the fin parameter, m , is

$$m = \sqrt{\frac{4h}{k_m D_{fin}}} \quad (2-16)$$

where I_1 and I_2 are the first and second order modified Bessel functions of the first kind, H is the height of the fin, h is the convection coefficient, k_m is the conductivity of the fin material, and D_{fin} is the base diameter.

2.1.3 Heat Sink Fin Configurations

Heat sinks can be divided into multiple subtypes based on the different fin shapes they have and the shape of the object they are manufactured upon. In the case of heat sinks

used for the cooling of CPUs and GPUs, the heat sink baseplates are usually flat. Common fin configurations for these applications are straight fins or pin fins, with constant or varying cross sectional areas, as shown in Figure 2.5. One of the first studies done on heat sink fin shapes was done by Kays and London [23], where a multitude of fin configurations were studied, and provided the basis for many experiments into heat sink design to this day.

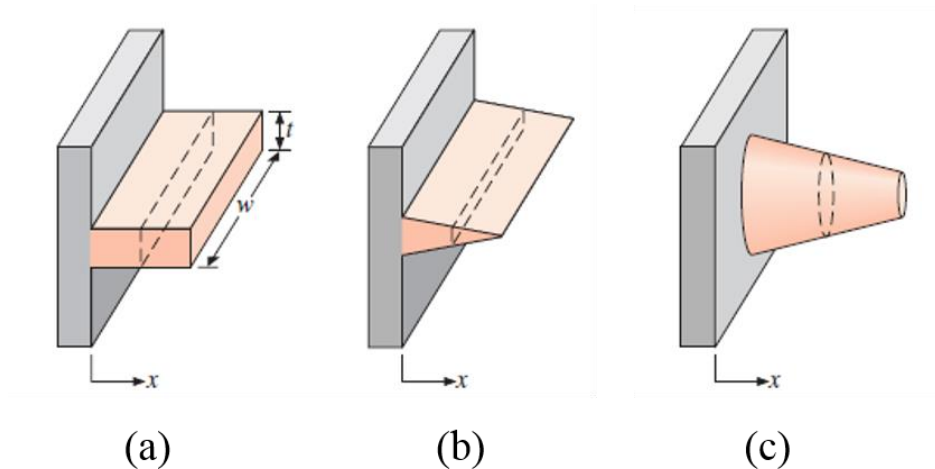


Figure 2.5: Different heat sink fin configuration on a flat baseplate: (a) uniform cross section straight fin, (b) varying cross section straight fin, (c) pin fin[22] (© 2011 Wiley Books).

In the design of heat sinks, there is an underlying incentive to reduce the volume of material used and required manufacturing costs, while still obtaining the proper heat transfer performance. Using pin fins instead of straight fins can reduce the volume of material required while maintaining cooling effectiveness. However the manufacturing of pin fins is more time consuming and costly [22]. The major advantage found when using pin fins over the more traditional straight fins is that, by having a discontinuous flow

path, the pin fins engender fluid mixing which has been proven to be beneficial to heat transfer [24]. Additionally, it has been shown that pin fin arrays are able to transfer heat more efficiently than other straight fin arrangements, although this generally comes with higher pressure losses [25], [26]. An illustration of fluid mixing in the case of pin fins is shown in Figure 2.6.

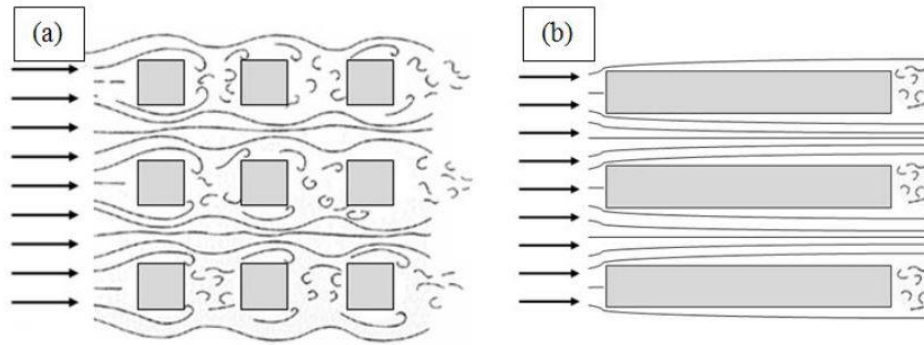


Figure 2.6: Fluid flow around (a) pin fins and (b) straight fins [17] (© 2013 Springer Nature).

The beginnings of pin fin research can be traced back to the work of Žukauskas et al. [24], where the heat transfer benefits of fluid mixing in the wake of circular tubes was demonstrated. Following this, exploration into the use of discontinuous fins instead of traditional straight fins began. Since then, there has been much research into the optimization of pin fin heat sinks for various applications, through varying the cross sectional shape of the fins.

Initial testing was done on circular fin cross sections, showing that these fins featured improved thermal performance over straight fins [27]. Many other fin shapes have been tested and compared, such as in the work of Chapman et al. where elliptical and square pin fins were compared to regular straight fins [28]. The bulk of research has gone into

finding a fin profile which reduces pressure losses, while maintaining good thermal properties. In a study done by Pandit et al. [29], diamond pin fins (square fins rotated 45° from flow direction) were compared to circular fins, and were found to give superior benefits in heat transfer. Furthermore, a comprehensive study was done by Sahiti et al. [30] where a variety of pin fin shapes, as shown in Figure 2.7, were compared in simulations for both inline and staggered fin arrangements. This demonstrated that staggered fin arrangements outperformed inline fin arrangements. This is due to the fact that the inline fins are situated directly in each other's wake, while the staggered fins are offset, causing the impact of the incoming flow to be more pronounced. It was also found that the elliptical fin profile featured the highest performance, while square fins were the least performant. Even though the square fins exhibited similar Nusselt numbers to other fin profiles (elliptical had the highest Nusselt number), the higher pressure drop associated with the square profile resulted in a lower performance on the basis of heat transfer versus pumping power [30].

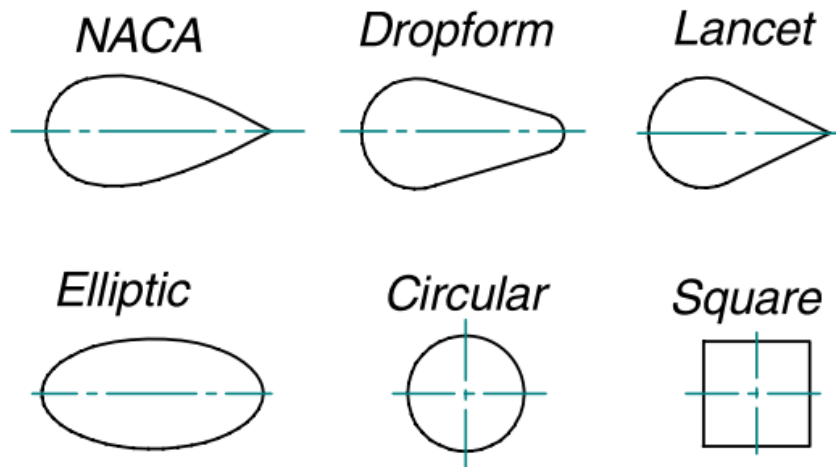


Figure 2.7: Various pin fin cross sections tested by Sahiti et al. [30] (© 2005 Elsevier).

An aspect of fin design that has not been explored in detail is the use of tapered, or pyramidal, pin fins. Investigations have been done on tapered circular pin fins, but fail to compare them to non-tapered pin fins [31], [32]. As for square base tapered fins, the only references that were found came from the works of Cormier and Dupuis [15]–[18]. They stated that the reason this shape is uncommon is most likely due to the complexity and cost of manufacturing such fins. The pin fins produced in one of their studies were tested in a cross flow of air to evaluate their heat transfer performance [17]. The results were compared to straight fins for validation, and the pyramidal pin fins were shown to exhibit far superior thermal performance.

2.1.4 Manufacturing of Heat Sinks

Heat sinks can be manufactured using many different processes, depending on the desired fin configuration and the bulk material used. Machining is a typical manufacturing technique used for the production of heat sink fins, which can produce both straight type fins and pin fins [33], [34]. By using a machining process such as end milling, shown in Figure 2.8a, it is possible to create three-dimensional shapes from a variety of metallic bulk materials. As such, machining is an attractive option for the production of heat sink fins. Since a robot controls the path of the end mill, this process is well suited to prototyping, but can become costly for mass production. Another drawback of this process is that the off-cut is lost, constituting a large loss of material [33], [34]. While these off-cuts could potentially be recycled by recasting them into a new block of bulk material, this constitutes an extensive extra processing step.

Another important process is extrusion, as seen in Figure 2.8b. In extrusion, a solid block is fed through a die and pushed into a continuous cross sectional shape by forcing material to flow under high pressure [35]. This process works well for high volumes of production, however, due to equipment costs, it is not suitable for prototyping. It also cannot be used for pin fin production, as it can only create shapes with a continuous cross section [36].

Casting, shown in Figure 2.8c, is capable of producing heat sinks with either pin fin or straight fin configurations. Casting can produce very high complexity shapes but it is a very expensive process at low rates of production due to the high cost of complex molds [33].

Forging is another popular technique for heat sink production [36]. In forging, extremely high forces are used to crush a bulk material into a desired shapes using a molding die, as shown in Figure 2.8d. The cost of equipment necessary for this process is however very high, and the level of precision attainable is relatively low, which is problematic for the fabrication of inherently small/fine features such as fins. Uneven fin heights and size limitations would be expected, leading to a need for post processing [36].

Various powder sintering techniques can also be used to produce both types of fins. One such technique is Selective Laser Sintering (SLS), seen in Figure 2.8e, whereby a fine powder is deposited onto a baseplate substrate and then selectively sintered layer by layer to produce a three-dimensional shape [33]. This process is difficult to use for mass production due to high process times, but is excellent for prototyping. A drawback of this process is that it can result in oxide inclusion and, in certain cases, porosity, effects which could potentially hinder heat transfer performance [37].

A process which seeks to eliminate the cost of expensive dies and molds is bonding. In this process, straight fins or pin fins are bonded to a baseplates using thermosetting epoxy [36], as seen in Figure 2.8e. This process is viable for low yield production, but it suffers from a considerable drop in thermal performance, as the interface material bonding the fins to the baseplate is less conductive than the bulk [36].

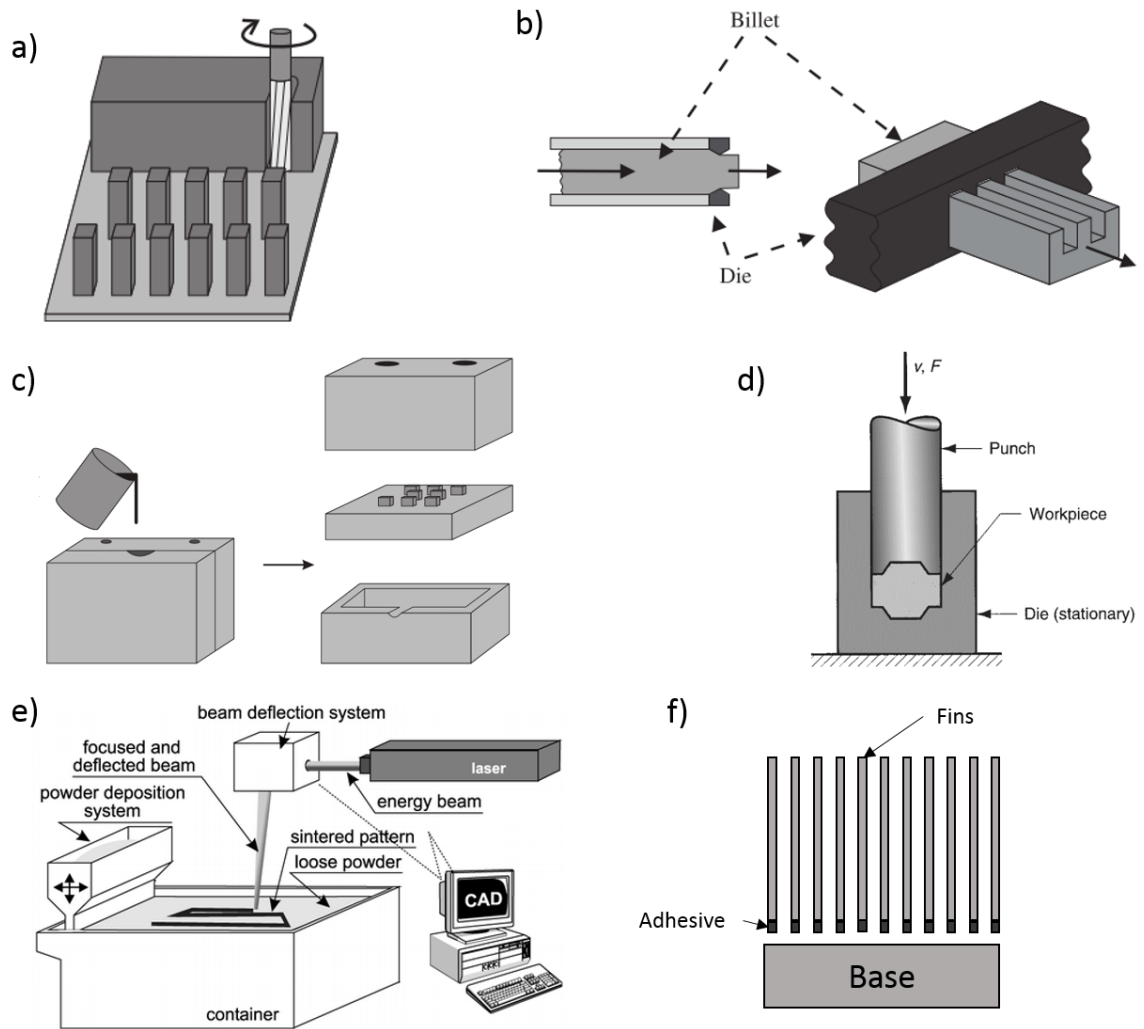


Figure 2.8: Heat sink manufacturing processes: a) machining [33] (© 2010 IEEE); b) extrusion [33] (© 2010 IEEE); c) casting [33] (© 2010 IEEE); d) forging [35] (© 2009 Wiley Books); e) selective laser sintering [37] (© 2015 Elsevier); f) bonding.

The above mentioned processes all have prohibitive drawbacks in the production of pin fin heat sinks. Whether it be through loss of material, lack of precision, reduced thermal properties, equipment cost, or the inability to cope with various production rates (prototyping vs. high volume) there is no single process that is perfectly suited to the task of producing heat sinks. Table 2.1 provides an overview of the pros and cons of each process.

Table 2.1: Overview of the Pros and Cons of Various Manufacturing Processes for Heat Sink Production

Manufacturing Process	Pros	Cons	Pin Fin Production
Machining	-Easily variable -Good for prototyping	-Wastes material -Bad for mass-production	Yes
Extrusion	-Good for mass production	-Expensive equipment -Bad for prototyping	No
Casting	-Complex shapes achievable	-Expensive equipment -Bad for prototyping	Yes
Forging	-High production rate	-Expensive equipment -Low accuracy	Yes
Selective Laser Sintering	-Complex shapes achievable -Good for prototyping	-Low production rate -Defects (porosity/oxides)	Yes
Bonding	-Low cost of equipment	-Low production rate -Low product performance	Yes

More recently, pyramidal pin fin heat sinks were produced using CGDS as an additive manufacturing technique, by Cormier and Dupuis [14]–[18]. CGDS, a process that is discussed in detail later in this work, is a type of thermal spray process used for the

production of metallic coatings. This is achieved by projecting fine powder particles at high velocity onto the surface to be coated. In order to use this technique to produce pin fin heat sinks, a wire-mesh mask is added between the CGDS nozzle and the substrate. While the mask wires effectively block off some of the projected powder particles, others flow through the square openings in the mask and form a discontinuous coating on the substrate. As the coating builds up, discrete pyramidal pin fins are produced, as shown in Figure 2.9. This process has currently only been used to produce square base pyramidal fins, and investigation has been done into the optimal size and spacing of these fins, as well as studies on different materials that can be used to produce them [14]–[20]. The economic viability of producing pin fins in this manner is an important aspect to consider that has not yet been explored in much detail. In the work of Stier et al., selective laser melting is compared to CGDS for the production of pin fin arrays using a model based on material deposition rate per hour, showing that CGDS is far superior both in terms of cost and production capacity [38]. In this work, the author also proposed that powder recycling could be used as a means to reduce the cost of pin fin production.

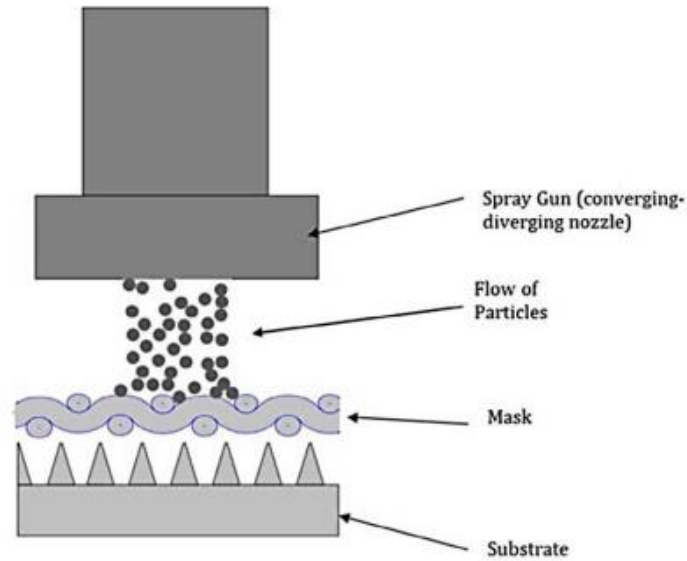


Figure 2.9: Masking technique for producing pyramidal pin fin arrays [15] (© 2014 Elsevier).

2.1.5 Heat Sink Materials

Heat sinks must be made with highly conductive materials in order to provide a low resistance path through which heat may be extracted from the sensitive part. Several materials are used to produce heat sinks, such as magnesium ($k=159 \text{ W/m}\cdot\text{K}$), copper ($k=386 \text{ W/m}\cdot\text{K}$), and aluminum ($k=204 \text{ W/m}\cdot\text{K}$) [36]. In certain high temperature applications, nickel ($k=90 \text{ W/m}\cdot\text{K}$) and stainless steel ($k=45 \text{ W/m}\cdot\text{K}$) have also been used, despite lower conductivity [39]. The decision of which material to use for a heat sink comes down to the application it will be used for, the manufacturing process, and cost.

For instance, aluminum alloys are the most common heat sink materials due to their low cost and ease of manufacturing with multiple processes [36]. However, copper, which has a significantly higher conductivity, can only be made into a heat sink by machining or bonding [36]. Copper is also heavier than aluminum and is more expensive. Thus, with

machining being one of the only ways to process copper, the wasted material in off cuts constitutes a large production cost. A less common material used is magnesium, which has been found to be the most efficient material for heat dissipation with respect to the mass of the heat sink [36]. Thus, in situations where the mass of the final product is of comparable importance to heat dissipation, it may be desirable to use magnesium instead of aluminum or copper.

2.1.6 Convective Medium

The main function of a heat sink is to transfer heat away from a high temperature source, such as heat sensitive electronic equipment, to a lower temperature fluid. This fluid is the convective medium by which heat can be drawn away from the heat sink. In most cases, the fluid flows over the heat sink so that heat transfer can take place through forced convection, as this greatly increases the efficiency of heat transfer when compared to that of natural convection in a stationary fluid [22]. In forced convection, a pressure differential is created by a fan or a pump, causing the fluid to flow over the heat sink.

The most common fluid used to cool heat sinks is air, but more frequently water, water based solutions, or nano-fluids are being used. The capacity of air to absorb heat is no longer sufficient for many applications, whereas liquids offer far superior performance [40]. Water has a much higher heat capacity than air, meaning that it can absorb more heat per unit mass than air [40]. However, the use of water near electronics runs the risk of causing a short circuit if not managed properly. Mixing water with other fluids, such as glycol, has also been done to prevent freezing in lower temperature applications [4]. More recently, there has been research into using water as a convective medium with phase transition, where the latent energy from evaporation maintains the fluid

temperature constant, aiding in heat transfer [41]. Nano-fluids are also used to further enhance heat transfer efficiency. A nano-fluid consists of a base fluid, usually water, into which very small nanometer-size particulate has been mixed. In the case of nano-fluids being used as the convective medium in heat sinks, the particles are usually high conductivity materials, such as copper, aluminum, or alumina, to enhance the thermo-physical properties of the fluid [4], [42], [43]. However, nano-fluids are more costly than water and require more maintenance. Furthermore, they can cause undesirable side effects such as particle agglomeration and deposition [4].

2.1.7 Graphics Processing Unit (GPU) Cooling Systems

GPUs constitute one of the greatest electrical loads in modern PCs, and therefore produce a large amount of heat that must be dissipated to prevent damage to the unit, and to ensure it functions efficiently. GPUs are constantly becoming more powerful with the demands of modern usage, and, coupled with an overall decrease in size, the efficient cooling of these parts has become a subject of much interest in recent years [2], [3]. Currently, most commercial GPUs use an integrated air cooling system that comes pre-assembled onto the GPU chip. This consists of a fan, which draws in cool ambient air and forces it through a heat sink mounted on top of the GPU, as seen in Figure 2.10.

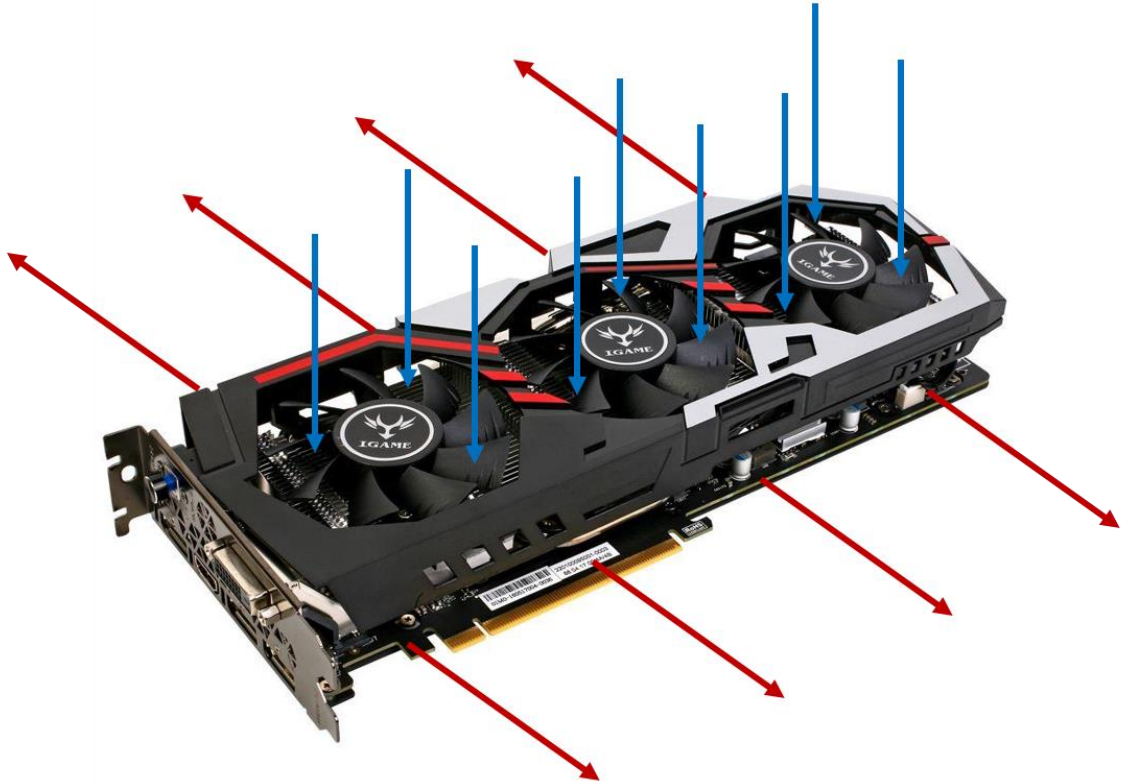


Figure 2.10: Example of GPU cooling system showing cool incoming air (blue arrows) and warm exiting air (red arrows).

The underlying heat sinks in these cooling systems consist of a combination of straight fins made from copper or aluminum plate, and heat pipes or loop heat pipes. Heat pipes are a two phase cooling device that transfer heat by evaporating a coolant liquid, which is circulated by its change in density to a cooled section and then condensed to be circulated back to the hot area [44]. Loop heat pipes are different only in that the evaporator and condenser portions are separate [45]. In the case of GPUs, the heat pipes are put into contact with the GPU processor, and contain a coolant fluid which is heated and then circulated through the length of the pipe. The heat pipes are surrounded by the straight fins, which are cooled by the air being forced over them by the fans. The coolant in the pipes is cooled and recirculated to the GPU processor, completing the coolant cycle and

providing a continuous means of heat dissipation. A schematic of the functioning of a heat pipe is shown in Figure 2.11.

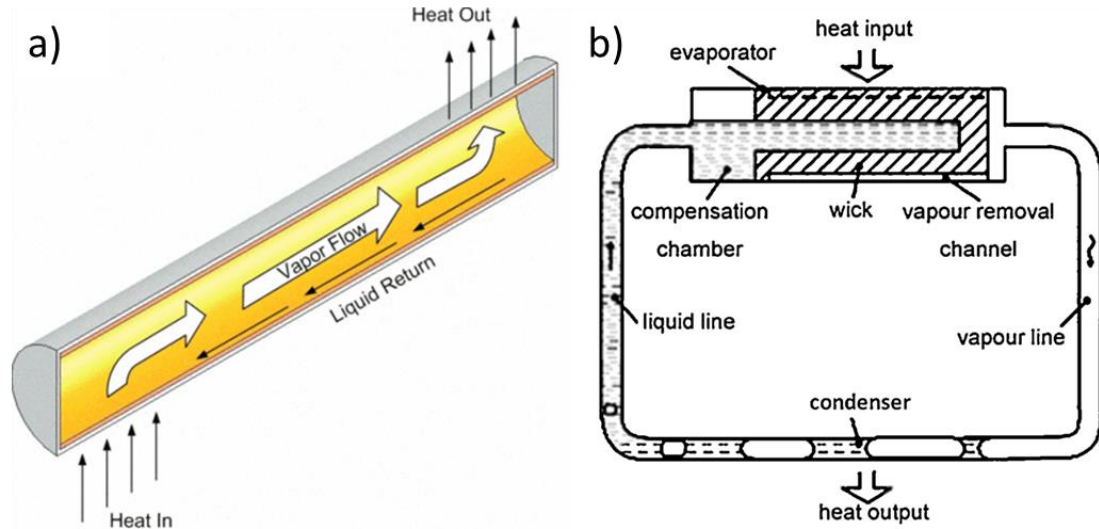


Figure 2.11: Schematic of a) a heat pipe [46] (© 2008 IEEE) and b) a loop heat pipe [45] (© 2014 Elsevier).

With increasing regularity, liquid-cooled heat sinks are being implemented in GPUs for higher performance applications, as the ability for air to absorb heat is comparatively limited [4]. In liquid cooling, a heat sink, which is in contact with the high temperature piece, is immersed in a dielectric fluid. The higher heat capacity, density and conductivity of these liquids, coupled with higher mass flow rates, allows for the absorption of far more heat per unit time than air [40], [44]. Liquid cooling systems for GPUs, as seen in Figure 2.12, have a sealed system incorporated into the PCs casing, and usually cool other systems as well, such as the CPU. The liquid is pushed throughout the system using a pump, bringing the cold liquid to the hot hardware components, whereupon it is circulated over a heat sink for heat extraction [47]–[49]. The heat sink design varies greatly, and can either be made up of straight or pin fins. The hot liquid then reaches a

radiator, which spreads the liquid through a series of winding channels equipped with straight plate fins and a fan to extract the heat from the liquid [47]–[49]. In such a system it is impossible to cool the water below ambient temperature, however there are systems that incorporate small scale refrigeration systems in order to cool the liquid further [50]. The liquid is then returned to the pump for recirculation, sometimes with a reservoir being used as an intermediary stage.

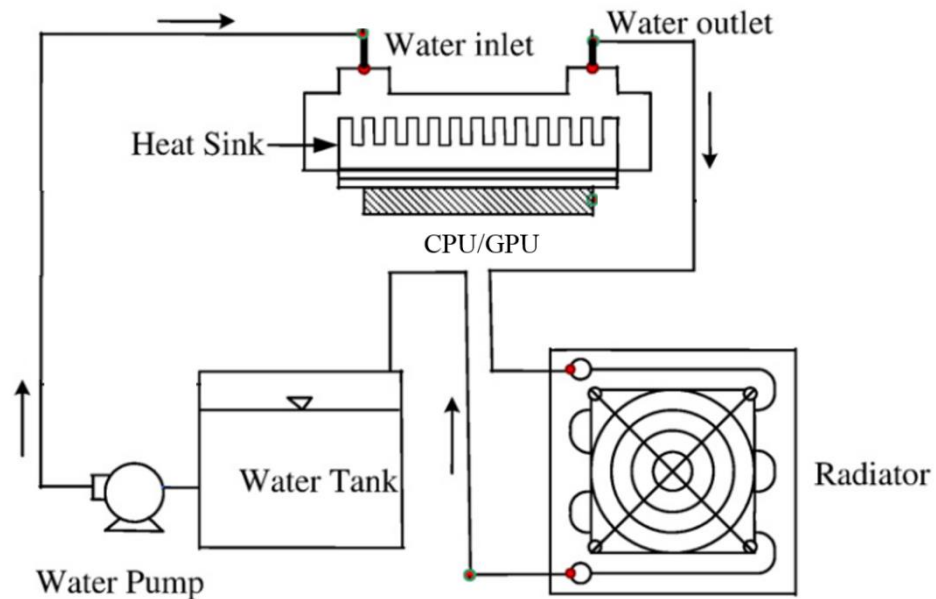


Figure 2.12: PC liquid cooling system, adapted from the work of Naphon et al. [49] (© 2008 Elsevier).

2.2 Additive Manufacturing

Additive manufacturing is an emerging branch of fabrication techniques whereby components can be produced by the addition of material in a layer based process, as opposed to forming parts with the use of molds or dies, or the subtraction of material by machining. In the case of metals, components are produced by laying successive layers of

metallic feedstock materials, either by depositing metallic powders with a binder and subsequently consolidating them, as seen in powder bed systems and powder feed systems, or by creating the final part directly from fusing a continuous stream of metal wire, as in wire feed systems [51], [52]. These processes are relatively new, and therefore only certain materials have been investigated for use as feedstock materials, mainly titanium, aluminum, and steel alloys. More recently, CGDS has emerged for use as an additive manufacturing technique through the use of masking techniques [10], [14]–[18], [51], as well as through the use of removable support structures for the production of complex shapes [11], [12].

2.2.1 Powder Bed Systems

Powder bed systems, as shown in Figure 2.13, function by raking a metallic powder over the work area, and subsequently using an energy source, such as a laser or electron beam, to sinter the powder bed into the desired shape [52]. This process is repeated layer upon layer until a solid three-dimensional object is created. The advantages of such a system are the high resolution achievable, and the ability to produce internal passages, while a disadvantage is that these units are not generally suited to building large volumes [52].

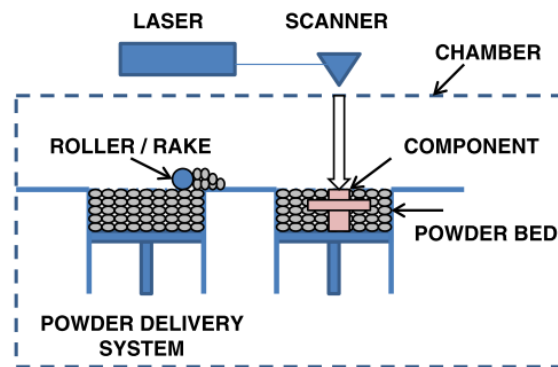


Figure 2.13: Powder bed additive manufacturing [52] (© 2014 Springer Nature).

2.2.2 Powder Feed Systems

Another powder based additive manufacturing technique is the powder feed system, seen in Figure 2.14. In this technique, a stream of metallic powder is fed through a nozzle onto the build surface, forming a pattern by moving the feed head in relation to the build surface. As with the powder bed system, a laser melts each successive layer of metallic powder, forming a three-dimensional shape [52]. This process can create larger build volumes than powder bed process and can also be used to repair damage to existing components [52].

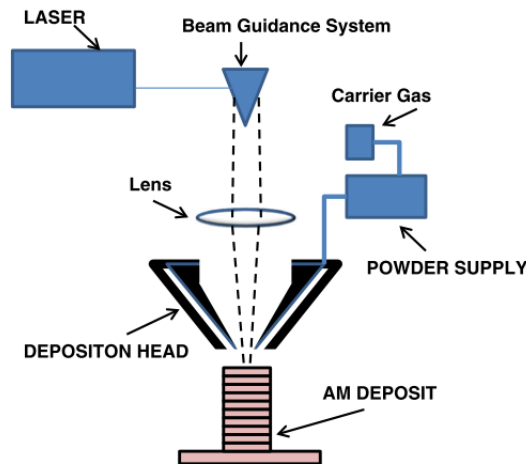


Figure 2.14: Powder feed additive manufacturing [52] (© 2014 Springer Nature).

2.2.3 Wire Feed Systems

The final type of metallic additive manufacturing system is the wire feed system, shown in Figure 2.15. In this system, a continuous metallic feedstock wire is fed into an energy source, either electron beam, laser, or plasma arc, and the molten material is then layered onto the work surface [52]. This process is suited to applications that require high

deposition rates and high build volumes, however the results of this process usually have defects that require post-process machining [52].

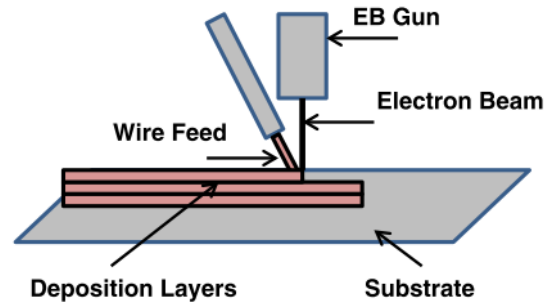


Figure 2.15: Wire feed additive manufacturing [52] (© 2014 Springer Nature).

2.2.4 CGDS as Additive Manufacturing Technique

In recent years, the CGDS process has been used for additive manufacturing applications. Thermal spray processes are generally used for the production of coatings and are generally not well suited to additive manufacturing. This is due to the relatively high levels of tensile residual stress present in the coatings, especially important when producing higher thickness deposits, which can cause failure through spalling (cracking from tensile residual stresses stemming from thermal mismatch) [53]. These tensile stresses are a result of thermal mismatch engendered by the high temperatures used in most thermal spray processes [54], [55]. However, the lower processing temperatures and higher impact velocity of incoming feedstock particles in CGDS generally result in compressive residual stresses within the produced coatings, allowing for CGDS to produce much thicker coatings than other thermal spray processes [9], [56]–[58]. This makes it a potential candidate for use as an additive manufacturing technique.

The advantages of using CGDS are that it requires no high temperature processing, as with previously discussed additive manufacturing techniques, which leads to far lower levels of oxidation [10]. CGDS also has a distinct advantage in deposition rate when compared to other additive manufacturing techniques, which can lead to higher production rates [51]. Conversely, one disadvantage of using CGDS is the low level of resolution achievable, although this problem has been mitigated through the use of micro-nozzles and masks [10].

In the work of Kim et al., CGDS was used for the production of copper electrodes on silicon wafers for solar cell applications, resulting in elongated pyramidal shapes [13]. Blose et al. showed the feasibility of rapid prototyping of a Ti6Al4V axisymmetric bulk part by CGDS [11]. Studies have also been done into spraying onto removable templates with CGDS for additive manufacturing of shell elements, as well as spraying onto removable support structures for the creation of more complex parts [12]. Another application, as previously mentioned, is the production of pin fin array heat sinks by using a masking technique [14]–[17]. Furthermore, it has been shown that producing pin fin arrays using CGDS is more economically viable than using selective laser melting techniques [38]. An in depth explanation of the CGDS process and its applications is found further in this chapter.

Other thermal spray processes generally produce much thinner coatings and contain defects which make them unsuitable for additive manufacturing applications [59]. However, while thick coatings have been made with high velocity oxygen fuel spray, there are generally high residual stresses and there can be oxide inclusion at higher

operating temperatures, again reducing its viability as an additive manufacturing process [59], [60].

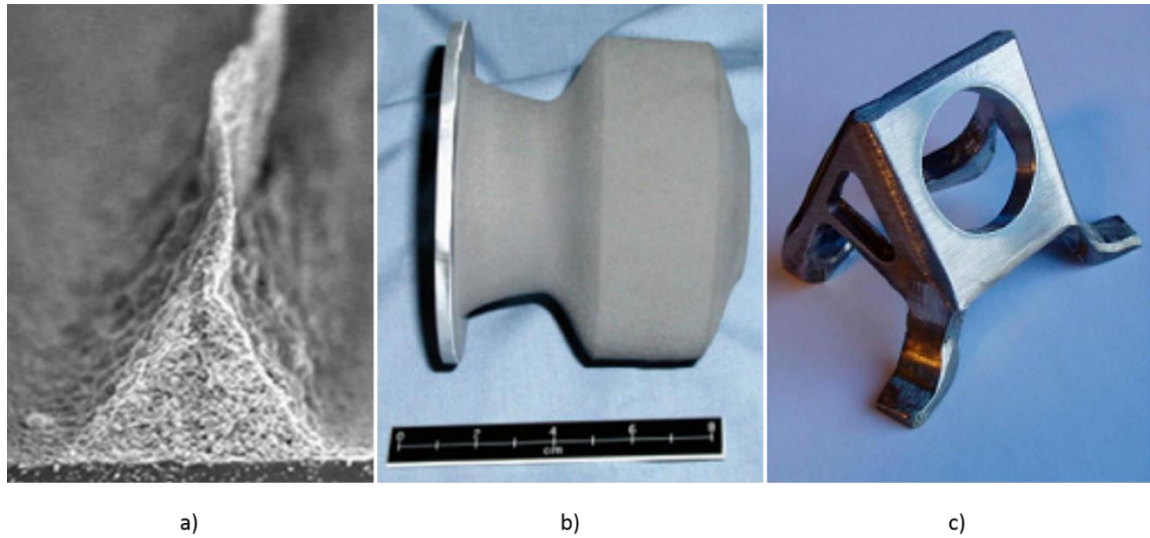


Figure 2.16: Items produced by CGDS additive manufacturing: a) copper electrode on a silicon wafer [13] (© 2013 Springer Nature); b) axisymmetric Ti6Al4V part [11] (© 2006 Elsevier); c) cold sprayed part after removal from support structure and post-process machining [12] (© 2013 Taylor & Francis).

2.3 Cold Gas Dynamic Spray

CGDS is a thermal spray process which has more traditionally been used as a coating deposition technique, but has more recently also been used as an additive manufacturing technique [11]–[16]. CGDS is a solid state process: the feedstock material does not reach its melting point during deposition. It relies on the kinetic energy of the impacting feedstock particles to create coatings [61]. In order to accelerate the feedstock material, the particles are injected into a gas stream which has been accelerated through a converging-diverging (de Laval) nozzle [62]. Particles that achieve sufficiently high velocities upon impact with the substrate will plastically deform and bond with the

substrate. An illustration of the typical CGDS system components is presented in Figure 2.17.

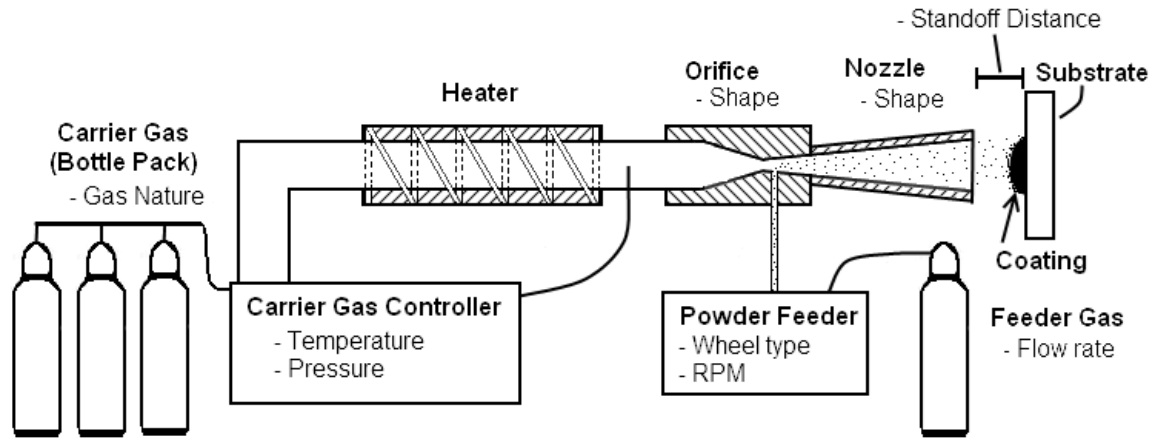


Figure 2.17: Illustration of the CGDS process (image courtesy of Patrick Trahan).

2.3.1 Background

The basis for CGDS was first proposed in 1900 by Thurston, where a blast of high pressure gas could be used to propel metallic particles onto a metal plate, causing them to become permanently embedded and thereby creating a coating [63]. However, it took many years for this process to be refined into the conventional form known today. In 1958, Rocheville filed a patent for a process that was to use a DeLaval type nozzle with high pressure air to propel fine metallic particles at higher velocity in order to form a coating. This set up was not capable of creating thick coatings on a part [63]. It wasn't until the 1980's, when Papyrin and colleagues at the Institute of Theoretical and Applied Mechanics of the Russian Academy of Science in Novosibirsk were able to deposit a variety of metals and composites onto various substrate materials [64]. This process has

evolved into a well-known commercial spray process and has been the focus of extensive research over the last decade.

2.3.2 CGDS Fundamentals

2.3.2.1 *Bonding Mechanisms*

In order to bond, a cold sprayed particle must either create metallic or mechanical links with a substrate or other previously deposited particles. For this to occur, the particles must be accelerated beyond a minimum threshold velocity known as the critical velocity. Particles travelling with speeds in excess of this velocity can deform swiftly and locally and are able to bond to the substrate. In certain cases, a phenomenon known as adiabatic shear instability can take place [65].

Adiabatic shear instability occurs when a material is subjected to extremely high strain rates. The plastic strain energy results in highly localized heat generation at impact, which has insufficient time to dissipate into the material due to the extremely short time scales associated to cold spray particle impacts. This causes a rise in temperature and material softening [66], [67], which overcomes the inherent work hardening of the material and causes the material to experience a reduction in flow stress [65], [66]. Under these conditions, the substrate and particle material flow readily to form what is known as metal jetting, shown in Figure 2.18. This figure illustrates how the particle material flows outward as a crater is formed in the substrate. Under these high shear conditions, surface oxide layers can be removed and a clean metal-metal interface can be created [68]. This intimate contact, coupled with the high contact pressures experienced in cold spray, promotes the likelihood of metallurgical bonding [66].

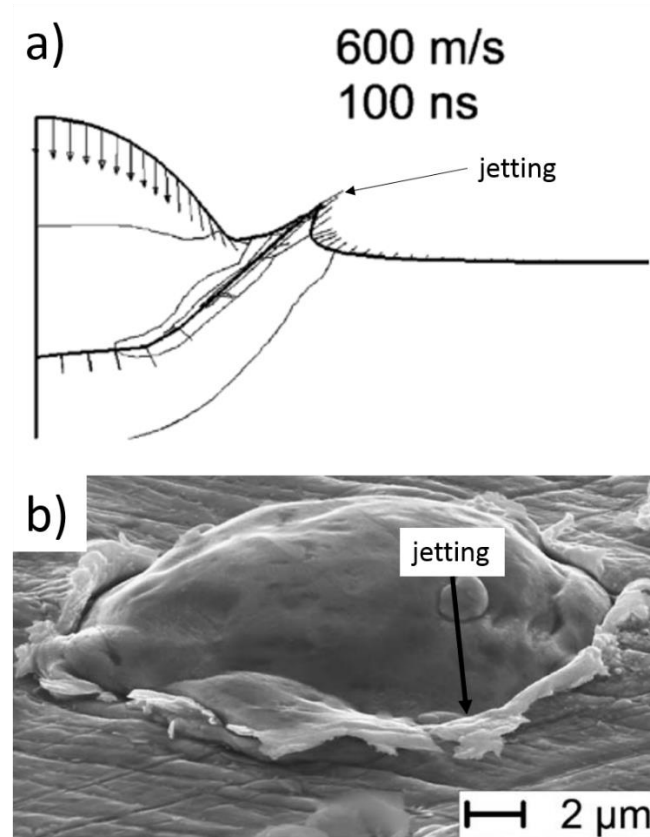


Figure 2.18: Example of jetting seen in cold spray, shown in a) the results of simulation and b) a copper particle on copper substrate [65] (© 2003 Elsevier).

The severe deformation experienced in jetting can also result in mechanical interlocking of the particle and substrate. This type of mechanical bonding can only occur when both the particle and the substrate experience severe deformation [67]. An example showing particles and substrate mechanically interlocked can be seen in Figure 2.19.

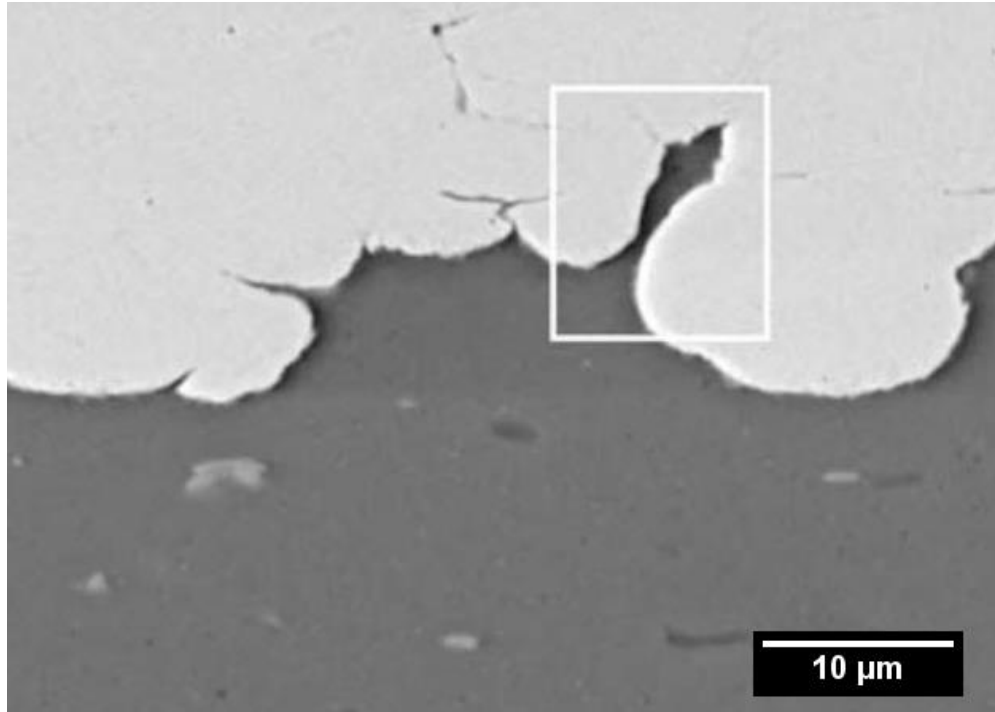


Figure 2.19: High magnification image of a copper coating on an aluminum substrate showing an area where the substrate has been deformed in between particles [67] (© 2009 Springer Nature).

In order to improve the conditions for mechanical bonding on the substrate, grit blasting can be done as a way to clean the surface and prepare it for deposition. The substrate is thus pre-deformed, aiding in the formation of points where mechanical anchoring could potentially occur [69], [70]. This is, however, a material dependent result, as it has been shown that grit blasting for certain material combinations can in fact hinder bond strength [71]. This is because grit blasting changes the mode of adhesion to mechanical anchoring, in essence hindering the potential for metallurgical bonding. Thus, for material combinations where the main mode of bonding is metallurgical, grit blasting can reduce adhesion strength [70].

However, after the initial layer of deposition, the particles are no longer experiencing impact with the substrate, but with other previously deposited particles. Therefore, in order for bonding to occur, mechanical interlocking and/or metallurgical bonding must now take place between particles [72]. This phenomenon is shown by the etched samples in Figure 2.20, where it is seen that the particle boundaries have disappeared in certain locations, indicating that metallurgical bonding has occurred.

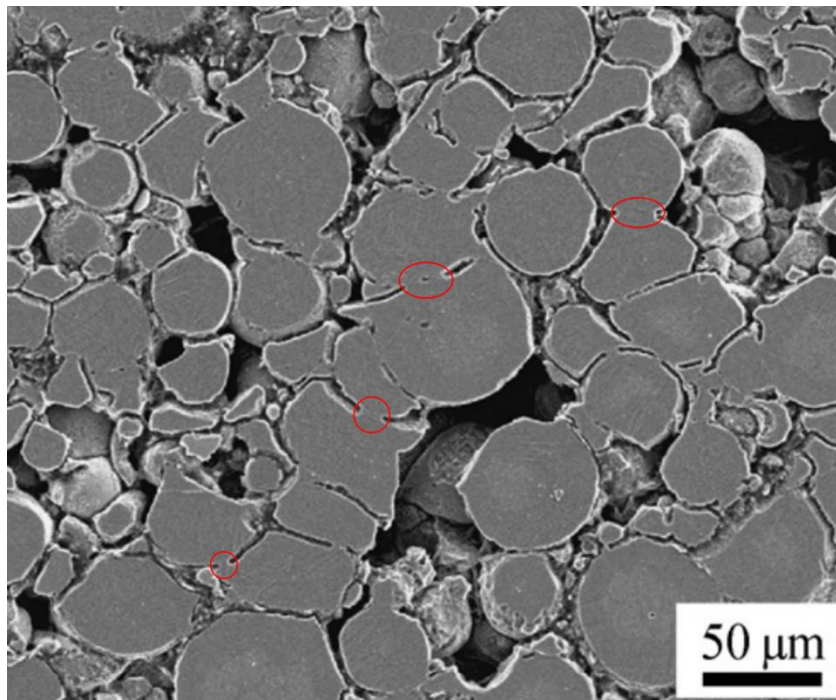


Figure 2.20: Image of an etched Ti-6Al-4V coating, where the circles show local bonding between deposited particles [72] (© 2007 Elsevier).

2.3.2.2 Critical Velocity

The critical velocity is the minimum velocity at which cold spray particles need to travel in order to bond to a substrate, as opposed to simply rebounding or causing erosion. The critical velocity is a value that is dependent on material type, particle size, particle and substrate temperatures, melting temperature, and surface finish [65], [73], [74]. As such,

it is difficult to determine the exact critical velocity for any one substrate-particle combination. On the other hand, at extremely high velocities impacting particles cause erosion and do not adhere; this is known as the erosion velocity [73], [75]. Thus, it is important that the particle velocity be between the limits of the critical and erosion velocities for deposition. Figure 2.21 shows this deposition window for 25 μm particles of various materials when impacting a substrate with identical material properties to the particle.

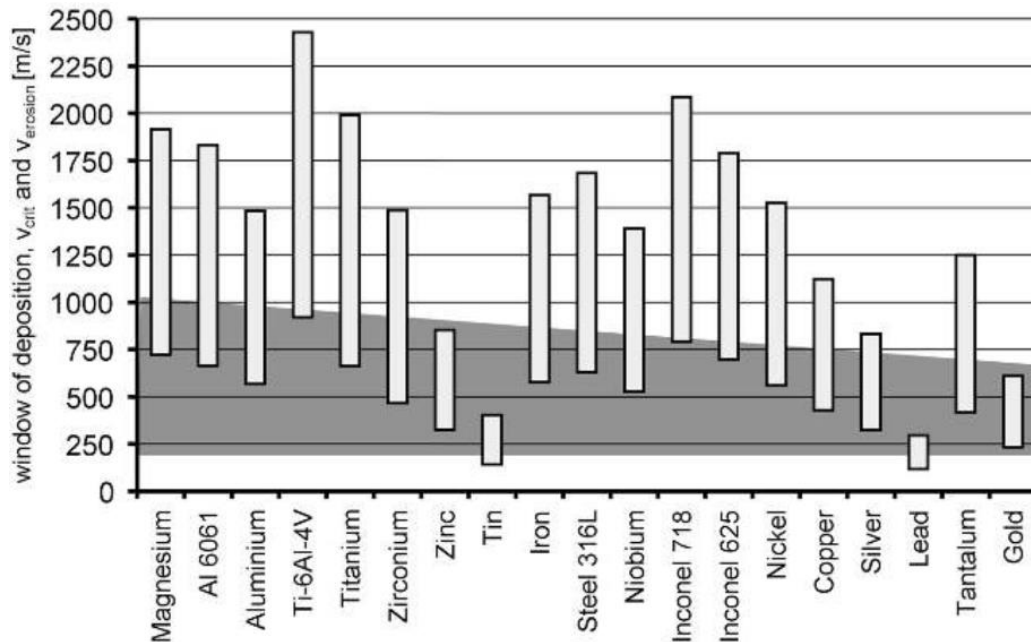


Figure 2.21: Deposition window of 25 μm particles of various materials impacting on substrates with identical material properties, with the grey area indicating attainable process conditions at the time of the study [75] (© 2010 John Wiley & Sons).

Also important to note, is that the critical velocity between particle and substrate is not the same as that between an incoming particle and those that are previously deposited

[72]. Thus, the critical velocity is altered following the deposition of the initial particle layer.

2.3.2.3 *Deposition Efficiency*

Deposition efficiency (DE) is a measure of the percentage of impacting particles that actually stick to the substrate or other previously deposited particles. The DE of any particle-substrate combination can be enhanced in many ways. Substrate preheating is one such technique, where the substrate materials are thermally softened, improving plastic deformation [76]. Various surface preparation techniques can also be employed to aid the deposition of initial coating layers, such as grit blasting or polishing [69]. Powder preheating can be used to improve the DE through every layer of the coating, by thermally softening the particles and therefore aiding them to plastically deform [77], [78]. Nonetheless, the most important contributing factor to DE is particle velocity. All preheating or surface treatments only have the effect of lowering the required critical velocity for bonding. There is a clear trend demonstrating that the higher the particle velocity, the higher the DE, as can be seen in Figure 2.22. This trend continues until erosion occurs [74], [79]–[81].

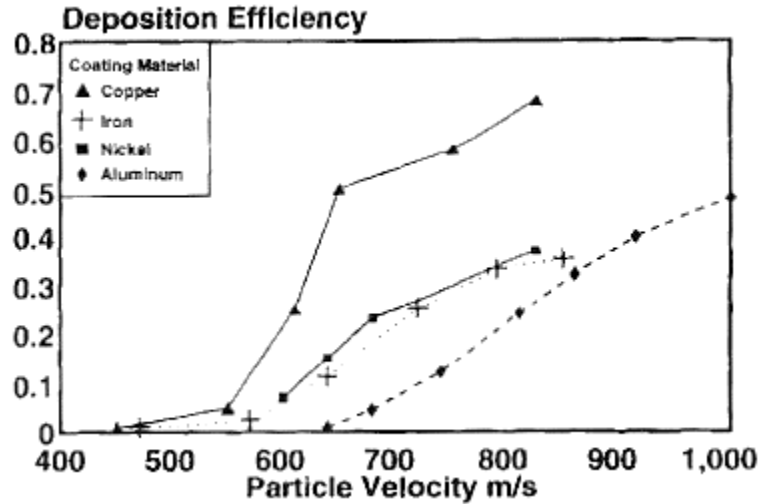


Figure 2.22: Increasing DE with increasing particle velocity [62] (© 1998 Springer Nature).

2.3.3 Effect of CGDS Parameters and Relevant Gas Dynamics

Principals

Many fundamental gas dynamic concepts are crucial to the performance of the CGDS process. Through the application of these principles, cold spray feedstock particles can be accelerated to very high speeds before impact with the substrate. In order to reach such high velocities, the carrier gas is accelerated through a converging-diverging nozzle (De Laval), which in turn is used to accelerate the powder particles by means of momentum transfer. The nozzle design is thus critical to ensure that the required supersonic carrier gas velocities are achieved.

The first important parameter which affects gas velocity in CGDS is the gas nature [74].

This can be explained by examining the definition of the Mach number [62]:

$$V_{gas} = M\sqrt{kRT} \quad (2-17)$$

where V_{gas} is the gas velocity, M is the Mach number, k is the gas specific heat ratio, R is the gas constant, and T_{gas} is the gas temperature. For a given design Mach number, it is evident that by changing the carrier gas nature, the velocity of the carrier gas can be drastically altered. In the case of CGDS, the most common carrier gases used are air ($k=1.4$, $R=287$ J/kg·K), nitrogen ($k=1.4$, $R=297$ J/kg·K), or helium ($k=1.67$, $R=2080$ J/kg·K). Given that helium exhibits a far superior individual gas constant than nitrogen or air, its use allows for greater gas velocities to be achieved for a given gas temperature, resulting in a higher DE [74], [82]. However, while the use of helium in CGDS can have great benefits in terms of DE, its cost is significantly higher than that of air or nitrogen, which does not make it an economically viable solution for most applications. Helium recycling apparatus have been used in order to reduce the cost of helium use, though this still constitutes a sizable capital investment in equipment. Therefore, it is typically reserved for materials that are very difficult to deposit (i.e. that have very high critical velocities). Nitrogen is thus used in most applications as it can achieve slightly higher velocities than air, and ensure no oxidation of the sprayed materials can take place.

The next parameter which affects gas velocity is carrier gas temperature. The gas temperature can be derived from the one-dimensional isentropic relation [62]

$$\frac{T_o}{T} = 1 + \frac{k-1}{2} M^2 \quad (2-18)$$

where T is the local gas temperature and T_o is the gas stagnation temperature. From this relation, raising the gas stagnation temperature also increases the local gas temperature, which in turn will increase the gas velocity, as per equation (2-17). This will increase the particle velocity [74], [83], and the particle temperature [83], as is shown in Figure

2.23. A higher gas temperature also increases the substrate temperature as the high temperature gas flow heats the underlying material [76], [82]. All of these effects are beneficial to DE.

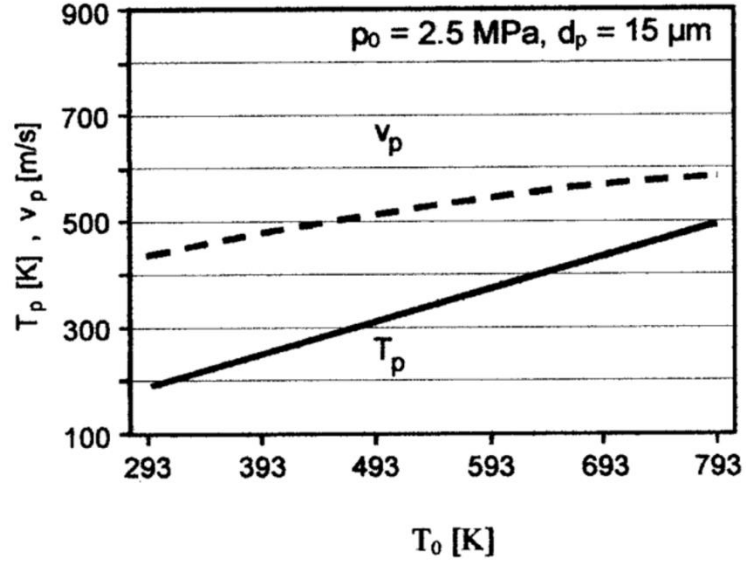


Figure 2.23: Particle velocity (v_p) and temperature (T_p) as a function of gas stagnation temperature [83] (© 2001 Springer Nature).

Next, momentum transfer from the carrier gas to the particle must be examined. The acceleration of the particles is a result of the force imparted to them by the carrier gas:

$$F = m_p a_p \quad (2-19)$$

where F is the net force acting on the particle, m_p is the mass of the particle, and a_p is the particle acceleration. The force on the particle is the drag force resulting from the high velocity carrier gas flow. To note is that the gravitational forces are neglected as they are orders of magnitude smaller than drag force. The drag force, F_D , is given as [84]

$$F_D = \frac{1}{2} C_D \rho V^2 A_p \quad (2-20)$$

where C_D is the drag coefficient, ρ is the gas density, V is the gas velocity relative to that of the particle, and A_p is the cross sectional area of the particle perpendicular to the flow direction. This can be rewritten to include the relative particle velocity [62], [85]:

$$F_D = \frac{1}{2} C_D \rho (V_{gas} - V_p)^2 A_p \quad (2-21)$$

where the V_p is the particle velocity. Thus rewriting equation (2-19) it is possible to see the effect of each parameter:

$$m_p a_p = \frac{1}{2} C_D \rho (V_{gas} - V_p)^2 A_p \quad (2-22)$$

In terms of the shape and size of the particle itself, this shows that the higher the particle mass, the lower the acceleration becomes, and therefore the lower the particle velocity at impact, typically resulting in lower DE [86]. However, it has been noted that larger particles have a lower critical velocity which has been attributed to temperature effects as shown by Schmidt et al. [73]. Another factor is the drag coefficient and cross sectional area of the particle, showing that the higher these values are, the higher the particle acceleration, leading to higher impact velocity. However, the cross sectional area and the mass of the particle are not independent, as can be seen by rewriting equation (2-22) in terms of particle diameter for a spherical particle, d_p ,

$$\frac{4}{3} \rho_p d_p a_p = \frac{1}{2} C_D \rho (V_{gas} - V_p)^2 \quad (2-23)$$

where ρ_p is the density of the particle. Here it is seen that the particle acceleration decreases with particle diameter, leading to the conclusion that larger particles result in lower impact velocities. The drag coefficient depends on the shape of the particle, with

spherical particles exhibiting lower drag coefficients than irregularly shaped particles [86].

Another factor that can have an effect on particle acceleration is gas density. From the one-dimensional isentropic relation, the gas density is written as [62]

$$\frac{\rho_o}{\rho} = \left(1 + \frac{k-1}{2} M^2\right)^{\frac{1}{k-1}} \quad (2-24)$$

where ρ_o is the stagnation density, which can be found from the perfect gas law [84]

$$P_o = \rho_o R T_o \quad (2-25)$$

where P_o is the stagnation pressure. As a result of this, by raising the stagnation pressure, the stagnation density is also raised, consequently raising the particle acceleration through additional drag force, again raising the impact velocity [83], as can be seen in Figure 2.24. This also shows that stagnation pressure has no effect on particle temperature. On the other hand, raising stagnation temperature for a fixed pressure decreases stagnation density, but raises gas velocity, both of which influence drag force. However, since particle acceleration varies linearly with gas density, but varies with the square of gas velocity, thus the gas velocity increase has a more important effect on particle impact velocity than gas density.

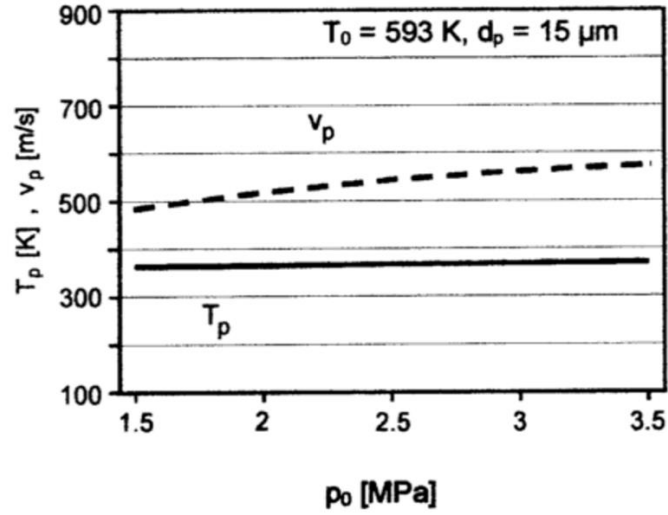


Figure 2.24: Particle velocity (v_p) and temperature (T_p) as a function of gas stagnation pressure [83] (© 2001 Springer Nature).

The final parameter which has an effect on the particle velocity is the Mach number of the flow, which is dictated by nozzle geometry. The area ratio between the nozzle throat, A^* , and any axial location along the length of the nozzle, A , determines the Mach number at said location, as determined by the one-dimensional isentropic relation [62]:

$$\frac{A}{A^*} = \frac{1}{M} \left[\left(\frac{2}{k+1} \right) \left(1 + \frac{k-1}{2} M^2 \right) \right]^{\frac{k+1}{2(k-1)}} \quad (2-26)$$

This shows that the larger the area ratio along the length of the nozzle, the larger the local Mach number will become, increasing gas velocity and particle velocity, as shown in Figure 2.25.

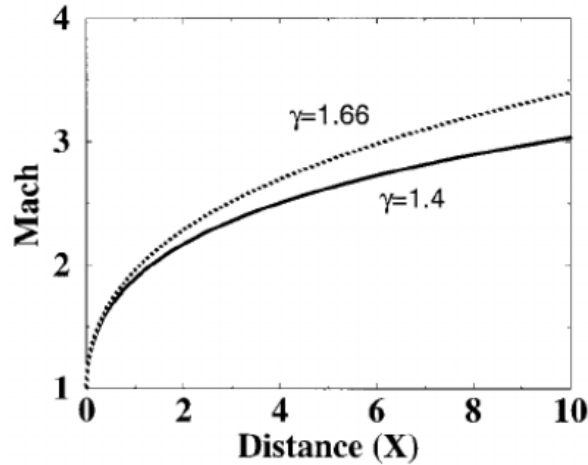


Figure 2.25: Increasing Mach number with increasing distance along the nozzle length as the area ratio increases [62] (© 1998 Springer Nature).

The length of the nozzle also determines the amount of time the particles remain in the accelerated gas stream: longer nozzles provide for a longer opportunity for the particles to be accelerated. However, the nozzle length and area ratio cannot be infinitely increased for use in cold spray due to the onset of complex shock wave-particle interactions at Mach numbers higher than 2 [87] and practical manufacturing limitations. Thus, nozzle designs for commercial cold spray systems are generally limited to the models supplied by the manufacturer.

Other parameters that can influence deposition are the nozzle traverse velocity and the nozzle standoff distance. Traverse velocity itself has not been shown to have an effect on DE, but using low traverse velocities with high carrier gas temperatures has been shown to heat the substrate, changing deposition conditions [76], [82], [88]. Nozzle standoff distance also has an effect on DE, where it has been shown that DE is reduced if the standoff distance is too short or too long, due to bow shock effects and particle deceleration, respectively [89].

2.3.4 Cold Spray Process Costs

In the work of Stier, the costs of the CGDS process and its consumables are highlighted [90]. The primary cost is that of deploying the cold spray equipment itself, which incorporates the cost of the equipment, installation, maintenance, repair, and depreciation. Next is the cost of labour, which is dependent on the level of automation of the system. As for consumables, the most important costs are the cost of the feedstock powder, the cost of electricity (predominantly for heating of the carrier gas), and the cost of the carrier gas consumption. These costs are heavily dependent on process parameters, as these affect the rate of consumption, as well the spray time necessary to produce the required final product.

2.3.5 Annealing of Cold Spray Coatings

Annealing has been investigated on cold sprayed deposits as a means to eliminate certain undesired artifacts of the spray process. The predominant defect that annealing seeks to remedy is work hardening from the extensive plastic deformation in the spray process, which results in very low ductility in the as-sprayed coating [56], [91]. Annealing has also been shown to relieve residual stresses, and improve particle cohesion [91], [92]. In a study done by Wen-Ya et al. [93] it was found that annealing of a copper coating improved the interfacial bonding between particles, as well as returned the electrical resistivity and micro hardness to values comparable to those of bulk copper.

2.3.6 Powder Recycling

In the CGDS process, if the spraying of a coating has a DE of less than 100%, there are by definition particles that do not adhere to the substrate or previously deposited

particles. In certain cases, this can account for a substantial loss of feedstock material. In order to mitigate this, a possible solution would be to reclaim this un-deposited powder, and reuse it in future sprays. However, there is no published work on the subject of spraying reclaimed powder. Furthermore, there is only sparse work done on the analysis of un-deposited particles provided by Fernandez et al. [94]. In this work, the feedstock powder used was chromium carbide – nickel carbide cermet. The un-deposited particles were reclaimed using an adhesive carbon tape placed near to the zone being sprayed. The results showed that un-deposited particles either fractured or severely deformed upon impact with the substrate.

3 RESEARCH OBJECTIVES

3.1 General Objectives

The main objectives of this work are to determine cost effective parameters for the production of near net shape pin fin arrays, and to improve on these cost effective parameters by implementing powder recycling. CGDS has been proven to be a viable technique for the production of this fin configuration [14]–[20], but as a considerable portion of the feedstock powder is lost in the process, it is important to explore the possibility of reclaiming and reusing it. Powder recycling in CGDS has not been explored in the literature, and therefore the effects experienced by the powder and their influence on deposition, must be explored. Consequently, the goals of this work were to: a) determine the most cost effective CGDS parameters, b) Spray pin fin array heat sinks and reclaim the un-deposited powder using an in-house reclamation apparatus, c) analyze the reclaimed powder and ascertain its sprayability, d) determine the cost savings of using powder recycling, e) verify the thermal performance of the resulting heat sinks, f) improve the coatings and fins with annealing, and g) use a CFD model to verify assumptions made in thermal performance testing.

3.2 Determination of Cost Effective Parameters and Fin Analysis

The first specific objective was to create a cost analysis model to determine the least costly parameters for pin fin production. In this work the requisite fin height was set to 1.7 mm to ensure a tight fit in the heat testing channel, and as such it was important to determine a specific gun traverse velocity for every set of CGDS parameters to be tested

that would produce fins which could then be milled down to 1.7 mm. In this way, the cost of each set of parameters could be compared on an even basis. By using a few different traverse velocities for all sets of parameters, it was subsequently possible to determine the appropriate traverse velocity by linear interpolation. Following this, fins were sprayed up to the requisite height using every set of parameters and analyzed. Analysis of the cross sections by scanning electron microscope (SEM) and optical microscope were used to characterize density, as well as to quantify the shape of the resulting fins. Digital microscopy was used to render 3D representations of the fins.

With the deposition results of these same-height sprays, it was possible to create a cost model to compare the sets of fins from an economic standpoint. This took into account the powder costs, gas costs, electricity costs, masking costs and the prospective cost of labour. The lowest cost spray parameters determined were used for the majority of the remainder of the work.

3.3 Production of Pin Fin Array Heat Sinks

Once the most cost effective parameters were selected, the next specific objective was to produce full sized pin fin arrays. Given the dimensions specified by the heat test rig design, a pin fin array spray rig was designed and manufactured. The spray rig needed to be able to mask the required area of the substrate to be sprayed, as well as keep everything centered and square with respect to the borders of the baseplate. Another important consideration was to maintain the desired standoff distance between the substrate and the wire mesh-mask.

Once the spray rig was completed, final heat sinks were sprayed using the previously determined lowest cost parameters. In order to have lower flow bypass in the heat test rig, and to eliminate local variations in fin height, the fins were sprayed taller and subsequently milled down to the requisite 1.7 mm height of the heat test rig test channel. The un-deposited powder from these full sized sprays was reclaimed using an in-house reclamation apparatus for subsequent sprays and further analysis.

3.4 Powder Reclamation and Characterization

Once full sized heat sinks were produced, the next specific objective was to reclaim the un-deposited powder and compare it to the as-received powder. For reclamation, a in-house powder reclamation apparatus was used within the spray chamber for in-situ powder collection. Once the powder was reclaimed, it was sieved for impurities (delaminated mask coatings, tape, etc.) and analyzed using SEM. This analysis consisted mostly of a morphological characterization of the reclaimed powder compared to its original spherical shape. Particle velocity measurements were also taken using the reclaimed powder and as-received powder to compare whether the powder morphology had an effect on in flight dynamics.

3.5 Sprayability of Reclaimed Powder

The next specific objective was to use the reclaimed powder to produce coatings in order to establish its sprayability and its effect on coating quality. Three powder blends were compared: 100% as-received powder, 50wt% as-received with 50wt% reclaimed powder, and 100% reclaimed powder. Full coatings were produced with each powder blend and DE was determined, as well as a comparison of powder flowability. From these coatings,

porosity and hardness were also measured. The coatings cross sections were then etched to show the particle boundaries.

3.6 Cost Reduction from Powder Recycling

The cost analysis model was then updated to include the savings associated with powder recycling. This was done by removing the cost of the un-deposited powder leaving only the cost of the powder which stuck to the mask assembly and that which resulted in pin fins.

3.7 Thermal Performance Characterization

The pin fin arrays sprayed with as-received powder were tested in the heat test rig to determine their thermal performance characteristics. Also tested were bare heat sink baseplates as a grounds for comparison. Water flow rate was varied while the temperature of the heating element and the water inlet and outlet temperature were recorded.

3.8 Coating and Fin Annealing

The next step was to attempt to improve the conductivity of the fins and coatings. Thus, the coatings produced with each powder blend and the full sized pin fin arrays underwent annealing. After annealing, the coatings were again tested for porosity and hardness, as well as re-etched to observe any changes to the particle boundaries. The heat sinks were retested in the heat test rig to determine the effects of annealing on the fin performance.

3.9 Heat Transfer Modeling

Having determined the thermal performance of the pin fin arrays, a CFD model was developed to better understand the flow of heat and water throughout the heat transfer test

rig. The goal of the model was to verify the validity of certain assumptions made in the characterization of the pin fin performance, as well as to visualize the temperature distribution in the heater block, pin fin array, and the flow of water.

4 EXPERIMENTAL PROCEDURES

4.1 Feedstock Material

The feedstock powder used for this work was spherical Cu-159 copper powder from Praxair Surface Technologies (Indianapolis, IN, USA). Laser diffraction analysis (Microtrac S3500, Nikkiso, Japan) was done to determine the mean particle sizes, yielding an average particle diameter of 11.43 μm based on volume (MV) and 8.05 μm based on the number of particles (MN). The small variance between MN and MV shows that the particles have a fairly tight size distribution.

This copper powder was chosen since the size is in the optimal range for CGDS [73]. Spherical particles were chosen for ease of comparison between as-received powder, and powder reclaimed post-impact. In the frame of powder recycling, this allows for simple direct morphological comparison between the perfectly spherical initial case, and the severely damaged reclaimed particles. An SEM image of the as-received powder is shown in Figure 4.1.

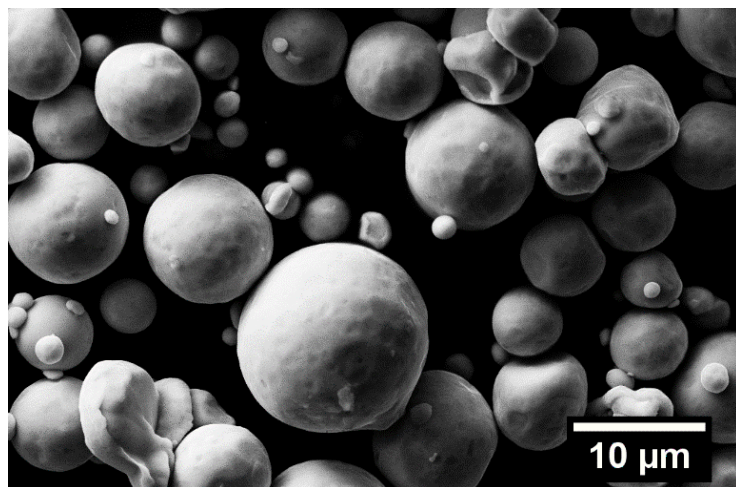


Figure 4.1: SEM image of Praxair Cu-159 copper powder.

4.2 Substrates

4.2.1 Copper Plate Substrates

The substrates used were 1/8" (3.175 mm) thick copper plates (Metal Supermarkets Ottawa). These substrates served as the baseplates for the pin fin array heat sinks produced by CGDS. For the production of the final pin fin array heat sinks, the substrates were precision milled to a size of 61.5 mm by 149 mm. These dimensions were determined with respect to the heat transfer test rig. A picture of the final baseplate is shown in Figure 4.2.



Figure 4.2: Final pin fin array copper baseplate.

4.2.2 Substrate Surface Preparation

The substrate surface was prepared by grit blasting to roughen the surface, as it was found that this enhances the bond strength of copper coatings on copper substrates [69].

The grit blasting unit propels grit by passing compressed air through a 13/64" (5.2 mm) inner diameter steel nozzle. The gun is hand held and the grit is gravity fed from a hopper mounted on top. All blasting was done at approximately a 45° angle with a standoff distance of approximately 2 inches, and a steel holder was used to back the plates during blasting in order to prevent warping from the gas stream.

The average air consumption of the unit is 6 CFM (170 L/min) at a maximum pressure of 90 psi (621 kPa) (Princess Auto SKU: 8140709). The grit used was Ebonygrit - copper slag blasting abrasive (Princess Auto SKU: 8200594). This grit is composed of #EG 20 (1.5-3.0 mm) ferrosilicate. After grit blasting the substrates were cleaned using acetone in an ultrasonic bath and dried.

4.3 CGDS Equipment

4.3.1 Heater and Controls

The CGDS system used for the production of all coatings and pin fin arrays in this work was the SST Series EP cold spray system (Centerline (Windsor) Ltd., Windsor, ON, Canada). This system uses a 15 kW heater and has a maximum stagnation temperature and pressure of 650°C and 3.4 MPa, respectively. The temperature can be set using a touch screen on the cabinet, while the pressure is controlled by a valve on its side. Pressure and temperature readings of the system are calibrated regularly using a calibration kit. A picture of the heater and control cabinet is seen in Figure 4.3.



Figure 4.3: CGDS gas heater (left) and control cabinet (right).

4.3.2 Nozzle Assembly

The CGDS nozzle assembly is composed of the nozzle holder, which connects the powder feed line, the converging-diverging portion of the nozzle, and an extension of the nozzle which is typically referred to as the nozzle. The model used in this work is the UltiLife nozzle assembly, which is commercially available through Centerline Ltd. (Windsor, ON, Canada). A picture of the assembly is seen in Figure 4.4.

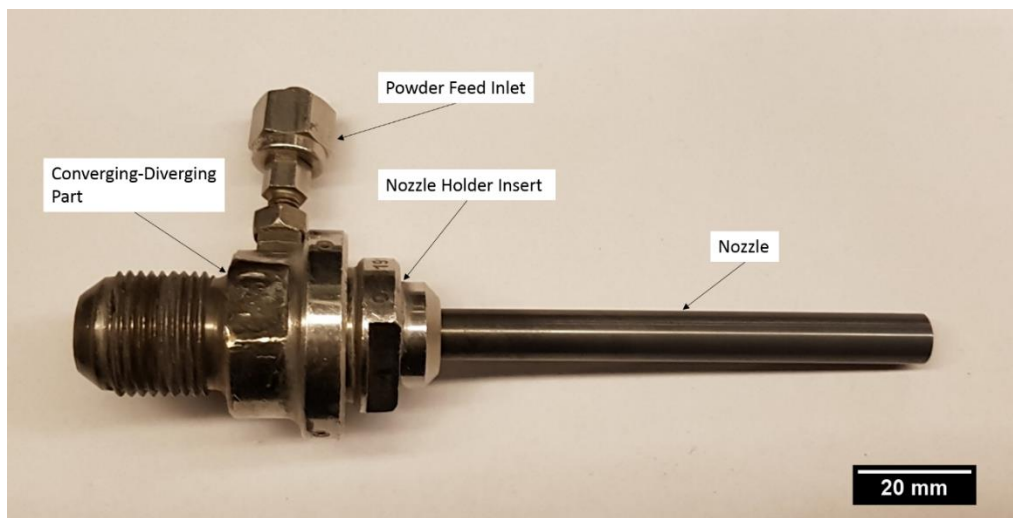


Figure 4.4: CGDS nozzle assembly.

The nozzle holder connects the gas heater to the nozzle and incorporates the converging-diverging part of the nozzle necessary to accelerate the main propellant gas to supersonic speeds. The holder used in this work has a 2 mm throat diameter. It also has a channel into which the powder inlet is inserted. The converging-diverging part of the holder is shown in Figure 4.5.

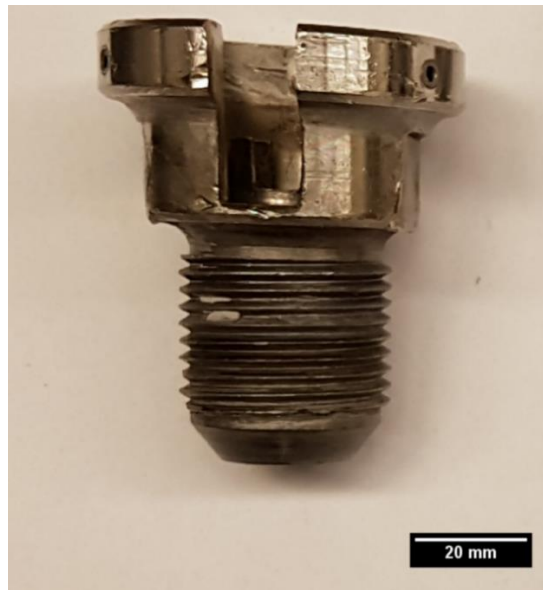


Figure 4.5: CGDS converging-diverging part.

The nozzle powder inlet has the dual role of maintaining the nozzle in place, centered in contact with the throat exit of the converging-diverging piece, and of feeding powder through lateral injection into the nozzle. A spring is compressed between the powder inlet and an insert which fits into the converging-diverging part. These components are shown in Figure 4.6.

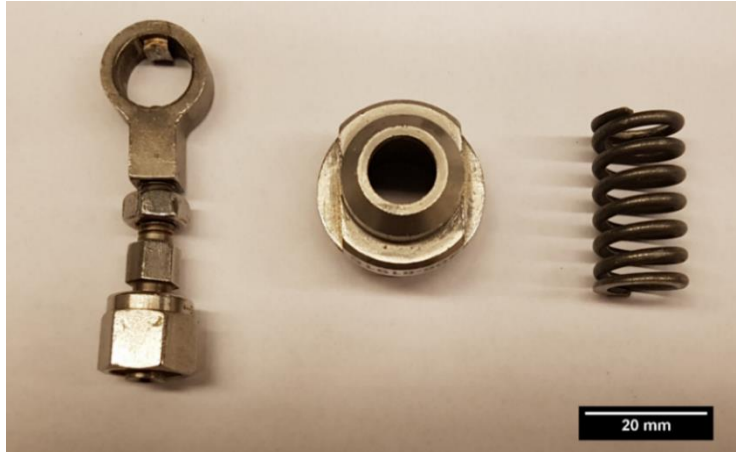


Figure 4.6: From left to right: The nozzle powder inlet, the insert into the converging-diverging part, and the spring.

The CGDS nozzle used for this work is the Centerline Ltd. UltiLife nozzle, and is shown in Figure 4.7. It consists of 120 mm long diverging section, with an exit diameter of 6.3 mm. The nozzle also has a built in feedstock powder inlet, positioned for lateral injection near the nozzle carrier gas inlet and connecting directly to the powder inlet piece. This nozzle is made of a clog-resistant material, tungsten-carbide in a cobalt matrix, and was selected because the copper feedstock powder caused clogging in steel nozzles during initial testing.



Figure 4.7: CGDS UltiLife Nozzle.

4.3.3 CGDS Robotic Traverse System

The CGDS gun is mounted on an automated X-Y robotic traverse system, seen in Figure 4.8. It uses electrical stepped motors to rotate linear screws, resulting in the planar motion of the gun. The velocity and displacement of the gun are controlled digitally with software included with the system. The maximum traverse speed of the system is rated at 200 mm/s.



Figure 4.8: CGDS traverse system.

4.3.4 Powder Feeding Equipment

The powder was supplied to the CGDS system using a commercially available powder feeder (Model: AT-1200HP, Thermach Inc., Appleton, Wisconsin, United States of America). The feedstock powder is placed into a pressurized canister within which a rotating perforated wheel controls the powder feed rate. A hammer, mounted on top of this wheel, hits the surface of the wheel periodically to aid in the continuous flow of the powder and to ensure that the holes in the wheel are filled. Both the hammer and wheel

assembly, and the pressurized canister are shown in Figure 4.9. As the wheel rotates, the holes line up with the carrier gas stream, and the powder is carried to the CGDS system. The canister is pressurized with nitrogen to a pressure of approximately 30 psi during operation to prevent back flow of the carrier gas into the canister.

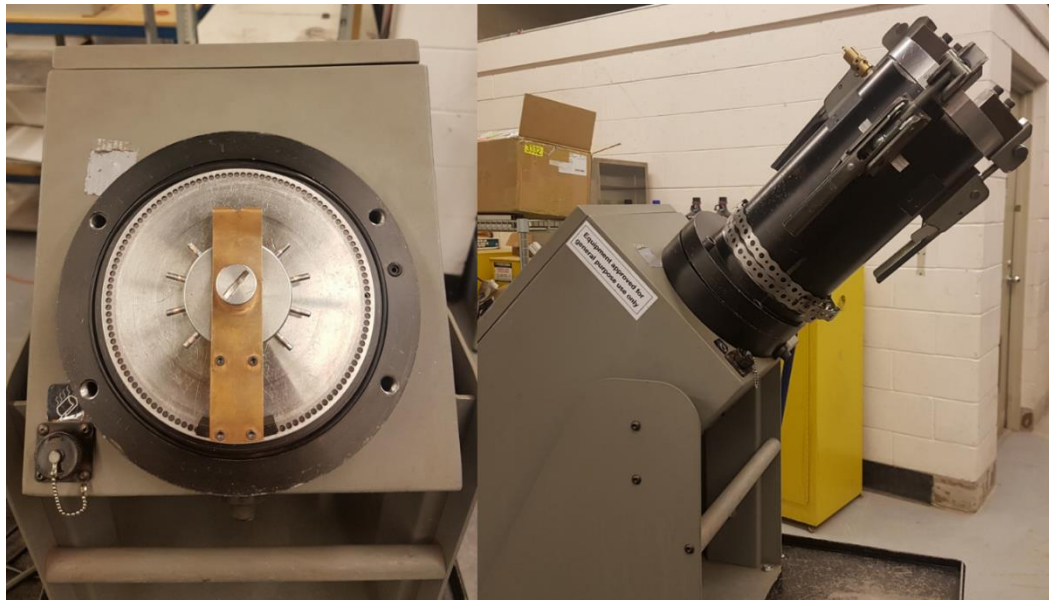


Figure 4.9: Powder feeder assembly showing the hammer and perforated wheel (left), and the pressurized canister (right).

In order to control the powder feed rate, the rotational velocity of the wheel and the size of the holes in the wheel can be adjusted. The faster the wheel and larger the holes, the higher the powder feed rate. In this work, two different wheel hole sizes were used; the 240 medium hole wheel, and the 120 large hole wheel. The medium hole wheel was used initially with the as-received powder, while the large hole wheel was adopted later for the reclaimed powder. This adjustment was necessary since the reclaimed powder exhibited a reduced flowability when compared to the as-received powder. Both feeder wheels are shown in Figure 4.10.



Figure 4.10: 240 medium hole feeder wheel (left), and 120 large hole feeder wheel (right).

4.3.5 Spray Chamber

The chamber within which the CGDS was performed is shown in Figure 4.11. It is equipped with a dust collection system which uses a vacuum to pull the non-deposited powder through a water filter so that it is safely trapped. The remaining air is passed through a HEPA filter before being recirculated to the room.



Figure 4.11: CGDS spray chamber.

4.3.6 Gas Delivery System

The feeder gas and main propellant gas was provided by preassembled 11 bottle nitrogen bottle packs purchased from Linde (Mississauga, Ontario, Canada). The bottle packs were delivered with a pressure of approximately 18 MPa.

4.4 Pin Fin Sample Sprays

In the initial testing and parameter selection for the production of finned samples, small versions of the final substrate and mask assembly were created. The smaller substrates were approximately 50 mm by 20 mm, made from 1/8" (3.175 mm) copper plate. In these smaller test samples, the grit blasted copper substrate had holes drilled directly through it, wherein bolts could be fastened to hold the wire mesh in place. Washers were used to ensure that the desired 2 mm substrate to mesh standoff distance was maintained. Tightly

fastening the bolts was enough to ensure that the wire mesh would not bow under the force of the CGDS system's gas stream. A picture of this experimental design is shown in Figure 4.12.

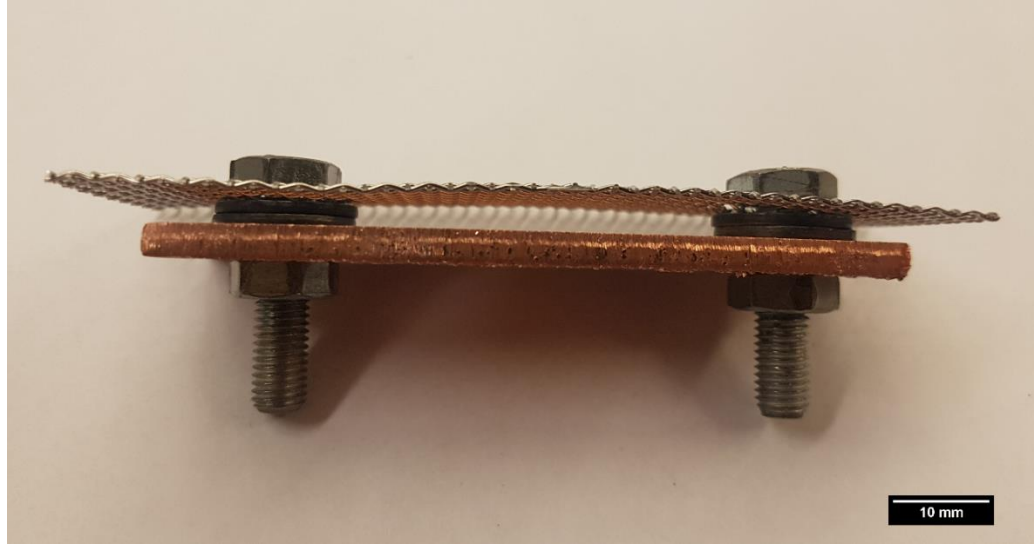


Figure 4.12: Small sample pin fin spray setup.

4.4.1 Traverse Velocity Determination

Pin fins were produced using various CGDS parameters. With these different parameters, it was important to determine traverse velocities that would result in the same fin heights for each set of spray parameters in order to ensure accuracy of comparison in terms of cost analysis. In order to determine the correct traverse velocity to achieve the same height of fins, traverse velocity was correlated to be inversely proportional to the fin height, as seen in the following equation:

$$H * V_T = c_{V-H} \quad (4-1)$$

where H is the height of the fin, V_T is the gun traverse velocity, and c_{V-H} is the constant correlating traverse velocity to fin height for a specific combination of process parameters. This correlation is later verified in the results Section 5.1.2.

4.4.2 Mesh Reuse

Tests were done to determine whether the wire-mesh mask could be used for multiple sprays. To determine this, a single piece of wire mesh was used for multiple sample pin fin sprays, while using a new substrate each time. The DE on the mesh and the substrate was calculated for each test.

4.5 Pin Fin Array Spray Rig

A spray rig was devised in order to hold in place the heat sink baseplate substrates. It was composed of a steel holder block and a wire mesh sandwiched between two sheet metal masks. The rig also had to maintain the required standoff distance of 2 mm between the wire mesh and the substrates. This was accomplished by manufacturing spacers to the required thickness, while taking into account the thickness of the baseplate. The full assembly is shown in Figure 4.13.

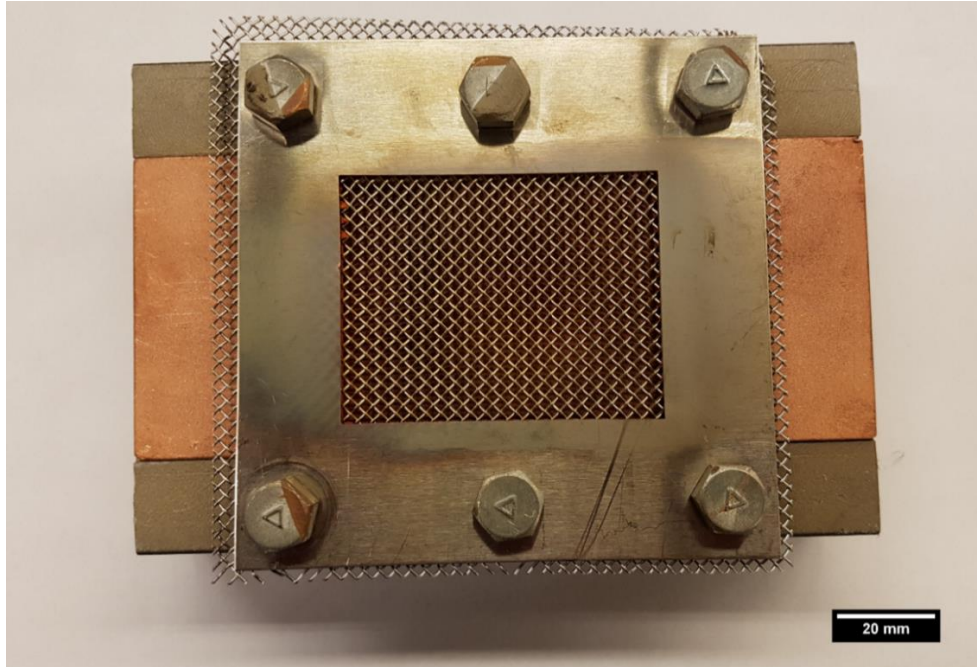


Figure 4.13: Pin fin array spray rig.

For clarity, an exploded view of the assembly is shown in Figure 4.14.

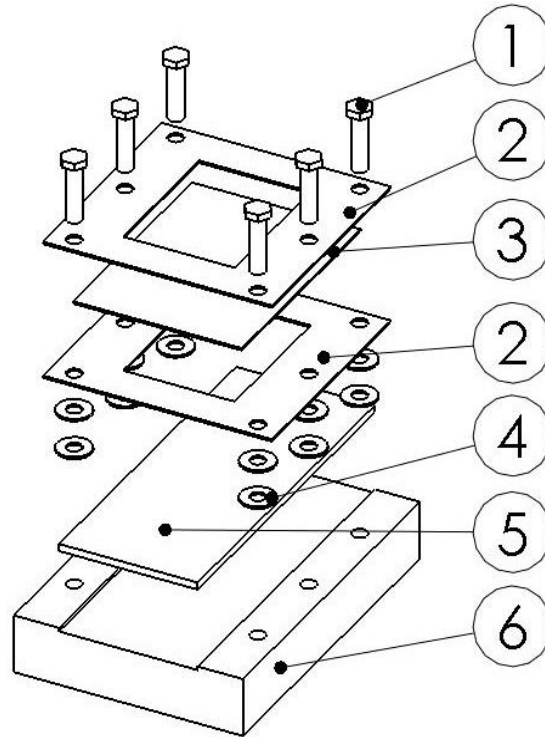


Figure 4.14: Spray rig: 1) Bolt; 2) Mask; 3) Wire mesh; 4) Spacer; 5) Substrate; 6) Holder block.

4.5.1 Steel Holder Block

To ensure the proper alignment of the baseplate with the sheet metal masks, a steel holder block was precision milled, and shown in Figure 4.15. A slot was machined into the block to ensure the centering of the baseplate in the widthwise direction, and the block itself matches the exact length of the baseplate for easy alignment in the lengthwise direction. The block has a series of 6 holes machined into it to fasten the mask and mesh into place.

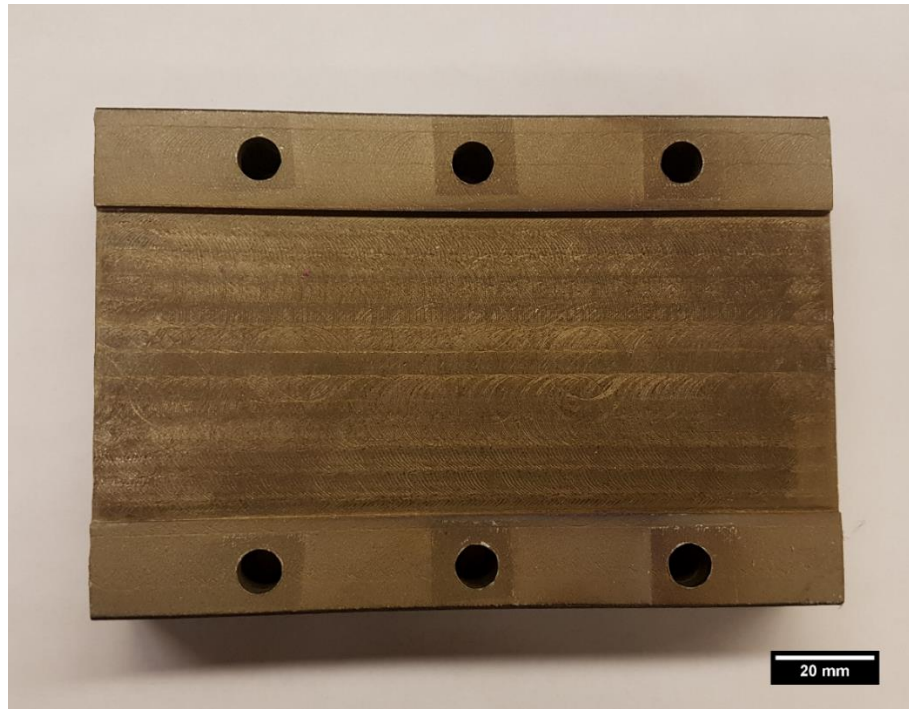


Figure 4.15: Steel holder block for the pin fin array spray rig.

4.5.2 Sheet Metal Masks

The sheet metal masks were machined at the University of Ottawa Mechanical Engineering Department machine shop by Electrical Discharge Machining (EDM). This process was used to ensure that the exact dimensions required for the footprint of the finned area could be obtained. This was critically important since this footprint is made to match the shape of the underlying hot element to ensure that the most heat possible is transferred through the finned area. The masks were inserted on both sides of the wire mesh so that they could be pressed together and aid in the rigidity of the mesh. Figure 4.16 shows a picture of the sheet metal mask.

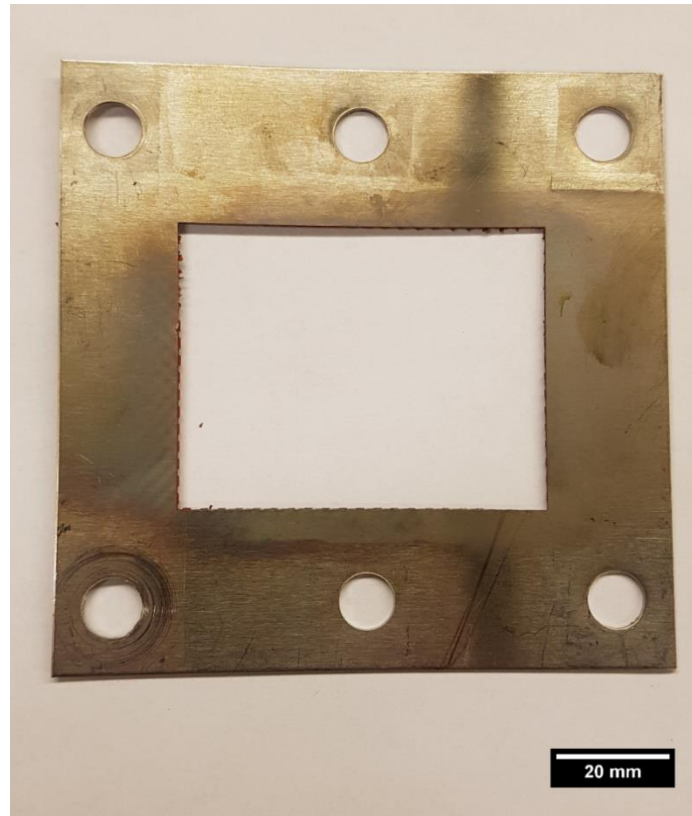


Figure 4.16: Sheet metal mask cut with wire EDM to match heat sink footprint geometry.

4.5.3 Stainless Steel Wire Mesh

The mesh used for pin fin production was a corrosion-resistant 304 stainless steel woven wire cloth mesh with 12 wires/inch in each direction and a 0.584 mm wire diameter. This mesh was chosen because it yielded good results in previous CGDS pin fin studies [14]–[20]. The mesh was cut using a pneumatic shear at a 45° angle from the direction of the weave in order to produce the desired staggered fin configuration. The holes through which the bolts were to hold the mesh in place were made using a manual punch press. The final mesh layout is shown in Figure 4.17.

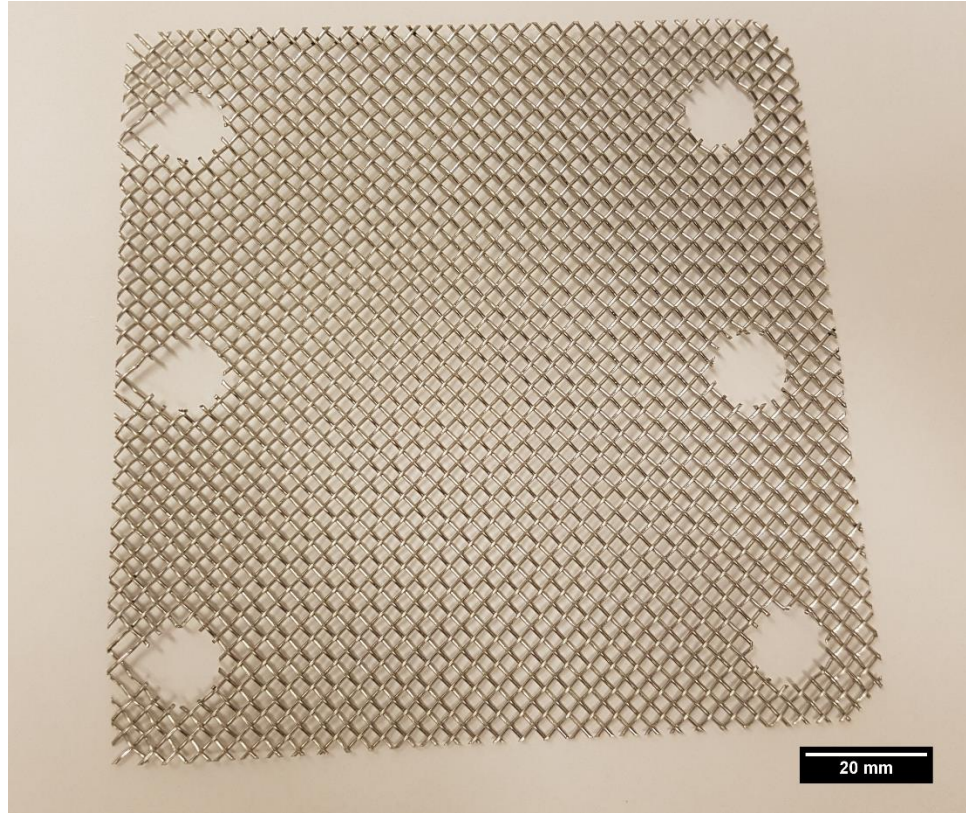


Figure 4.17: Wire mesh-mask used for the production of staggered pin fin arrays

4.6 Pin Fin Array Milling

In order to minimize flow bypass in the heat test rig, and to eliminate local variations in fin height, the fins were sprayed above their desired heights by a minimum of 100 μm and subsequently milled down to the requisite 1.7 mm height of the heat test rig test channel shown further in this section. The fins were machined down in 100 μm increments to avoid damaging or removing fins from the substrate.

4.7 Powder Reclamation Apparatus

In order to reclaim the un-deposited powder from the spraying, a reclamation system was built in house and it is shown in Figure 4.18. It consisted of a sheet metal box, which was

large enough not to affect the area being sprayed. An opening in the top panel of the box allowed the nozzle to enter and fulfill the desired spray pattern. A detachable side panel allowed for ease of access to the substrate. Finally, a ducting pipe equipped with a 0.5-micron filter was connected to the spray filtering vacuum system in order to avoid any backflow towards the gun, and to ensure that the maximum amount of un-deposited powder remained in the box. In this way, the hope was that the billowing effect of the high speed carrier gas would be mitigated, reducing the loss of powder through the opening in the top panel.

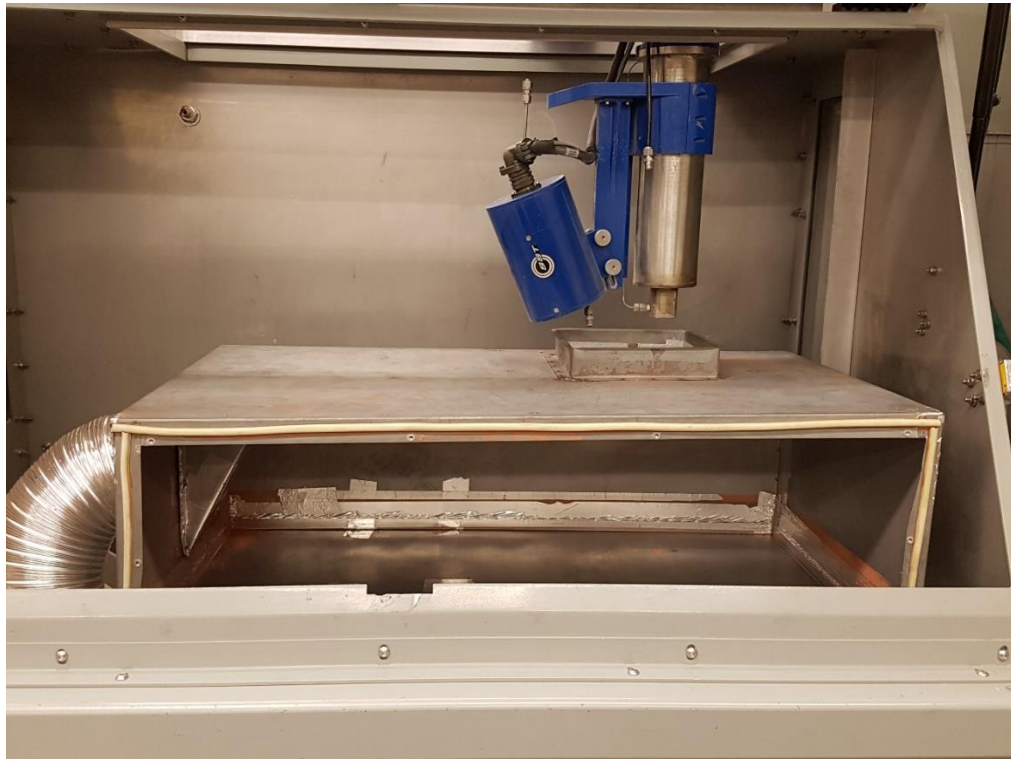


Figure 4.18: Powder reclamation apparatus inside the CGDS spray chamber.

4.8 Cost Analysis

Table 4.1 provides a list of the costs of all consumables associated with each expense, as they are available at the University of Ottawa. The hourly rate of labour was approximated based on typical local rates for manufacturing workshops.

Table 4.1: Cost of consumables used in the production of pin fin arrays.

Expense	Cost
Labour	70CAD/h
Mask	0.28CAD/unit
Electricity	0.11CAD/kW·h
Nitrogen Gas	3.02CAD/kg
Cu-159 Copper Powder	143CAD/kg

The costs for a single set of spray parameters for the production of fins derives most importantly from the time of spray. The required traverse velocity to reach the desired height, given a powder feed rate, was determined iteratively through experimentation and linear interpolation for every set of spray parameters. Thus, given a traverse velocity and spray area dimensions, the time of spray, t_{spray} , can be given by

$$t_{spray} = \frac{D_{nozzle}}{V_T} \quad (4-2)$$

where D_{nozzle} is the total linear spray distance traveled by the gun (m), and V_T is the gun's traverse velocity (m/s). Using the time of spray as the dependent variable for each spray, it is possible to determine the cost of every single expense depending on the spray parameters. In the case of the copper powder cost, the calculation is:

$$PC = t_{spray} * PFR * C_{Cu} \quad (4-3)$$

where PC is the total sprayed powder material cost, PFR is the powder feed rate (g/s), and C_{Cu} is the cost of copper per unit mass (cost/g). The powder feed rate was set to the

highest value possible without causing the equipment to clog. This cost can be further divided by including the concept of deposition efficiency. The powder cost can be broken up into three categories: (1) the powder that deposits onto the baseplate substrate to form the pin fins, (2) the powder that sticks and builds up onto the mask, and (3) the powder that does not deposit.

The cost of the powder deposited on the substrate, $PC_{substrate}$, is

$$PC_{substrate} = PC * DE_{substrate} \quad (4-4)$$

where $DE_{substrate}$ is the deposition efficiency on the substrate, determined experimentally.

The cost of the powder that is deposited onto the mask (and ultimately lost), PC_{mask} , is

$$PC_{mask} = PC * DE_{mask} \quad (4-5)$$

where DE_{mask} is the deposition efficiency on the mask, also determined experimentally.

The cost of the powder that does not deposit, $PC_{undeposited}$, is given by:

$$PC_{undeposited} = PC - PC_{substrate} - PC_{mask} \quad (4-6)$$

To calculate the gas cost, the gas flow rate used must be known. This can be measured or calculated from gas dynamics principles [84], taking the gas properties at the converging-diverging nozzle throat. The mass flow rate of gas, $\dot{m}_{N_2}$, is given by:

$$\dot{m}_{N_2} = \frac{P^*}{RT^*} \sqrt{kRT^*} A^* \quad (4-7)$$

where P^* and T^* are the pressure and temperature at the nozzle throat, A^* is the throat diameter, k is the specific heat ratio of nitrogen, and R is the individual gas constant of nitrogen. The pressure at the throat can be determined by

$$P^* = \frac{P_o}{\left(1 + \frac{k-1}{2}\right)^{\frac{k}{k-1}}} \quad (4-8)$$

where P_o is the stagnation pressure of the gas. The temperature at the throat can similarly be determined by

$$T^* = \frac{T_o}{1 + \frac{k-1}{2}} \quad (4-9)$$

where T_o is the stagnation temperature of the gas.

From this, the cost of gas, GC , is:

$$GC = t_{spray} * \dot{m}_{N2} * C_{N2} \quad (4-10)$$

where C_{N2} is the cost per unit mass of nitrogen gas.

To estimate the cost of electricity, the electrical consumption of the system was equated to the heat input necessary to heat the gas to the desired temperature. The electricity costs necessary to run the powder feeder and robotic traverse system were omitted as they are much smaller than that which is necessary to heat the carrier gas. This was done using a simplified version of the first law of thermodynamics [40],

$$\dot{Q}_{N2} = \dot{m}_{N2} * Cp_{N2} * (T_{N2,2} - T_{N2,1}) \quad (4-11)$$

where $\dot{Q}_{N2}$ is the heat input, Cp_{N2} is the specific heat of nitrogen, and $T_{N2,1}$ and $T_{N2,2}$ are the input and output gas temperatures, respectively. To get the total heat consumption, Q_{N2} , for the duration of a spray, the heat was converted to kilowatt-hours,

$$Q_{N2} = 2.78 * 10^{-4} * t * \dot{Q}_{N2} \quad (4-12)$$

to then find the total electricity cost, EC ,

$$EC = Q_{N2} * C_{electricity} \quad (4-13)$$

where $C_{electricity}$ is the cost of electricity per kilowatt-hour. The cost of the mask was a constant value for all spray parameters, as the characteristics of the masking did not vary.

Finally the labour cost, LC , is given by:

$$LC = t_{spray} * C_{labour} \quad (4-14)$$

where C_{labour} is the hourly cost of labour. However, the labour cost also depends on the level of automation of the system and set-up time between spray runs, and therefore must be evaluated on a case-by-case basis.

4.9 Heat Transfer Test Rig

The heat transfer test rig was designed to run a flow of water through the pin fin array heat sinks to be tested. A diagram of the full setup is shown in Figure 4.19.

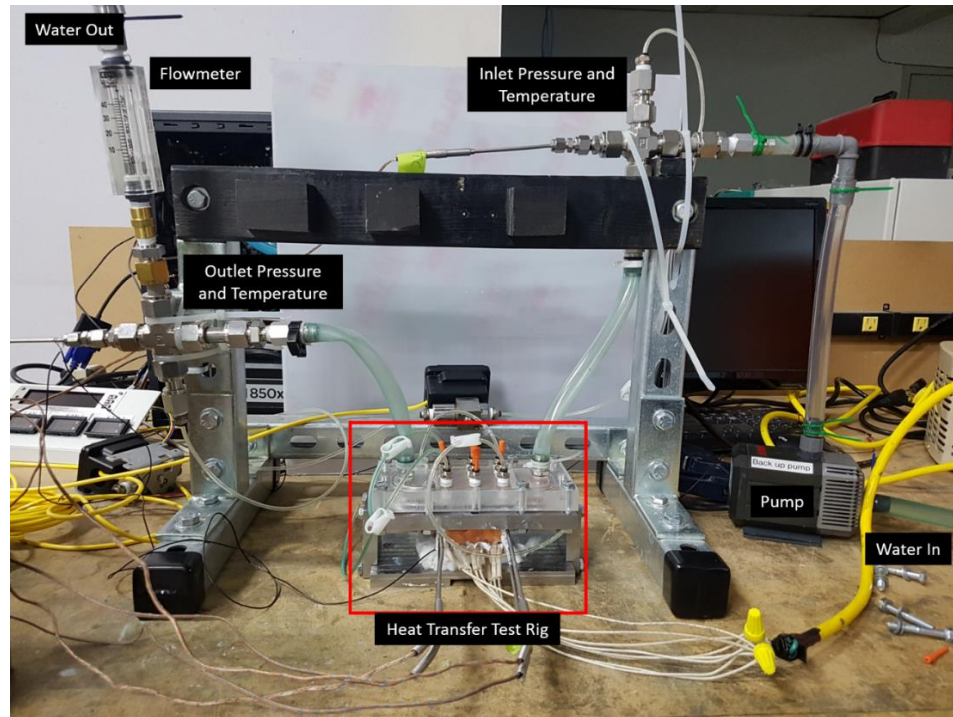


Figure 4.19: Representation of the heat transfer test rig flow.

From this experimental setup, several input parameters can be controlled, such as the water flow rate and the heat input. It is also equipped with multiple sensors to collect output data, such as the heater block and water temperatures, and the pressure drop across the finned area. An illustration of the inputs and outputs of the heat transfer test rig is shown in Figure 4.20.

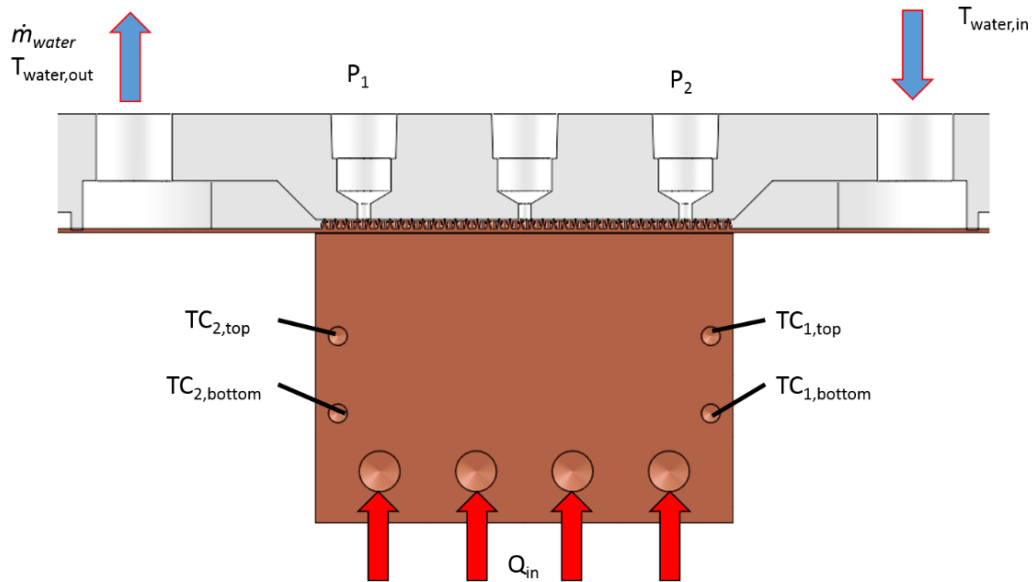


Figure 4.20: Illustration of the heat test rig input and output values.

The cover plate is made of transparent acrylic, for ease of identifying if there is air bubbles present in the system, and inset within this are the water inlet and outlet ports. The cover plate is sealed directly onto the heat sink baseplate using a Buna-N nitrile O-ring. The assembly of the heat transfer test rig is illustrated in Figure 4.21 in an exploded view.

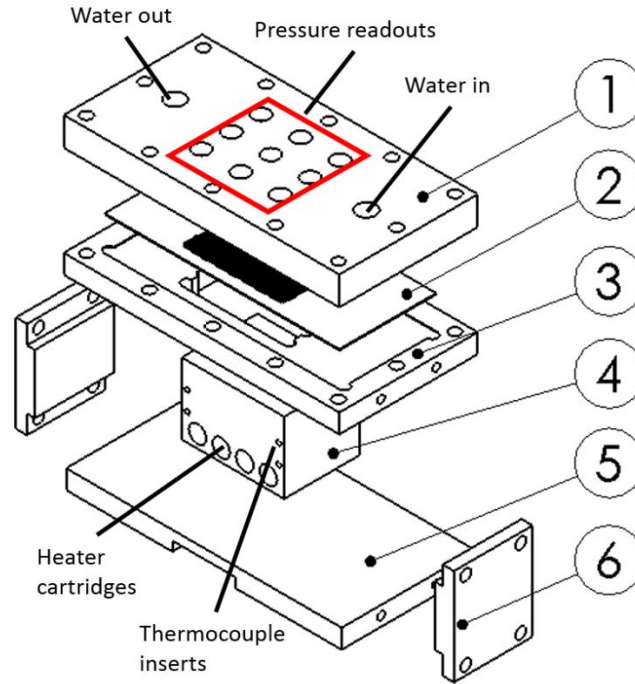


Figure 4.21: Heat transfer test rig: 1) Cover plate equipped with gasket for sealing, pressure taps, and water in/out; 2) Substrate/Baseplate with pin fin array; 3) Bottom plate with insert for the baseplate and heater block; 4) Heater block equipped with inserts for heating cartridges and thermocouples; 5) and 6) supporting pieces.

4.9.1 Heat Test Rig Inputs

4.9.1.1 Constant Power Pump

The flow of water was supplied to the system with a constant power centrifugal pump (EHEIM universal 300, EHEIM, Germany). The pump was fed by a reservoir containing water from a tap. The placement of the reservoir was arranged so that the water level was higher than the pump, aiding the pump by providing a positive head pressure.

4.9.1.2 Heater Cartridges and Transformer

A copper heater block is placed directly under the finned area and is heated by four heater cartridges (HDC00011, Omega, QC, Canada). These heater cartridges have a power of

150W at 120V input. Their heat output was controlled by a variable transformer power source (2kW Variable Transformer Variac 2000VA 0-250V, MASTECH, CA, USA). A silicone-based thermal paste with incorporated conductive particles ($k = 2.1 \text{ W/m}\cdot\text{K}$) (ICE Fusion, Cooler Master, Co., Taipei, Taiwan) was added at the interface between the heater block and the pin fin array baseplate as thinly as possible, as well as within the orifices of the heater cartridges, in order to minimize thermal contact resistance.

4.9.1.3 Flowmeter

Flow rate was measured at the outlet port by a vertical readout flowmeter (8051K39, McMaster-Carr, IL, USA). The water flowrate was set using a manually adjustable valve placed at the water outlet downstream from the flowmeter.

4.9.2 Heat Test Rig Outputs

4.9.2.1 Thermocouples

Two pairs of type T thermocouples (3871K66, McMaster-Carr, IL, USA) placed at the front and back of the heater block read the changes in block temperature as the flow passes over the finned area ($TC_{1,top}$, $TC_{1,bottom}$, $TC_{2,top}$, $TC_{2,bottom}$ in Figure 4.20). Two more type T thermocouples are placed within the inlet and outlet water flow ports in order to read the respective water temperatures (seen in Figure 4.19).

4.9.2.2 Digital Pressure Transducers

Nine pressure taps, inserted directly into the cover plate (seen in Figure 4.21), are available to read the pressures through the finned area using a digital pressure transducer (DPT) (002PD Model 230, Setra, MA, USA). There is also a pair of pressure taps placed

before and after the finned area in order to read the pressure loss through the whole system.

4.9.3 Thermal Performance Calculations

The thermal performance of the pin fin arrays is evaluated by their thermal conductance, a measure of power (heat transfer per unit time) per unit difference in temperature between the fins and the water. The heat transferred by the heater block was measured using two methods; one which estimates the heat transmitted to the water (based on the thermocouples placed in the water flow) and the other which estimates the conduction heat transfer through the heater block (based on the thermocouples placed within the heater block). The sides of the copper heater block were insulated, thus without losses the two heat transfer rates should be equivalent. The power (heat transmitted per unit time) into the water, $\dot{Q}_{water}$, was calculated by:

$$\dot{Q}_{water} = \dot{m}_{water} * C_{p_{water}} * (T_{water,out} - T_{water,in}) \quad (4-15)$$

where $\dot{m}_{water}$ is the mass flow rate of water through the system, $C_{p_{water}}$ is the specific heat of water, and $T_{water,in}$ and $T_{water,out}$ are the inlet and outlet water temperatures, respectively. For comparison, the heat transmitted through the block per unit time, $\dot{Q}_{Cu}$, was calculated using Fourier's Law [22]:

$$\dot{Q}_{Cu} = k_{Cu} A_{block} \frac{\bar{T}_{bottom} - \bar{T}_{top}}{L_{dT}} \quad (4-16)$$

where k_{Cu} is the conductivity of the copper block, L_{dT} is the vertical distance between the thermocouples, A_{block} is the cross sectional area of the copper block in the plane perpendicular to the direction of heat flow, and $\bar{T}_{bottom}$ and $\bar{T}_{top}$ represent the average temperature readings between the two bottom and top thermocouples within the heater

block. This method of calculating the heat transmitted through the block is only admissible because the temperatures read at points 1 and 2, as referenced in Figure 4.20, were found to be close to the same temperature. As such, it could be assumed that the temperature across the cross section of the heater block was constant.

The pin fin arrays were sprayed to be staggered in relation to the flow direction in order to promote mixing of the fluid, as is shown in Figure 4.22.

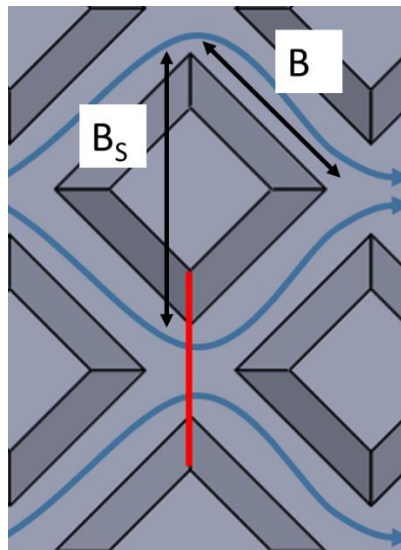


Figure 4.22: Staggered fin arrangement used for the pin fin arrays showing the direction of the flow around the fin and in red the cross sectional line where the characteristic flow area is taken.

The reference geometry used for the fin performance calculations is also shown in Figure 4.232 and Figure 4.23. All dimensions were taken in the plane perpendicular to the incoming flow.

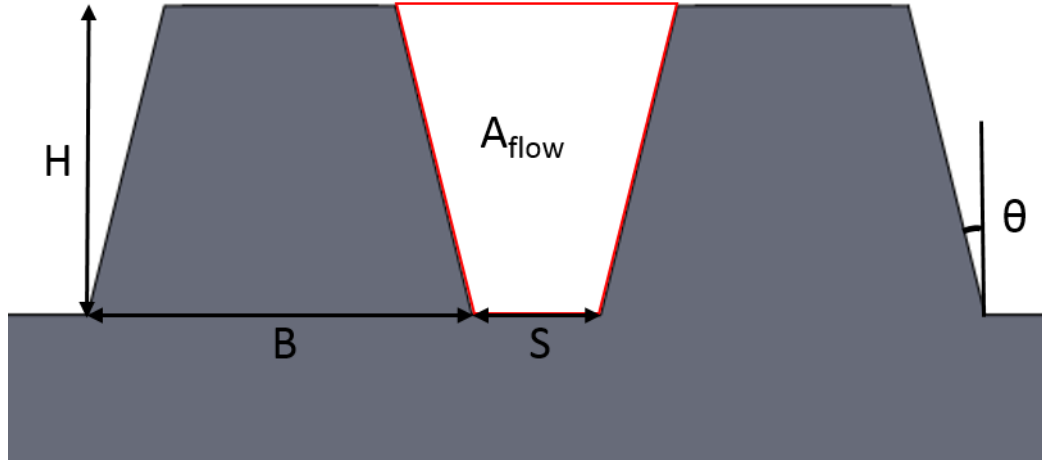


Figure 4.23: Fin cross section showing reference geometry used for the fin performance calculations where B is the width of the base of the fin, S is the width of the section between fins, H is the height of the fins, θ is the angle of the fin, and A_{flow} is the flow area between fins.

To compare directly between the finned plates and the non-finned plate, the Reynolds number for each configuration was found for comparison on a non-dimensional basis. Using the hydraulic diameter, D_h , to characterize the flow channel, the Reynolds number is found by

$$Re = \frac{\rho V_{\text{max}} D_h}{\mu} \quad (4-17)$$

where ρ is the density of the water, V_{max} is the maximum flow velocity, and μ is the viscosity of water. The hydraulic diameter is calculated from the fin geometry as

$$D_h = \frac{4A_{\text{flow}}}{P_{\text{flow}}} = \frac{2A_{\text{flow}}}{S + \frac{H}{\cos \theta} + H \tan \theta} \quad (4-18)$$

where A_{flow} is the flow area between fins, and P_{flow} is the length of the perimeter of this area. These values are taken with the dimensions at the narrowest junction between fins,

which corresponds to the area with the highest flow velocity. The flow area can be found by

$$A_{flow} = (S + H \tan \theta)H \quad (4-19)$$

and the velocity of the flow is found by

$$V_{max} = \frac{\dot{m}_{water}}{A_{flow} N_{fins,w} \rho} \quad (4-20)$$

where $N_{fins,w}$ is the total number of fins along the width of the heat sink.

Next, in order to characterize the performance of the fins, it is useful to reduce the pin fin array to its equivalent thermal circuit, as shown in Figure 4.24. Here the thermal resistances are reduced to the resistance of the finned area, R_{fin} , and the unfinned area, R_{unfin} .

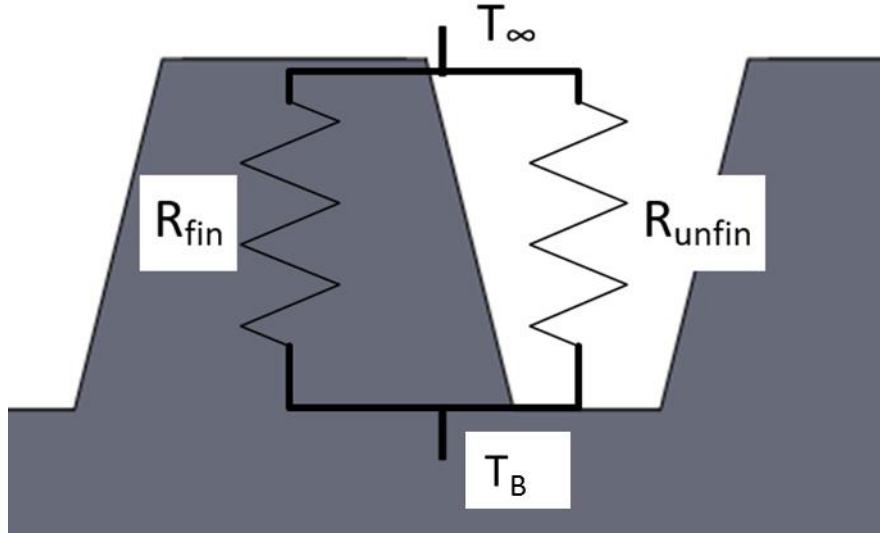


Figure 4.24: Thermal resistance circuit of the heat sink fins.

From this, the equivalent thermal resistance of the heat sink can be written:

$$\frac{1}{R_{eq}} = \frac{1}{R_{fin}} + \frac{1}{R_{unfin}} \quad (4-21)$$

This leads to the calculation of the thermal conductance, UA , which is written

$$UA = \frac{1}{R_{eq}} = hA_{tot}\eta_o \quad (4-22)$$

where h is the convection coefficient, η_o , is the surface efficiency of the pin fin array, and A_{tot} is the total heat transfer area, divided into the area of the fin surface, A_{fin} , and the unfinned area, A_{unfin} , as seen in

$$A_{tot} = A_{fin} + A_{unfin} = \frac{4(B_s - H \tan \theta)H}{\cos \theta} N_f + (WL - N_f B_s^2) \quad (4-23)$$

wherein B_s now denotes the length of the base on the side of the fin, as opposed to the transverse cross-section previously used, N_f is the total number of fins, and W and L are the width and length of the area of the baseplate covered by fins. Important to note is that the area of the tip of the fin is neglected as it is in direct contact with the acrylic cover plate, and therefore there is no heat transferred to the water across this surface. Furthermore, it has been noted that allowing the convective fluid to flow over the tip of the fin can hinder heat transfer performance by allowing a path for the fluid to bypass the bulk of the fins convective surface area [ref]. The surface efficiency is calculated by

$$\eta_o = 1 - \frac{A_{fin}(1 - \eta_f)}{A_{tot}} \quad (4-24)$$

where η_f is the efficiency of the individual fins. The fin efficiency is calculated using a relation found in Incropera et al. [22] for triangular pin fins with a circular base, adapted in the work of Cormier et al. [17] to suit the pyramidal shape of the fins produced by CGDS. This approximation for fin efficiency was chosen as it is the closest match to the

fin geometry seen in this work. For comparison, the fin efficiency for rectangular pin fins was also used, and demonstrated very little deviation from the values obtained by the pyramidal pin fin efficiency calculation. Furthermore, this deviation had almost no effect on the final fin conductance calculations. Fin efficiency is therefore calculated with

$$\eta_f = \frac{2I_2(2mH)}{mHI_1(2mH)} \quad (4-25)$$

and

$$m = \sqrt{\frac{4h}{k_{Cu}B_s}} \quad (4-26)$$

where I_1 and I_2 are the first and second order modified Bessel functions of the first kind.

The convection coefficient of the fin array is calculated as

$$h = \frac{\dot{Q}_{water}}{\Delta T A_{tot} \eta_o} \quad (4-27)$$

where ΔT is the approximation of temperature difference between the base of the fins and the water. This approximation for temperature difference can take two forms depending on the surface heat flux conditions; either the surface provides a constant heat flux to the flow, or it has a constant surface temperature, as illustrated in Figure 4.25. It should be noted that the conductivity between the base of the fin and the fin itself could be affected by the presence of residual imbedded grit particles from the grit blasting process, however this was neglected due to the low frequency of such occurrences.

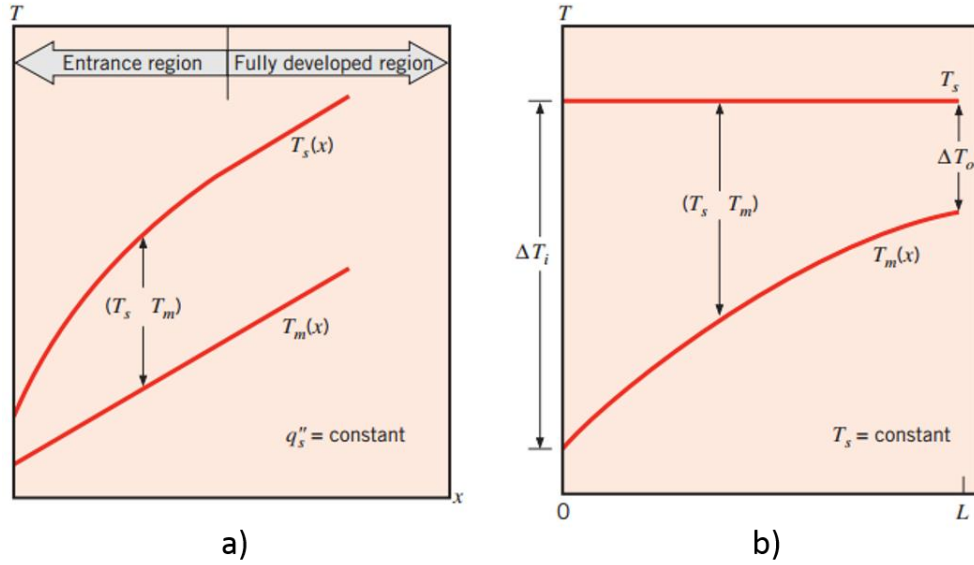


Figure 4.25: Temperature variation of an internal flow over a surface with a) constant heat flux and b) constant temperature [22] (© 2011 Wiley Books).

In the case of a constant heat flux, once in the fully developed region, the approximation for temperature difference takes the form of

$$\Delta T = T_s - T_m \quad (4-28)$$

where T_s is the surface temperature and T_m is the mean water temperature. This approximation becomes more accurate the shorter the entry region compared to the fully developed region. For the case of constant surface temperature, the temperature difference can be approximated as the log-mean temperature difference, which is calculated using

$$\Delta T = \Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} \quad (4-29)$$

where ΔT_1 and ΔT_2 represent the difference in temperature between the surface of the baseplate and the water at the flow inlet and outlet, respectively. As the thermocouple

readings were taken within the heater block, and not at the surface of the baseplate, the resulting surface temperature had to be interpolated using the concept of thermal resistance. Knowing the heat flux and the temperature of the block, T_{block} the surface temperature, T_s , is found using

$$T_s = (R_{Cu} + R_{tc})\dot{Q}_{Cu} + T_{block} \quad (4-30)$$

where R_{Cu} is the resistance due to the copper block and baseplate, and R_{tc} is the thermal contact resistance between the block and the bottom of the pin fin array baseplate. The thermal contact resistance in this work was taken to be $1.0 \times 10^{-5} \text{ K}\cdot\text{m}^2/\text{W}$ as this was found to be an average value for a copper interface with thermal paste [95]. A sensitivity study showed that slight variation in the value for thermal contact resistance had little effect on the resulting fin conductance.

In order to compare which fin configuration transfers heat more efficiently with respect to the associated pressure drop, the thermal performances were compared while also considering the required pumping power [16], [25], [26]. This gives an indication of which fin configuration has a higher thermal conductance for a given pumping power (pressure drop). The required pumping power, e_v , is found as

$$e_v = \frac{\dot{V}_{flow}\Delta P_{fin}}{\eta} \quad (4-31)$$

where $\dot{V}_{flow}$ is the volumetric flowrate of water, ΔP_{fin} is the pressure drop across the finned area, and η is the pump efficiency. As all fin configurations in this work are subjected to the same pump, and therefore the same pump efficiency, all values of effective pump power are left as a function of pump efficiency.

4.10 CFD Modelling Details

4.10.1 Computational Domain

A computer model was developed to verify the validity of the surface heat transfer assumption made in Section 4.9.3 for the calculation of results from the heat transfer test rig. This was done using the commercially available Computational Fluid Dynamic (CFD) package in ANSYS Fluent 16. The three-dimensional symmetric model was adapted and simplified for ease of meshing and to reduce processing cost, but the major geometrical characteristics of the assembly remained the same as the experimental setup. The dimensions of the simplified model are shown in Figure 4.26, matching the same bulk dimensions as the experimental heat test rig.

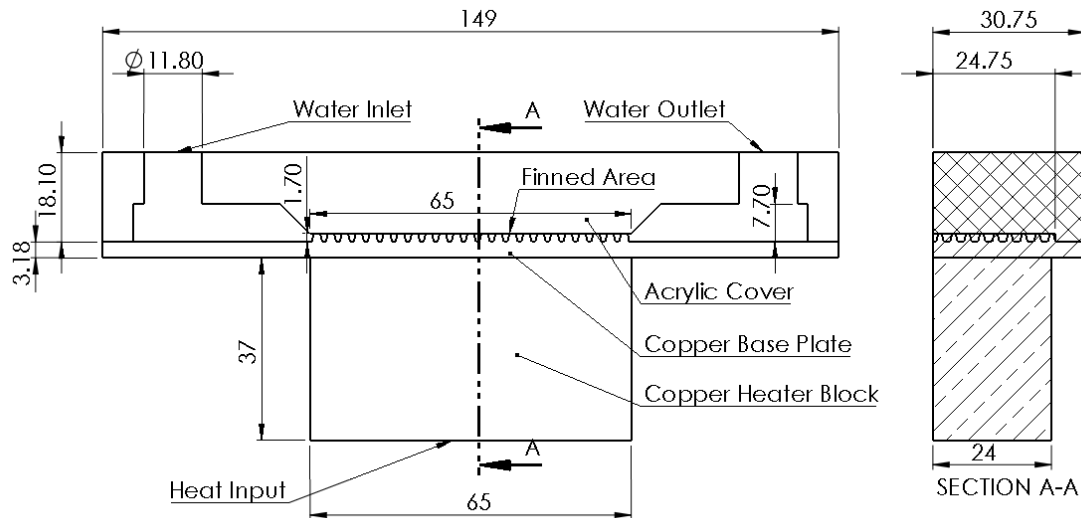


Figure 4.26: Dimensions of the computational domain (mm).

As boundary conditions, the water inlet was chosen to be a specified velocity inlet and a specified temperature, to match the experimental values. This condition can only be used for incompressible flows, which is the case for this study. The outlet used the outflow

boundary condition with a weighting of 1 as there is only the single outlet, with ambient initial values for back pressure and temperature. The heat flux was input at the bottom of the copper heater block. The block does not incorporate the shape of the heater cartridge orifices, for ease of meshing and computation, and therefore the dimensions of the block were chosen such that the heat inlet surface bisects the location where the heater cartridges would be in reality. The heat flux input, q'' , used was calculated from the experimental data using

$$q'' = \frac{\dot{m}_{water} * C_{p_{water}} * (T_{water,out} - T_{water,in})}{A_{block}} \quad (4-32)$$

which yields the total heat flux that was input to the water. The outer walls for all bodies were chosen to be perfectly insulated since the full amount of heat from the water is being used as the input at the base of the heater block, and thus any heat losses were already taken into account in the experimental setup. The interfaces between parts used the no slip condition for the case of fluid flow over the surface, and constant heat flux for the case of heat transfer across the interface. The symmetry axis utilizes the no shear slip wall condition for fluid flow, and no heat flux boundary condition for heat transfer. A summary of the boundary conditions used in this model are shown in Table 4.2, wherein n denotes the direction normal to the boundary, and the subscripts 1 and 2 denote the first and second side of the interface boundary.

Table 4.2: Boundary Conditions Used in the Model

Location	P	V	T
Water inlet	Ambient pressure	Specified velocity	Specified temperature
Water outlet	Ambient pressure	Outflow weighting = 1	Ambient temperature
Heater block bottom	NA	NA	$k \frac{\partial T}{\partial n} = -q$ (specified)
Outer walls	NA	0	$\frac{\partial T}{\partial n} = 0$
Body interfaces	$\frac{\partial P}{\partial n} = 0$	0	$\left(\frac{\partial T}{\partial n}\right)_1 = \left(\frac{\partial T}{\partial n}\right)_2$
Symmetry	$\frac{\partial P}{\partial n} = 0$	$\frac{\partial v}{\partial n} = 0; v_n = 0$	$\frac{\partial T}{\partial n} = 0$

4.10.2 Domain Meshing

The computational domain of the heat transfer test rig was meshed using the ANSYS WB Meshing Tool. The model is comprised of 4 different bodies that needed to be meshed: the copper heater block, the copper pin fin array, the water flowing through the heat test rig, and the acrylic cover plate. In order to properly mesh the complex features of the finned plate, and to obtain better accuracy in areas of high flow complexity, mesh refinement was done throughout the finned area and the surrounding water. The mesh study showed that by refining the mesh, the same flow results were obtained, but the finer mesh was used to obtain a higher level of detail. An unstructured three-dimensional tetrahedral mesh (Figure 4.27) was used for ease of conforming to the complex geometries encountered in the model.

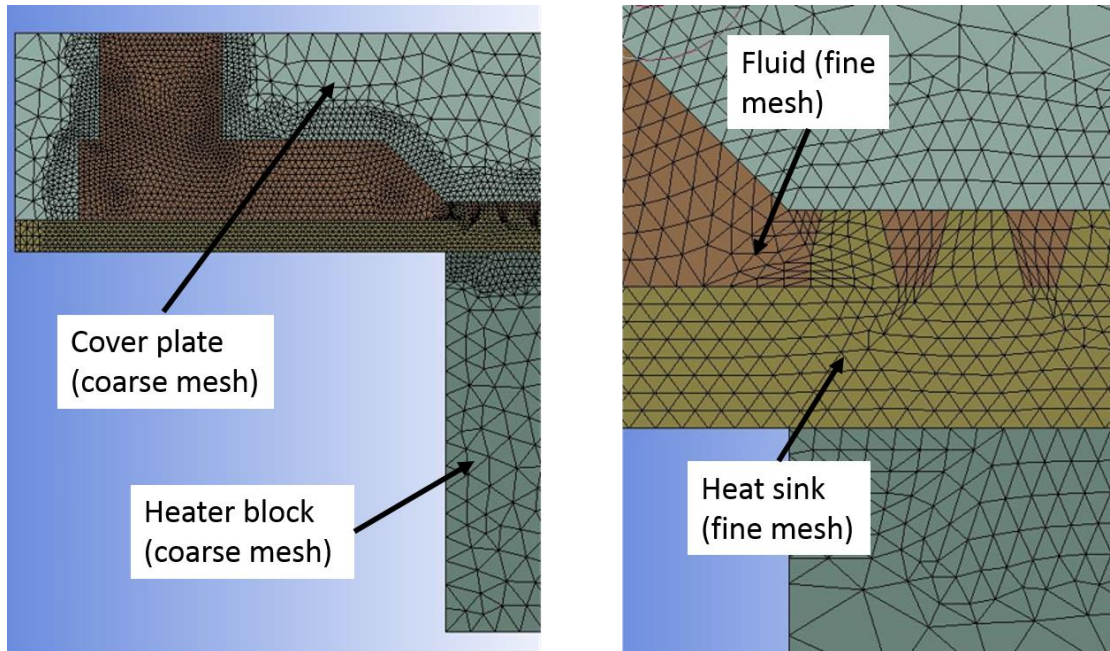


Figure 4.27: Tetrahedral mesh used in the computational domain showing areas of mesh refinement (left) and a zoom of the entry to the finned area (right).

A grid convergence analysis was done, refining the mesh multiple times until the computed results at the water exit plane ceased to change from one mesh density to the next. The final mesh consists of 3701734 cells with edge lengths that ranged between $2.3784 \times 10^{-5} \text{m}$ and $4.7568 \times 10^{-3} \text{m}$, comprising a minimal orthogonal quality of 1.48762×10^{-5} and a maximum aspect ratio of 1.25667×10^2 . These values are quite far outside the ideal ranges specified by ANSYS documentation (over 0.01 for orthogonal quality and under 20 for aspect ratio). However, it has been noted that even with high aspect ratios convergence can be achieved, and less than 0.1% of cells had orthogonal quality below 0.07, signifying that there were in fact very few cases of low quality cells. The presence of these defects was attributed to the high complexity of the model being meshed, and were not expected to affect the solution due to the relatively low prevalence of these issues.

The solution was iterated until convergence was achieved by showing that the residuals of the calculation had decreased by at least three orders of magnitude.

4.10.3 Governing Equations

4.10.3.1 Fluid Flow

The fluid flow in the CFD model uses the Navier-Stokes equations for continuity, momentum, and energy conservation expressed as:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = S_m \quad (4-33)$$

,

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla P + \nabla \left(\mu \left[\nabla \vec{v} + \nabla \vec{v}^T - \frac{2}{3} \nabla \cdot \vec{v} I \right] \right) + \rho \vec{g} + \vec{F} \quad (4-34)$$

and

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + P)) = \nabla \cdot \left(k_{eff} \nabla T - \sum_j h_{s_j} \vec{J}_j + (\bar{\tau}_{eff} \cdot \vec{v}) \right) + S_h \quad (4-35)$$

respectively, where ρ is the density, t is time, $\vec{v}$ is the directional velocity, P is the static pressure, μ is the viscosity, I is the unit tensor, the term $\rho \vec{g}$ represents the gravitational body force, $\vec{F}$ is the external body force, $\vec{J}_j$ is the diffusion flux of term j , and the term $\bar{\tau}_{eff} \cdot \vec{v}$ represents the viscous dissipation. In the energy equation

$$k_{eff} = k_f + k_t \quad (4-36)$$

where k_{eff} is the effective conductivity which is made up of the fluid thermal conductivity, k_f , and the turbulent thermal conductivity, k_t , determined by the turbulence model. Then

$$E = h_s - \frac{P}{\rho} + \frac{v^2}{2} \quad (4-37)$$

where h_s is the sensible enthalpy and is determined by

$$h_s = \sum_j Y_j h_{s,j} + \frac{p}{\rho} \quad (4-38)$$

for an incompressible flow where Y_j is the mass fraction of species j and

$$h_{s,j} = \int_{T_{ref}}^T C_{p,j} dT \quad (4-39)$$

where T is the temperature, $C_{p,j}$ is the specific heat of species j , and T_{ref} is 298.15 K. In these equations the terms S_m and S_h represent added mass and volumetric heat sources, respectively, and are eliminated. Furthermore, as there is only a single species (water), all terms referring to multiple species and mass fractions are simplified. External forces applied to the fluid are also eliminated.

For the modelling of the fluid flow, the simple viscous-laminar model was used. The reason for this is that in calculations derived from the experimental data, the Reynolds numbers in the flow regimes to be tested were below the turbulent transition point seen in previous testing of pin fin arrays ($Re < 500$) [30]. Since turbulence models, such as the $k-\epsilon$ model, are only suitable for fully turbulent flows [96], it was therefore determined that no turbulence model should be used.

In the current study, a Green-Gauss node based gradient method was used to simulate the water flow and heat transfer. The gradient method is needed for constructing values of a scalar at the cell faces, and for computing secondary diffusion terms and velocity derivatives. The Green-Gauss node based gradient method constructs the solution for a

given cell from the weighted average of the cell values surrounding the nodes. This gradient method was selected as it is more accurate for unstructured tetrahedral meshes than the other available cell based methods [96]. In addition, the pressure, momentum, and energy were modelled using second order accuracy spatial discretization and the default SIMPLE scheme was used to solve the pressure-velocity coupling. The SIMPLE scheme is a widely used algorithm for solving the Navier-Stokes equations, which couples the pressure and velocity to enforce mass conservation and obtain the pressure field. SIMPLE is generally used for uncomplicated steady-state problems, such as a laminar flow, and was therefore deemed sufficient for the present case [96].

4.10.3.2 Heat Transfer through Solids

Heat transfer through solid parts was calculated using the following energy transport equation:

$$\frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\vec{v} \rho h) = \nabla \cdot (k \nabla T) + S_h \quad (4-40)$$

where k is the conductivity. The volumetric heat sources are again eliminated as there are no such occurrences. Furthermore, the second term on the left-hand side of the previous equation can be eliminated, as it relates to convective energy transfer due to rotational or translational motion of the solids, and there is no such motion in the model. In this case, sensible enthalpy is

$$h = \int_{T_{ref}}^T c_p dT \quad (4-41)$$

where T_{ref} is again 298.15K.

4.10.3.3 Physical Properties

The various bodies of the model were made of copper, water, and acrylic, and the physical properties used for these are shown in Table 4.3. The properties of water and copper were taken from the ANSYS Fluent database, while the properties of acrylic were taken as average values found in the literature [97].

Table 4.3: Material Properties used in CFD Model

Body Material	Density [kg/m³]	Specific Heat [J/kg·K]	Thermal Conductivity [W/m·K]	Viscosity [kg/m·s]
Copper	8978	381	387.6	NA
Water	998.2	4182	0.6	0.00103
Acrylic	1180	1500	0.189	NA

4.11 Characterization Procedures and Analysis Equipment

4.11.1 Deposition Efficiency

The DE of the spray process for the production of coatings and pin fin samples were calculated using two different methods, depending on the powder being used. For the pin fin arrays, which were all made with as-received powder, powder feed rate measurements were taken and DE was determined the conventional way by common lab practices. For the coatings made with reclaimed powder, limitation on the quantity of powder available made it impossible to do powder feed rate measurements, and therefore a different method was adopted.

4.11.1.1 Deposition Efficiency of Pin Fin Arrays

For the case of all pin fin samples, the DE was calculated using

$$DE = \frac{\Delta m_{substrate}}{m_{sprayed}} \quad (4-42)$$

where $\Delta m_{\text{substrate}}$ is the mass added to the substrate by the spray and m_{sprayed} is the total mass of powder sprayed while the nozzle is over the substrate. The mass sprayed can be calculated by

$$m_{\text{sprayed}} = t_{\text{spray}} \cdot PFR \quad (4-43)$$

where PFR is the powder feed rate. The powder feed rate is found by averaging multiple measurements of the difference in mass of a receptacle before and after it has been connected to the powder feeder for a minute. To verify that there was a consistent powder feed rate, measurements were taken both before and after the spray. The time of spray is calculated by equation (4-2).

4.11.1.2 Deposition Efficiency for Coating Comparison

For the case of the coatings to be made with as-received powder and reclaimed powder, DE was determined by placing a known amount of powder into the powder feeder and running the system continuously over the substrate until no powder remained. This ensured that an exact known amount of powder was projected onto the substrate, and therefore DE could be determined by taking the ratio of mass added to the substrate and the known mass of powder run through the system. The calculation is as follows:

$$DE = \frac{\Delta m_{\text{substrate}}}{m_{\text{Cu,feeder}}} \quad (4-44)$$

where $\Delta m_{\text{substrate}}$ is the mass added to the substrate by the spray and $m_{\text{Cu,feeder}}$ is the total mass of powder inserted into the powder feeder. The reason for using this method was mainly due to limitations in the availability of the reclaimed powder. The method described in the previous subsection requires a powder feed-rate measurement, which would have wasted some of the already limited supply of reclaimed powder. Therefore, it

was determined that the method of spraying the whole contents of the powder feeder would be the most effective use of the powder, as no powder is wasted in the measurement of DE. In order to mitigate the loss of powder in the feeding lines for these measurements, as-received powder was fed through the system prior to the spraying of the reclaimed powder so as to saturate the lines and allow for the most possible recycled powder to actually impact the substrate.

4.11.2 Powder Reclamation Efficiency

While the main goal of the work was not to devise an apparatus efficient in powder reclamation, it was in the best interest of this work to reclaim as much impacted powder as possible per spray, and as such powder reclamation efficiency was calculated to see whether reclamation was affected by any changes to the reclamation box layout. The reclamation efficiency, RE , is simply

$$RE = \frac{m_{reclaimed}}{m_{undeposited}} \quad (4-45)$$

where $m_{reclaimed}$ is the mass of powder that was reclaimed from the box, and $m_{undeposited}$ is the total mass of powder that did not deposit on the substrate or mask. The mass that was not deposited on the substrate and mask can be found by

$$m_{undeposited} = t_{box} * PFR - \Delta m_{substrate} - \Delta m_{mask} \quad (4-46)$$

where t_{box} is the amount of time that powder is fed within the box, measured from when the powder feeder is turned on to when it's turned off, and Δm_{mask} is the mass added to the mask.

4.11.3 Powder Morphology Analysis

The powder was analyzed both before and after spraying in order to identify the morphological differences between the as-received powder and the reclaimed powder. This was done using a scanning electron microscope (SEM) (EVO-MA10, Zeiss, UK), which is shown in Figure 4.28. This equipment has a maximum acceleration voltage of 30 kV and a theoretical maximum resolution of 2 nm point to point. It also has secondary and backscatter electron detectors.



Figure 4.28: Scanning electron microscope.

4.11.4 Single Particle Impact Analysis

A study of individual particle impacts was performed to provide qualitative information on the differences between the deposition behavior of as-received particles and reclaimed particles. The powder feed was cut during the spray so that only the residual particles would impact the substrate, and the nozzle traverse speed was kept the same as for other spray trials to ensure similar deposition conditions. For these trials, the substrates were

polished to minimize surface roughness and allow clear and unobstructed observation of the impacted particles. The resulting single particle impacts were analyzed using the SEM.

4.11.5 Powder Velocity Measurements

The in-flight velocities of the as-received and reclaimed powder particles were measured using the ColdSprayMeter (CSM) eVOLUTION (Tecnar Automation Ltd., St-Bruno, Canada), shown in Figure 4.29. This system calculates the particle velocities using a continuous 2.4 W ($\lambda=810$ nm) laser and a dual slit photomask. In-flight particle velocities are calculated by measuring the time difference between the blocking of the two photomasks, signifying that a particle has flown past. For the purposes of this study, velocity measurements were taken at a distance of approximately 10 mm downstream from the CGDS nozzle, as this corresponds to the standoff distance between the nozzle and substrate for the coating deposition tests. Testing was run until a data set of 1000 particles was achieved, and the results are represented in box and whisker plots.

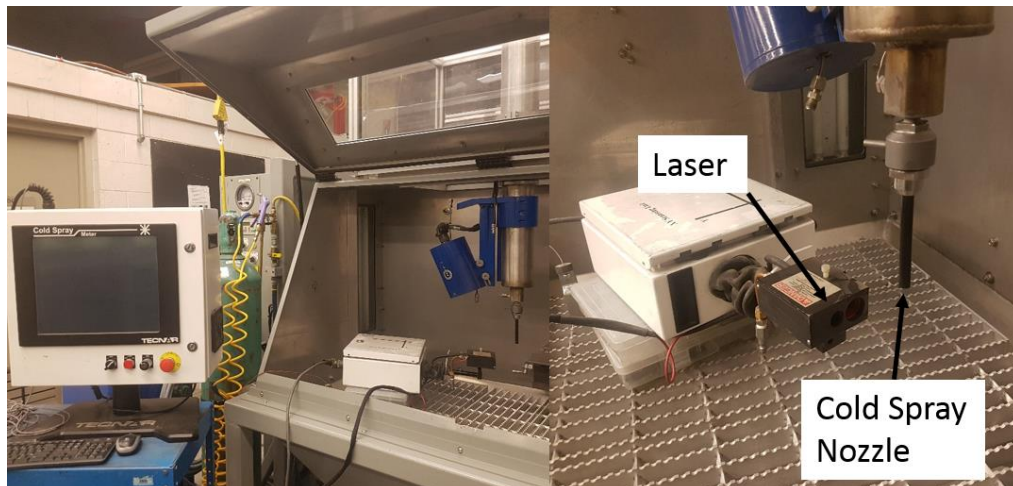


Figure 4.29: ColdSprayMeter showing setup in spray chamber (left) and the laser and nozzle (right).

4.11.6 Coating and Fin Characterization

4.11.6.1 Sample Preparation and Polishing

All fins and coatings were cut for cross sectional analysis using a precision rotary saw (Struers Secotom-10, Struers Ltd., Canada) equipped with a silicon carbide blade for cutting non-ferrous materials. Cutting parameters were chosen per manufacturer specifications. After cutting, the samples were mounted using a cold mounting procedure. The mounting resin was a thermosetting mixture of epoxy resin and epoxy hardener (Anamet, Canada). Finally, the samples were polished using a Struers TegraPol (Struers Secotom-10, Struers Ltd., Canada).

4.11.6.2 Fin and Coating Visualization

Coatings and fins were analyzed using an SEM and an optical microscope (VHX-2000E, Keyence, Canada). The optical microscope, seen in Figure 4.30, has a computer controlled XY stage, and a maximum magnification of 1000X. A built-in software allows for autofocus and image stitching, creating fully focused 2D images over large areas. It can also produce 3D imaging by using a focused depth composition function, making it possible to obtain full 3D representations of the finned area.



Figure 4.30: VHX-2000E digital optical microscope.

4.11.6.3 Etching

In order to view the particle boundaries, coatings were etched with a solution of 25vol% FeCl_3 -75vol% ethanol for a duration of 5 seconds. FeCl_3 is a commonly used etchant for copper based materials [98].

4.11.6.4 Micro-hardness

The coating micro-hardness was measured using a Vickers micro-hardness tester (Struers Duramin-1, Struers Inc., Cleveland, OH, USA), presented in Figure 4.31, with a 500g load and a dwell time of 10s. The distance between indentations was kept above 3 times the length of the previous indentation, to avoid stress field effects. The reported values are the average of 10 micro-hardness tests per sample.



Figure 4.31: Vickers micro hardness tester.

4.11.6.5 Porosity

Porosity measurements were achieved through analysis of the stitched optical images over fin or coating cross sections. An open source image processing software (ImageJ) was used to transform images into binary form, rendering the copper fins or coatings white and the porosities black. A built-in process was then used to measure the percentage of porosity for a highlighted area in the fin or coating, by returning the percentage area of either shade. An example of the binary image produced is shown in Figure 4.32.

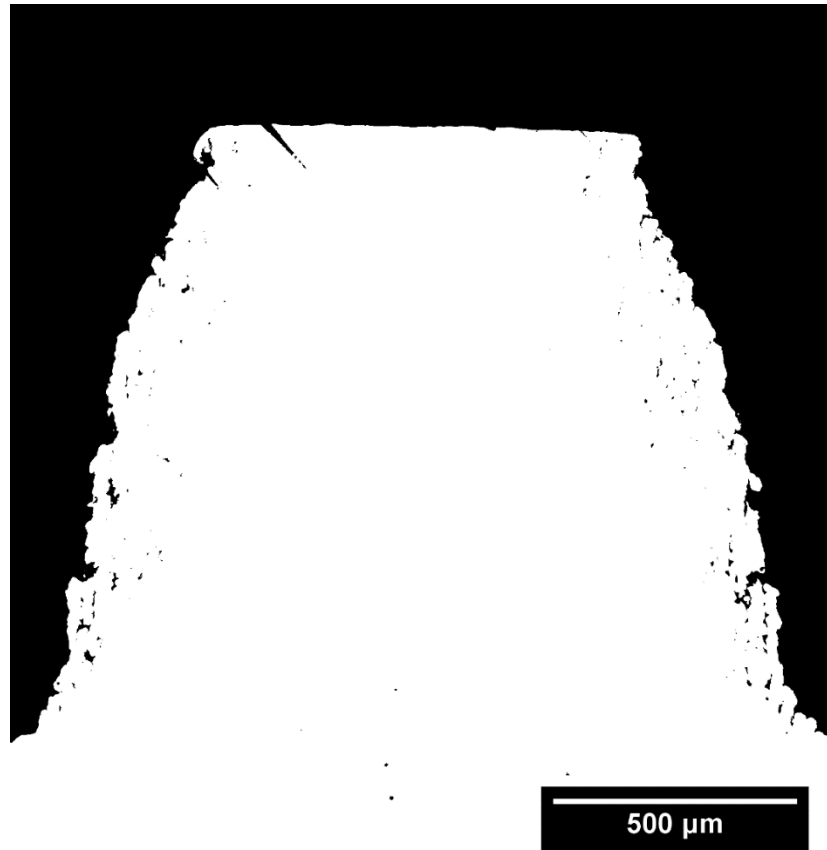


Figure 4.32: Binary image of a pin fin cross section produced by ImageJ for porosity measurement.

4.11.7 Post Spray Annealing Procedure

The copper pin fin arrays and test coatings were annealed after spraying, in order to restore the samples to the near-bulk thermal properties of copper, as seen in the work of Sudharshan et al. [99]. Annealing was done in a Lindberg 4880 Watt furnace (Lindberg model 51442, Lindberg, WI, USA), as seen in Figure 4.33, at 300°C for 4 hours in air and the samples were then allowed to slowly cool within the furnace. The samples were then cleaned using acetic acid 5vol% to remove any excess copper oxides which may have been formed [100].



Figure 4.33: Lindberg furnace used for annealing of the copper coatings and fins.

4.11.8 Surface Roughness Measurements

The substrates surface roughness measurements were taken using a portable surface roughness gauge, or profilometer, (SRG-4000, Phase II Plus, NJ, USA), shown in Figure 4.34. This profilometer comes with a diamond stylus sensor that provides precision accuracy down to $0.001\mu\text{m}$. The range of measurable roughness was from 0.005 to $16\mu\text{m}$ and the tracing length of the stylus has 1.3 to 17.5 mm in length.

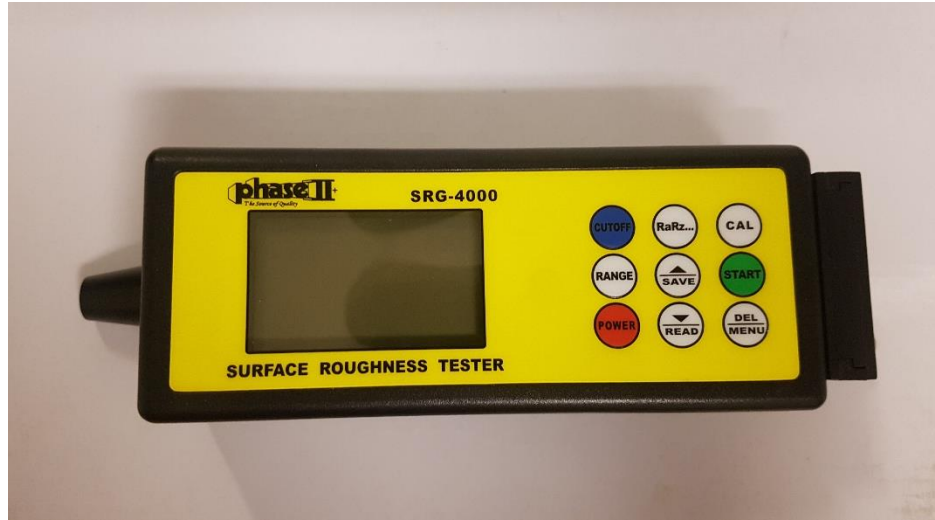


Figure 4.34: Portable surface roughness gauge.

5 RESULTS AND DISCUSSION

5.1 Pin Fin Array Production and Characterization

5.1.1 CGDS Parameter Selection

A matrix of tests was carried out to determine the most cost effective parameters for the copper pin fin arrays production on copper substrates. The initial assumption was that higher DE would translate into lower manufacturing costs. Since higher temperatures and pressures have been shown to result in higher DE, the initial matrix of parameters was framed to reach the limits of the CGDS system: 500°C and 3.45MPa. However, early in testing, it was found that spraying copper at 500°C caused the CGDS nozzle to clog. Therefore, the matrix of parameters that was used ranged from 200-400°C and 2.06-3.45MPa, as 400°C did not result in nozzle clogging.

The propellant gas selected for this work was nitrogen, a standard low-cost gas used in CGDS. High nozzle standoff distances have been shown to lower DE [89], and therefore a default 10 mm standoff distance was chosen for this application. Traverse velocity, while causing little direct effect on DE [76], [88], was adjusted to produce fins of the same height for each set of spray parameters, to ensure accurate comparison in the cost model. It was expected that the shape of fin buildup would dominate DE effects, and that the effect of substrate preheating from lower traverse velocities would be negligible for the range of temperatures used. Other CGDS parameters, such as gun step size, powder feed rate, and feeding gas flow rate, which have not been shown to have an effect on DE were taken as typical values used at the University of Ottawa Cold Spray Lab. The list of the spray parameters used is shown in Table 5.1.

Table 5.1: CGDS Parameters.

Parameter	Value
Gas Temperature	200-400°C
Gas Pressure	2.06-3.45 MPa
Gun Traverse Speed	TBD
Step Size	1 mm
Passes	1-2
Powder Feed Rate	30 g/min
Feeder Gas Flow Rate	0.85 m ³ /h
Powder Feeder and Propellant Gas	Nitrogen
Standoff Distance	10 mm

5.1.2 Fin Height - Traverse Velocity Correlation

The initial spray trials to produce pin fins were done on small samples, using a constant traverse velocity of 50 mm/s. These trials aimed to determine the deposition efficiency on both the substrate and the mask (mesh) when spraying pin fin arrays, as well as the rate of fin height buildup. The resulting substrate and mesh DE are shown in Figure 5.1. It is seen that the DE on both the substrate and the mesh follow similar trends, increasing with temperature and pressure, while demonstrating a slight reduction in DE when switching to a higher temperature but lower pressure.

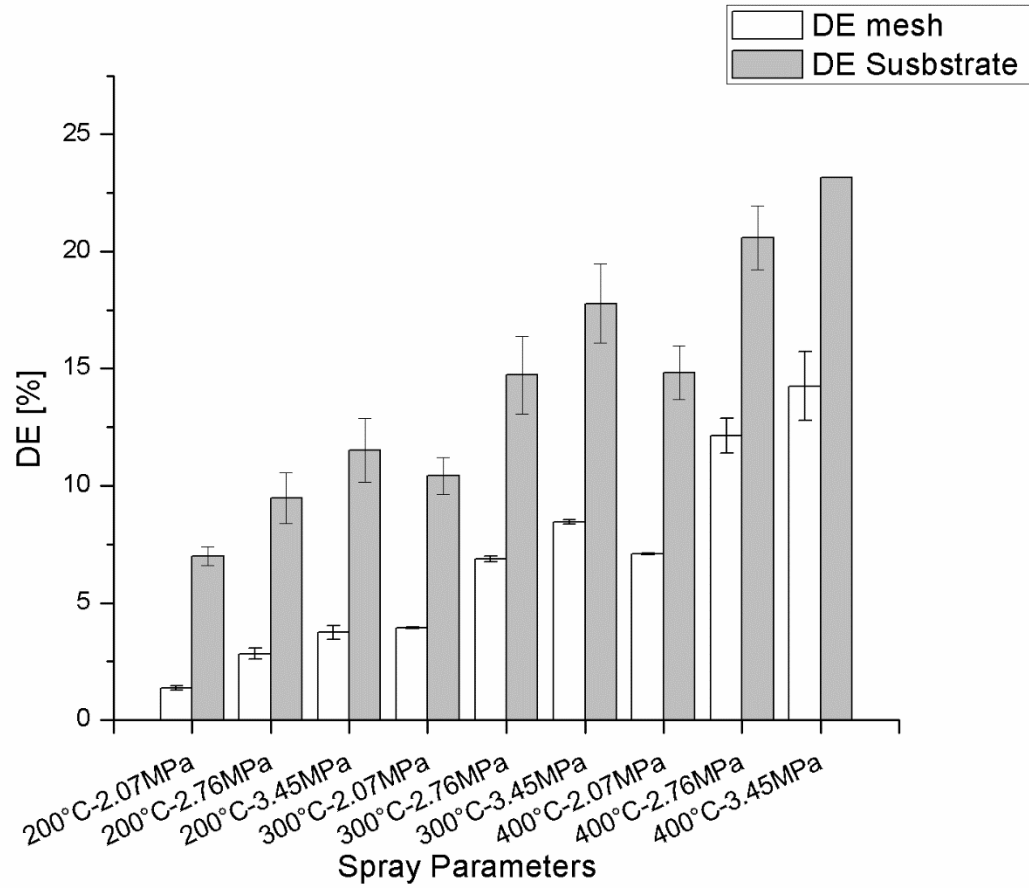


Figure 5.1: Substrate and mesh DE for the production of pin fins arrays with 50 mm/s traverse velocity.

The resulting fin height per pass from these spray trials are shown in Figure 5.2. Here it is shown that the fin heights follow the same trend as the DE, generally increasing with pressure and temperature.

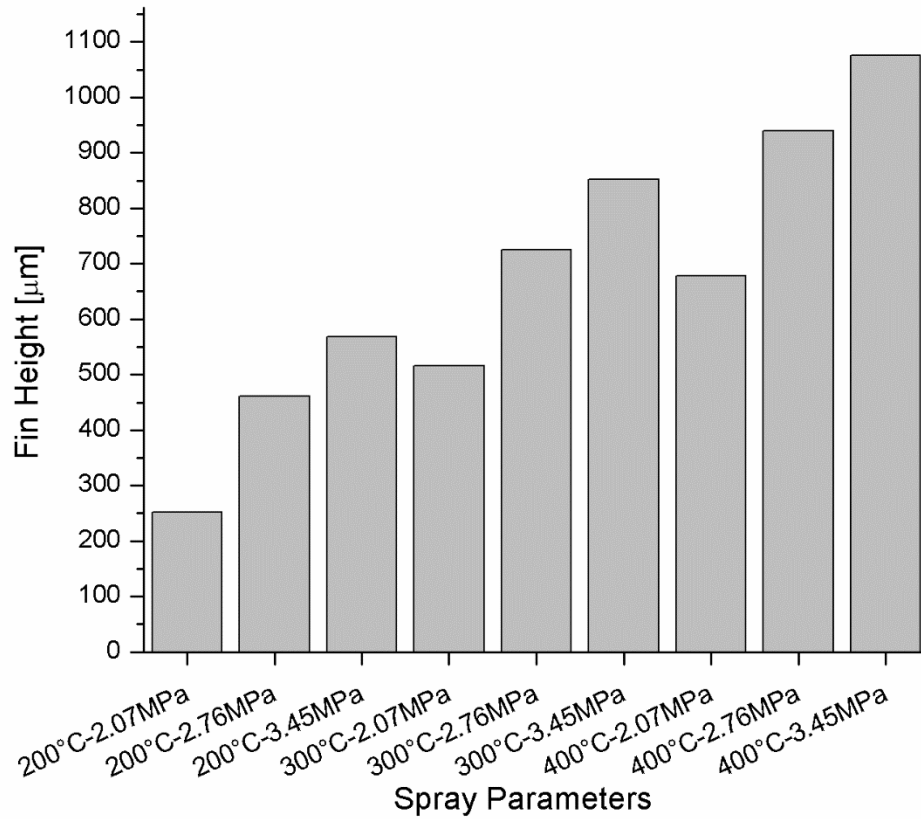


Figure 5.2: Fin height per pass from the production of pin fins arrays with 50 mm/s traverse velocity.

Next, for the production of pin fin arrays with the same height for each set of parameters, it was determined that only a single pass should be used in order to eliminate the possibility of any oxide buildup between passes. Thus, using the previous results of fin height per pass with 50 mm/s traverse velocity, it was possible to use the relation in equation (4-1) to estimate the constant correlating traverse velocity to fin height, c_{V-H} , for each parameter set. With this constant, the traverse velocity necessary for the production of any fin height can be estimated. The constants for each set of parameters

can be found in Table 5.2. To verify the correlation, fins were sprayed to a desired height of 1.5 mm. The traverse velocities used can also be seen in Table 5.2.

Table 5.2: Constant Correlating Traverse Velocity to Fin Height and Traverse Velocity Required for 1.5 mm Fins

Temperature [°C]	Pressure [MPa]	Constant, c_{V-H} [mm ² /s]	Predicted required Traverse Velocity [mm/s]
200	2.07	12.625	8.4
200	2.76	23.075	15.4
200	3.45	28.375	18.9
300	2.07	25.775	17.2
300	2.76	36.225	24.2
300	3.45	42.625	28.4
400	2.07	33.875	22.6
400	2.76	47.0	31.3
400	3.45	53.775	35.9

The heights that were achieved with these traverse velocities are shown in Figure 5.3, with a line indicating the desired 1.5 mm fin height. The fin heights were very close to the desired 1.5 mm, thereby demonstrating the validity of the fin height to traverse velocity correlation. The maximum deviation recorded was 150 μm for the 400°C-2.07 MPa test. These relatively small deviations from the desired heights can be attributed to variations in powder feed rate during the spray, as this is a factor that is known to slightly fluctuate. If the deviation becomes too large between spray runs, it would be a simple matter of reapplying the fin height to traverse velocity correlation iteratively until the correct height is achieved.

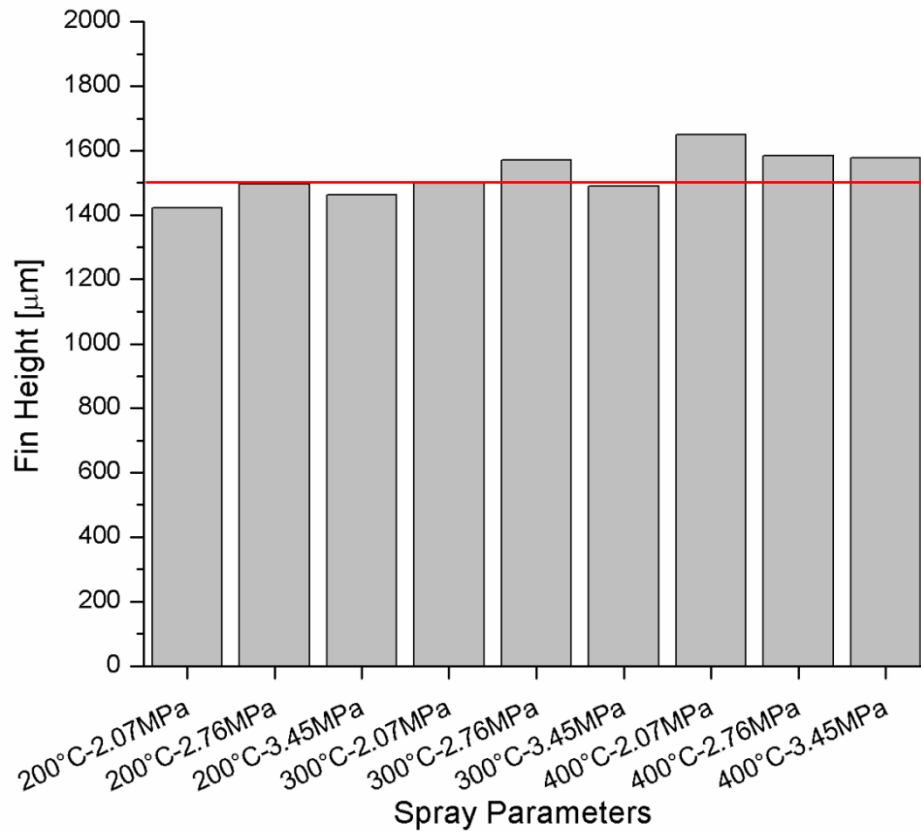


Figure 5.3: Fin height achieved by varying traverse velocity to get 1.5 mm height fins for various CGDS parameters.

The DE measurements found from these 1.5 mm fin test were used for the cost analysis and are shown in Figure 5.4. This shows that there is a slight difference in the DE for the 1.5mm fins compared to that of the fins produced with the 50mm/s traverse velocity.

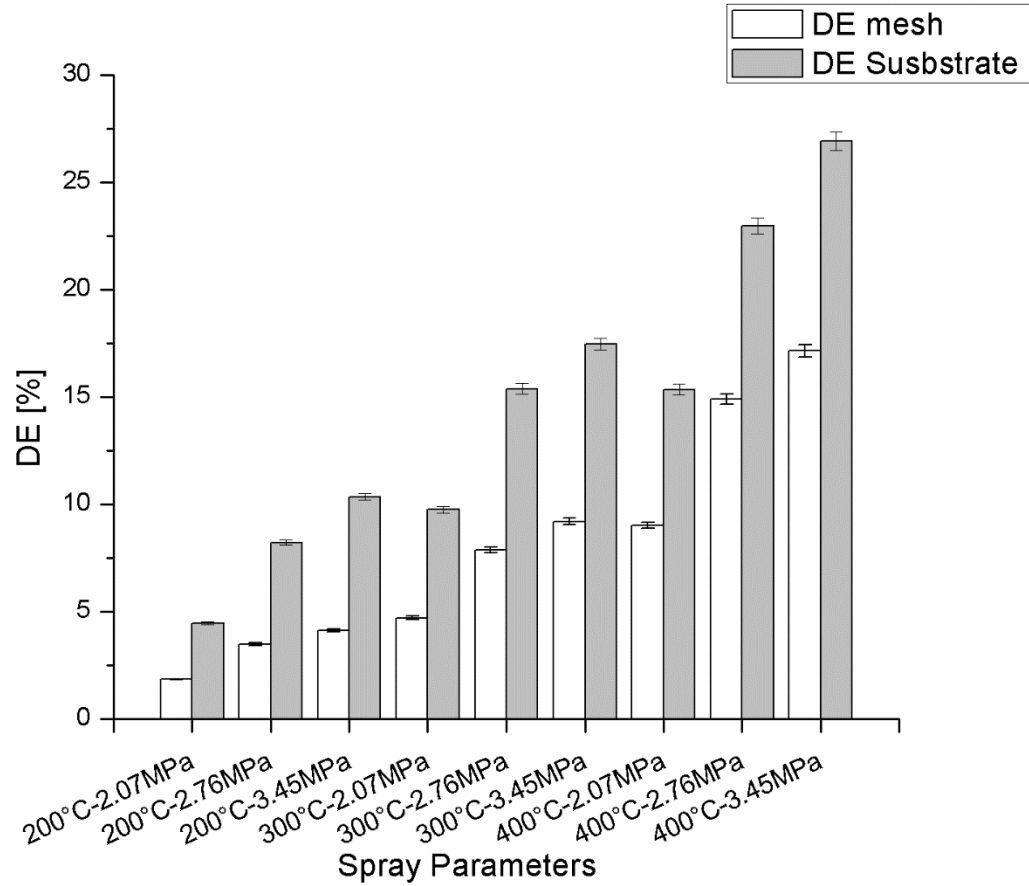


Figure 5.4: DE measurements from spraying 1.5 mm fins.

Examining the values closely, it is revealed that there is a drop in DE for the lower parameters and a jump for the higher parameters. This is expected to be due in part to thermal effects from the change in traverse velocity, but also due to the change in final fin height. As the fin grows through the spray process, the impacting particles no longer experience a flat substrate, but instead hit the slanted face of the growing fin. It has been shown that lowering the particle impact angle lowers DE [101], and this is the case when impacting the fins. For the lowest parameters (Figure 5.5a), the fins were so small in the 50mm/s sprays that this effect was not seen, however for larger fins the resulting impact angle is too small for the lower parameters to overcome. On the other hand, the higher

parameters allow for particles to stick even at low impact angles, and it has been noted that taller fins made with higher parameters start to build out wider (Figure 5.5b). This would provide a better platform for adhesion than the steep sides seen with lower parameters. As mentioned, the lower traverse velocities used in this spray may also be a contributing factor to the rise in DE, as this causes more substrate preheating.

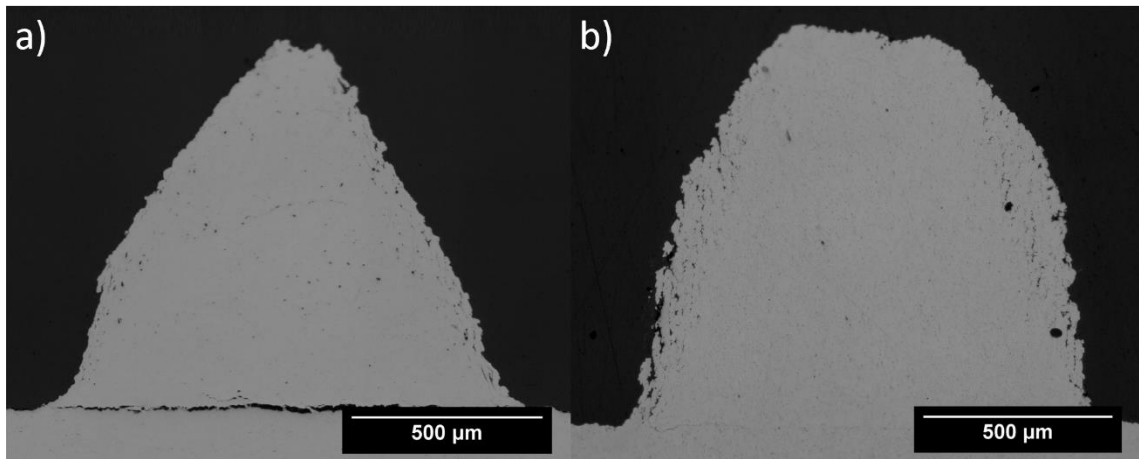


Figure 5.5: Comparison of 1.5 mm fin build shape for a) low parameters and b) high parameters.

5.1.3 Cost Analysis for Parameter Selection

Having verified that the fin height to traverse velocity correlation was suitably accurate for the production of pin fins of uniform height, the next step was to use the cost analysis model to determine the lowest cost spray parameters. The cost analysis was done using the DE recorded from the trials outlined in Table 5.2, as well as the traverse velocities required for the production of 2.0 mm pin fins. This 2.0 mm fin height was adopted at this stage in the project development, as it was determined to be the final height of the as-sprayed fins, before milling down to 1.7 mm as required by the dimensions of the heat transfer test rig. These specifications had previously been unavailable. The traverse

velocities used were determined by equation (4-1) using an adjusted constant from the actual fin heights achieved in Figure 5.3, and are shown in Table 5.3. From this, it was possible to conduct the cost analysis for the selection of the lowest cost parameters.

Table 5.3: Constant Correlating Traverse Velocity to Fin Height and Traverse Velocity Required for 2.0 mm Fins

Temperature [°C]	Pressure [MPa]	Constant, c_{v-H} [mm ² /s]	Predicted Required Traverse Velocity [mm/s]
200	2.07	11.9364	6.0
200	2.76	23.0538	11.5
200	3.45	27.6507	13.8
300	2.07	25.7828	12.9
300	2.76	38.0182	19.0
300	3.45	42.316	21.2
400	2.07	37.2674	18.6
400	2.76	49.6105	24.8
400	3.45	56.6143	28.3

Knowing the traverse velocities necessary to produce 2.0 mm height fins for each set of spray parameters, it was possible to determine the cost of fin production for a given set of conditions. This cost analysis took into account the cost of powder, gas, labour, spray masks, and electricity, all of which were dependent on the spray time (traverse velocity) and DE of each set of spray parameters, as previously described in experimental procedures, Section 4.8. In Figure 5.6, the results of the cost analysis are shown, with all resultant costs having been normalized on a unit scale with respect to the highest cost set of parameters by dividing the total cost of each parameter set by the total cost of the most expensive trial, 200°C-2.07MPa. This shows that there is a general downward trend in the cost of spray with increasing gas stagnation temperature and pressure. Of the various spray parameters considered, it was revealed that the lowest pressure/temperature

combination was most expensive, while the highest temperature/pressure was the cheapest.

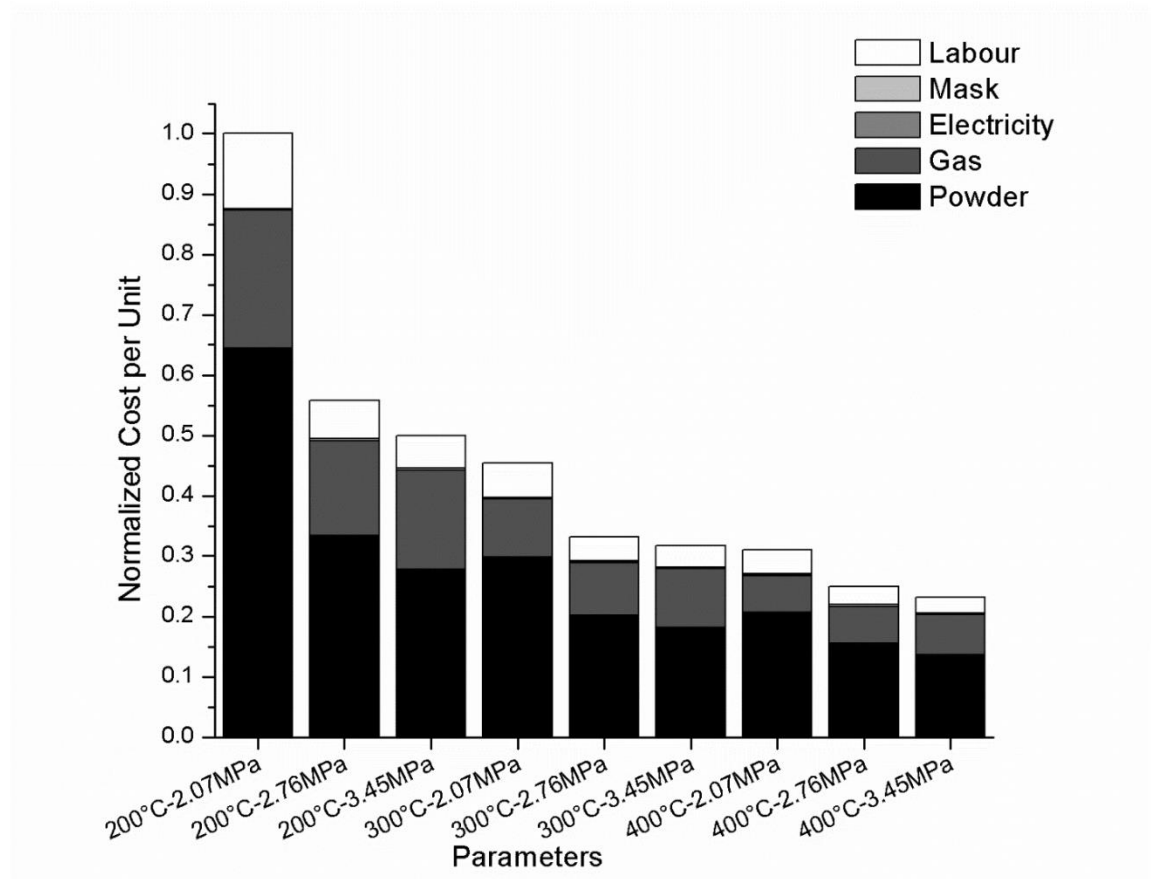


Figure 5.6: Normalized cost associated with producing pin fin array unit at various gas temperatures and pressures.

The factor that had the most drastic impact on the spraying cost of the pin fin array units was powder cost, which is directly correlated to DE. The DE increases with gas temperature, as this increases the carrier gas velocity and therefore the maximum velocity achievable by the particles. Increasing gas pressure also increases DE since higher pressure yields higher gas density, increasing the drag force on the particles in the stream, which allows for greater momentum transfer between the gas and the particles prior to impact. The lower the DE, the more powder was necessary to achieve the desired fin

height. Consequently, spraying with reduced parameters took much longer, compounding the gas and labour costs along with the powder cost. The cost of the mask and electricity were mostly negligible. Thus, a gas temperature of 400°C and pressure of 3.45 MPa was identified as the most cost-effective set of spray parameters for the production of 2.0 mm high copper pin fins. As such, these spray conditions were selected as the preferred parameters for the production of fins and coatings for the remainder of this study.

Another important factor to note is that, for the purpose of the cost analysis, the highest powder feed rate possible must be used. In so doing, the time necessary to spray the pin fin arrays for each set of parameters is reduced to its minimum, therefore limiting the costs of the time dependent parameters such as gas, electricity, and labour. Thus, for this study, the powder feed rate was set at the highest value for which the equipment could consistently feed powder, per common lab practices, without clogging the feeder lines or injection port.

Knowing the equipment limitations and the results of the cost analysis, the final list of the parameters used for the production of full sized pin fin arrays and coatings with powder reclamation are provided in Table 5.4. Particle reclamation was carried out from spray runs to produce these full sized pin fin arrays.

Table 5.4: Lowest Cost CGDS Parameters.

Parameter	Value – 2.0 mm Fins
Temperature	400°C
Pressure	3.45 MPa
Traverse Speed	28.3 mm/s
Step Size	1 mm
Passes	1
Powder Feed Rate	30 g/min
Feeder Gas Flow Rate	0.85 m ³ /h
Powder Feeder and propellant Gas	Nitrogen
Standoff Distance	10 mm

5.1.4 Substrate Bending

An issue that was encountered early in the experimental procedures was substrate bending. It was found that bending was caused by both grit blasting the surface of the copper plates, and by the cold spray process itself. This is due to residual stresses from the impacting particles, which causes the substrates to bow outwards. This issue was deemed to be of great importance for the purpose of the heat transfer tests, as even slight substrate bending could have an important effect on the heat transfer ability from the GPU unit to be cooled. To counter the effect of substrate bending, grit blasting was done on both sides of the substrate in an attempt to minimize bending: Side A represents the surface where the pin fins were to be deposited, and Side B represents the backside of the substrate. The goal of was to reduce bending to as close to zero as possible, minimizing any effect bending may have.

Four different samples (S1, S2, S3, S4) were tested for deflection, all using copper baseplate substrates with a 1/8” thickness (1/16” substrates were attempted but deflected too severely). The samples were grit blasted and sprayed in different configurations to compare the resultant bending. S1 was grit blasted on the full surface of Side A only,

while S2 was grit blasted on the full surface of both sides in an attempt to counter the bending of the substrate. S3 and S4 were grit blasted using a mask in order to restrict grit blasting to the finned area footprint, thereby reducing the total area of grit blasting. S3 was grit blasted on side A and S4 on both sides A and B. The pin fin spraying process was also expected to induce residual stresses that might cause bending of the substrate. As such, all samples were then subjected to a CGDS spray trial in order to produce pin fins. This was done at 400°C - 3.45MPa for all sample, as these are the highest parameters used in this work. Important to note is that for S2 and S4 the pin fin arrays were sprayed with the intent of using the bending inherently caused by the CDGS process to help further correct the bending from the grit blasting. Table 5.5 shows the bending of the substrates through the various preparation steps, expressing the maximum height of deflection of the plate at its center compared to the edges.

Table 5.5 Substrate Bending from Surface Preparation and Spray Process

CGDS Parameters	<u>Maximum deflection (μm)</u>			
	As-Received	Grit blast Side A	Grit blast Side B	After CGDS
S1 – Full surface grit blast	25.4	305	NA	356
S2 – Full surface grit blast	~0	254	76.2	50.8
S3– Restricted surface grit blast	25.4	228.6	NA	457.2
S4– Restricted surface grit blast	76.2	381	50.8	152.4

As shown in Table 5.5, the substrates were for the most part already slightly bent in their as-received state. After grit blasting on a single side, the substrates became significantly bent compared to the as-received conditions. After spraying on S1 and S3, the bend was

further aggravated by the residual stresses from CGDS [57]. A marked improvement in the deflection of the plate was achieved when grit blasting was carried out on both sides of the substrate, as opposed to just on one side, as seen in samples S2 and S4. Since S2 and S4 were grit blasted on both sides, it was possible to select the side on which to produce the pin fin array to counter the bending. As such an improvement in the bending was recorded in S2, while there may have been an over correction in the case of S4. Regardless of this, grit blasting of only the finned area in S3 and S4 did not have any benefit in terms of bending when compared to the fully grit blasted samples, S1 and S2. Therefore, for the remainder of this work all substrates used to produce full pin fin arrays for testing in the heat test rig were grit blasted on both sides, with the pin fins being sprayed on the side which would attenuate any pre-existing bending. The final substrate surface roughness was $4.97 \pm 0.82 \mu\text{m}$ as measured using the portable surface roughness gauge.

5.1.5 Spraying of Pin Fin Arrays

In this work, two different fin configurations were tested for heat transfer performance, in addition to a bare plate used as a reference baseline. The fin density was kept constant, at 12 fins per inch in each direction, however the as-sprayed height of fins (before milling down to the requisite height) was changed. As such, the resulting area of the fin cross sectional profile for each of the two sets of fins was different, and therefore the size of the flow channel around the fins (net flow area) was also different, as illustrated in Figure 5.7. Thus, full sized samples were produced with fins at a 2.0 mm and 1.8 mm height then both sets were milled down to the desired 1.7 mm height, as previously mentioned. The 2.0 mm pin fin arrays were produced using the lowest cost spray parameters

previously determined, 400°C and 3.45 MPa, whereas the 1.8 mm pin fin arrays were sprayed at 300°C and 3.45 MPa. These different parameters were chosen for the second set of fins, as lower parameters were found to yield a less vertical base angle, which results in lower flow obstruction. Thus, a lower set of parameters was chosen in order to investigate the effect of flow obstruction on the conductance of the fin array.

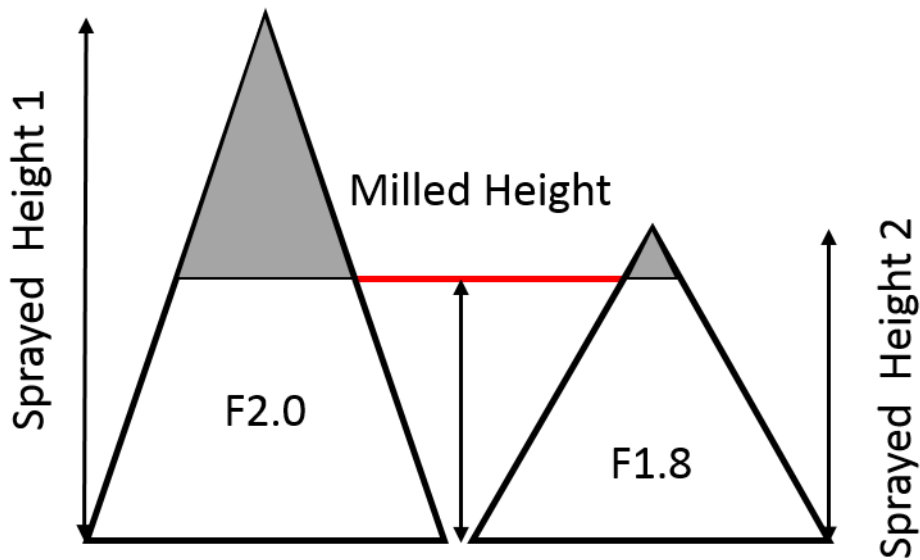


Figure 5.7: Variation in fin cross sectional profile depending on spray height before milling.

Both sets of fins were tested for heat transfer performance before and after annealing. Table 5.6 shows the various configurations used in heat testing, where the first letter F designates the finned samples and NF the non-finned plates. The number indicates the height of the as-sprayed fins, 2.0 mm and 1.8 mm, followed by the designation NA (not annealed) or A (annealed for 4 hours in air at 300°C).

Table 5.6: Sample to be tested in heat test rig

Sample Name	Sprayed Height (mm)	Milled Height (mm)
F2.0-NA	2.0	1.7
F1.8-NA	1.8	1.7
NF-NA	N/A	N/A
F2.0-A	2.0	1.7
F1.8-A	1.8	1.7
NF-A	N/A	N/A

Figure 5.8 shows three-dimensional images of the 2.0 mm and 1.8 mm fins before and after milling down to 1.7 mm. These images show clearly the difference in fin shape before milling. The 2.0 mm fins have a steep angle at the base before tapering to the peak, while the 1.8 mm fins have a shallower base angle. After milling, this results in the expected cross sectional profile seen in Figure 5.7, where the 2.0 mm fins give a larger cross sectional area than the 1.8 mm fins.

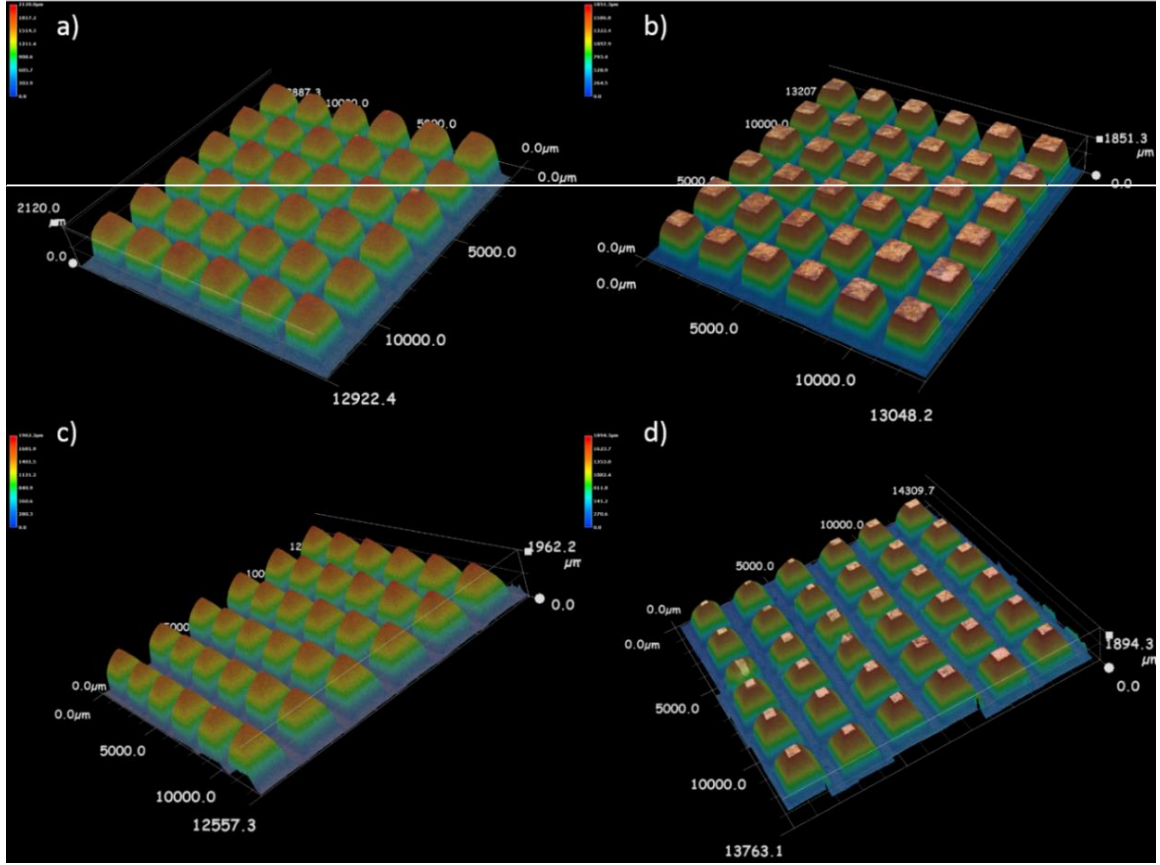


Figure 5.8: Digital microscope 3D images of: a) As-sprayed 2.0 mm pin fins; b) 2.0 mm fins milled to 1.7 mm; c) as-sprayed 1.8mm pin fin; d) 1.8mm fins milled to 1.7 mm.

The different fin geometries were then characterized. The 2.0 mm as-sprayed fins featured a base angle of $\theta=9\pm1^\circ$, base width $B=1496\pm27\mu\text{m}$, and fin spacing (at the base) $S=471\pm31\mu\text{m}$. Conversely, the as-sprayed 1.8 mm fins exhibited a base angle of $\theta=17\pm1^\circ$, base width $B=1506\pm24\mu\text{m}$, and fin spacing $S=461\pm20\mu\text{m}$. One should note that the fin spacing is narrower than the $584\mu\text{m}$ wire width from which it is produced. This can be attributed to the spreading effect of the gas stream between the bottom of the mesh and the substrate, but also to the natural funnel produced by the fins as they grow which would cause deviated particles to impact beneath the wire.

Next, samples of the final fin arrangements were cut and mounted for cross sectional analysis. Examination of the fin cross sections, as shown in Figure 5.9 reveals two distinct structures: a fully dense core with a porous periphery. While the core porosity level is too low to be measured reliably, the porosity of the sides of the fins is found to be approximately 5%. This effect is attributed to the fact that on the edges of the fins, particles experience a lower impact angle, which has been shown to result in increased porosity [101].

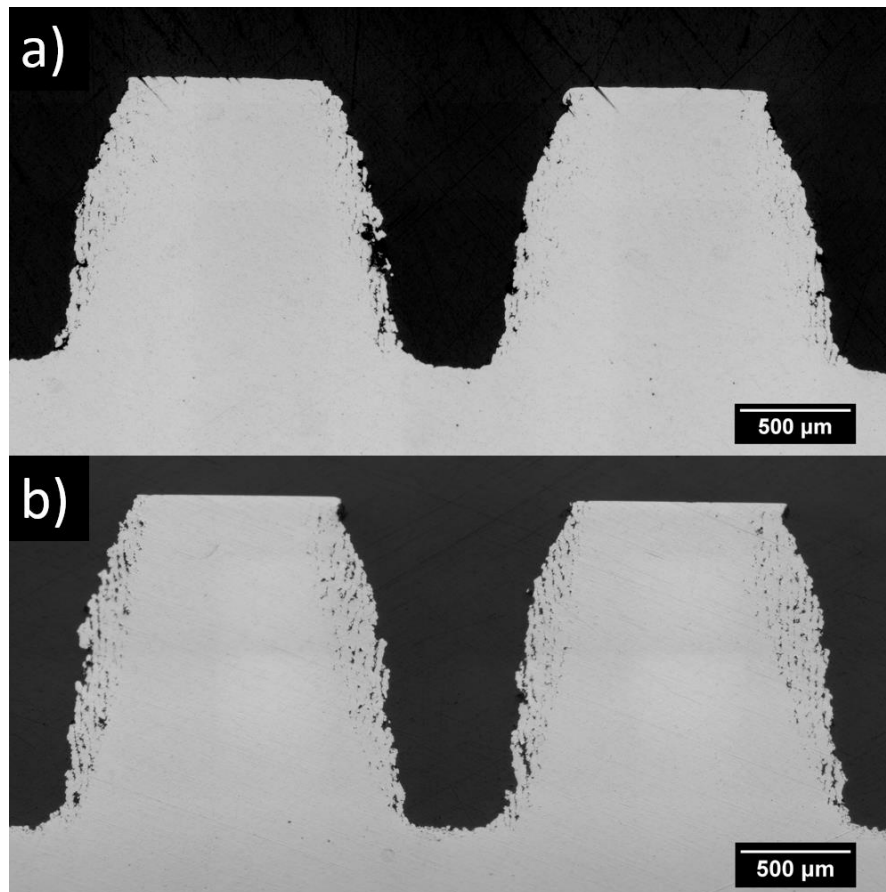


Figure 5.9: Milled pin fin cross section: a) pin fins produced with 300°C-3.45 MPa; b) pin fins produced with 400°C-3.45 MPa.

An illustration of the changing impact angle as a function of fin growth is provided in Figure 5.10a, showing that the taller the fin, the lower the angle of impact becomes.

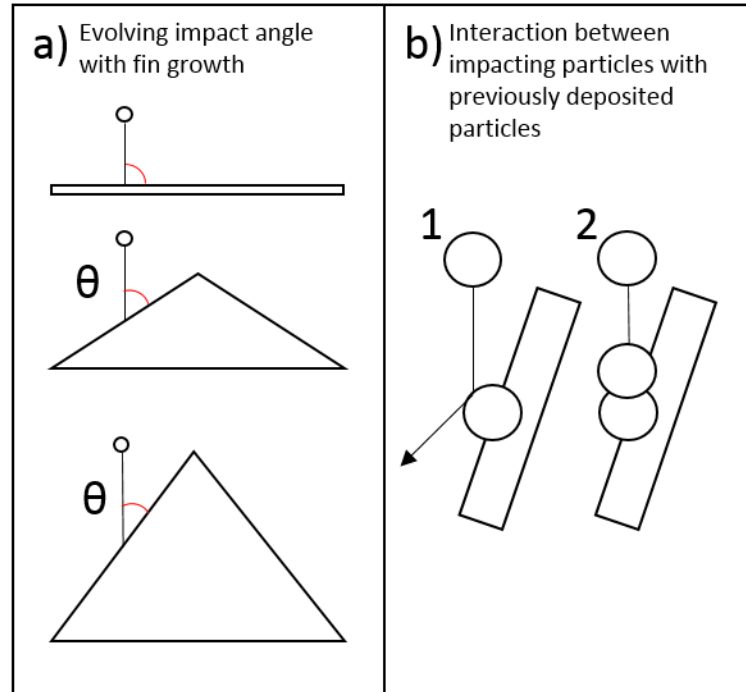


Figure 5.10: Evolving particle impact angle with fin growth with respect to a) the fins, b-1) particle to particle impact where the angle of impact causes the particle to ricochet off of a previously deposited particle, and b-2) particle to particle impact where the angle of impact is high enough to cause the incoming particle to stack on top of a previously deposited particle.

When closely examining the more porous areas of the fins, one can see what appears to be a columnar growth pattern as seen in Figure 5.11. This effect only takes place at extremely low impact angles. When the particles impact at this angle, many of them do not stick and rather ricochet to the side, but those that do adhere are the ones that impacted at the small outcroppings formed by previous particles, as illustrated in Figure 5.10b. This would explain the apparent stacking of particles on the periphery of the fins. The porosity this causes may result in a loss of thermal efficiency of the fin, as the

thermal conductivity of these voids is much lower than that of the surrounding copper. However, given the high thermal conductivity of copper and microscopic size scale of these defects, it is expected that the macroscopic thermal performance of the fin should not be significantly affected.

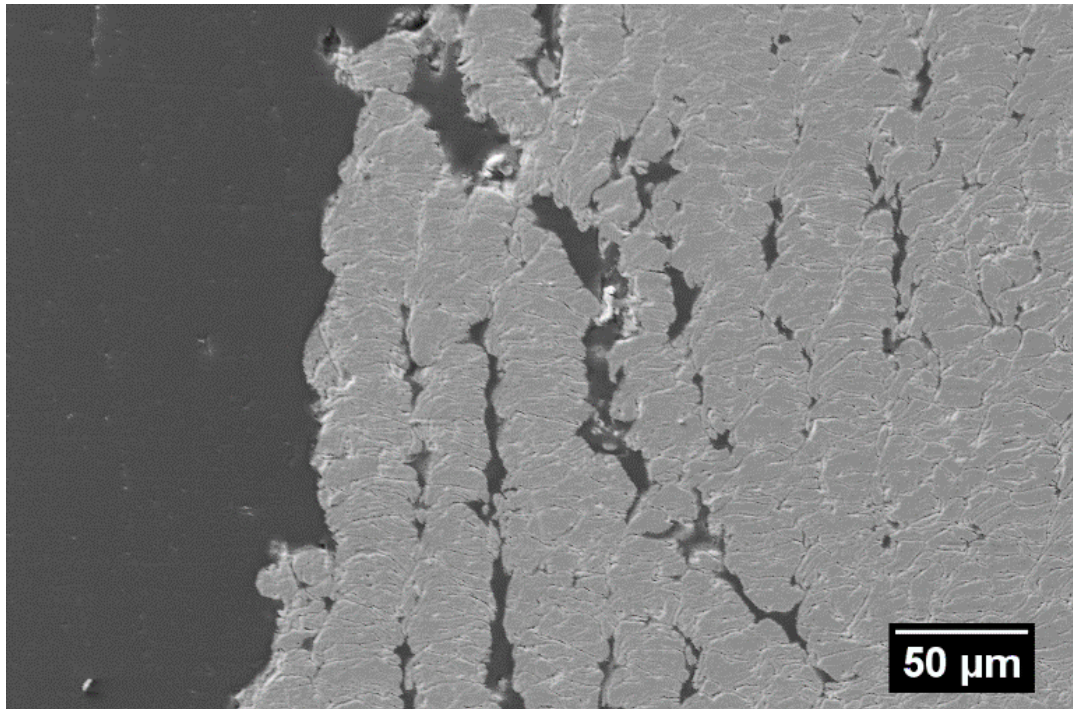


Figure 5.11: Columnar growth on the fin periphery.

5.1.6 Mesh Reuse

A drawback of using a masking technique to produce pin fin arrays is that some of the feedstock powder deposits on the wire-mesh mask, as shown in Figure 5.12.

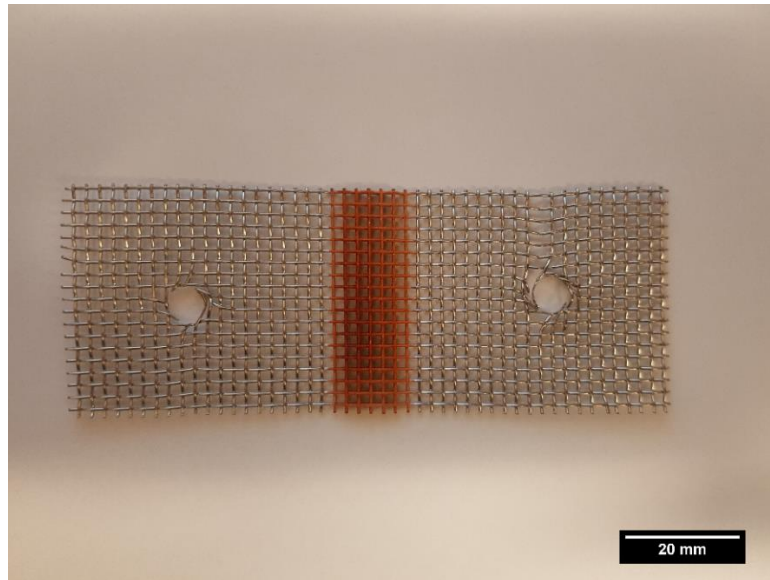


Figure 5.12: Example of copper buildup on wire mesh mask.

This buildup constitutes a substantial loss of feedstock powder and may also render the wire-mesh mask unusable for subsequent sprays. At this stage, it was unknown whether the deposition on the mask would affect the growth of the fins. Therefore, tests were done at 400°C-3.45 MPa to establish whether the mesh was reusable. In these tests, the hope was that the DE on the substrate would not change, while the DE on the mesh would begin to plateau. This would mitigate the loss of powder on the wire-mesh mask, and indicate whether the mask could be reused, resulting in a cost saving for both the feedstock powder and the masking material. The DE results are shown in Figure 5.13.

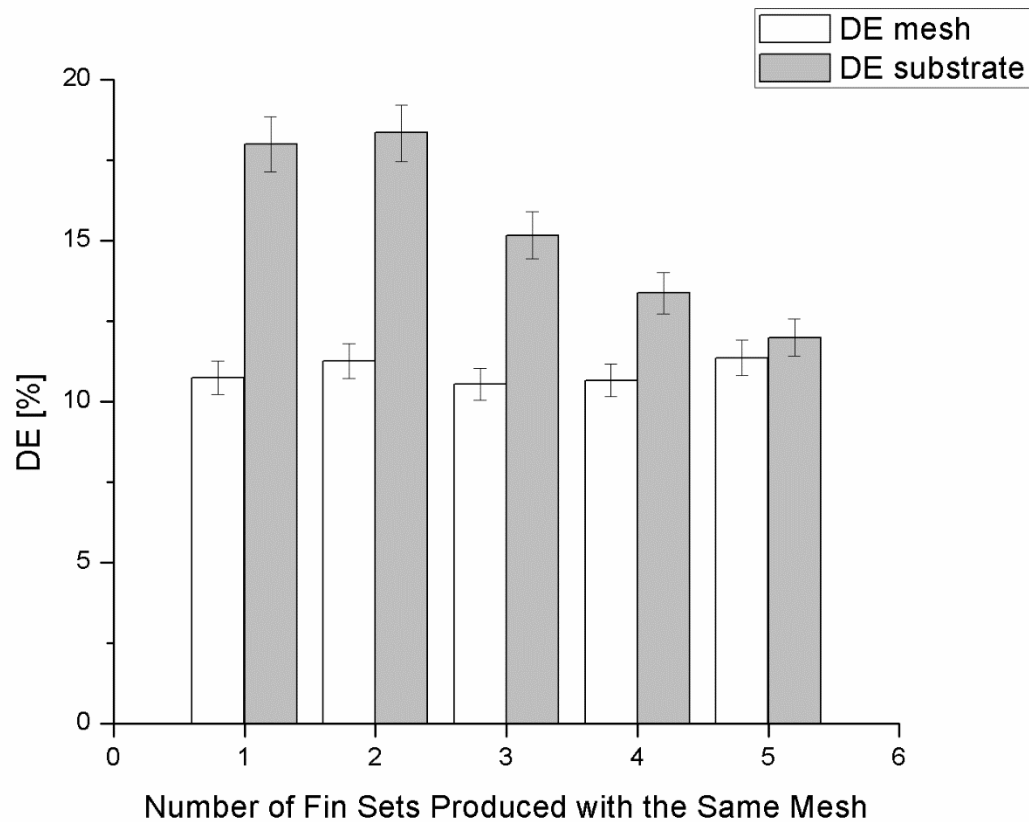


Figure 5.13: The DE on the substrate and mesh when reusing the mesh to spray pin fin arrays.

It was seen that the DE on the mesh remains constant through each reuse, while the DE on each new substrate became progressively lower. This is hypothesized to be due to interference from the increasing height of the buildup on the wire-mesh mask. This could negatively affect the carrier gas flow and incoming particles. From this result it is evident that the mesh cannot be reused for subsequent sprays.

5.2 Powder Recycling

5.2.1 Powder Morphology – As-Received and Reclaimed

Powder reclamation was carried out using the powder reclamation apparatus during spray runs of the full pin fin arrays. Various adjustments were made to the reclamation apparatus throughout testing in an attempt to improve the reclamation efficiency, such as the use of improved filters, and increasing the vacuum suction at the outlet of the reclamation apparatus to attract loose powder towards the filters. This allowed for 38% of all un-deposited powder to be reclaimed for analysis and reuse.

From Figure 5.14a, one can see that the as-received powder is composed mostly of spherical particles (with the exception of a few slightly deformed particles), and that smaller particles cluster around the larger ones as satellites. The image of the reclaimed powder (Figure 5.14b) reveals that most of the particles have retained little trace of their original spherical shape. The particles become deformed during various steps of the spraying process, such as during feeding or injection into the nozzle, but most importantly, they are deformed by impact on the substrate and mask assembly.

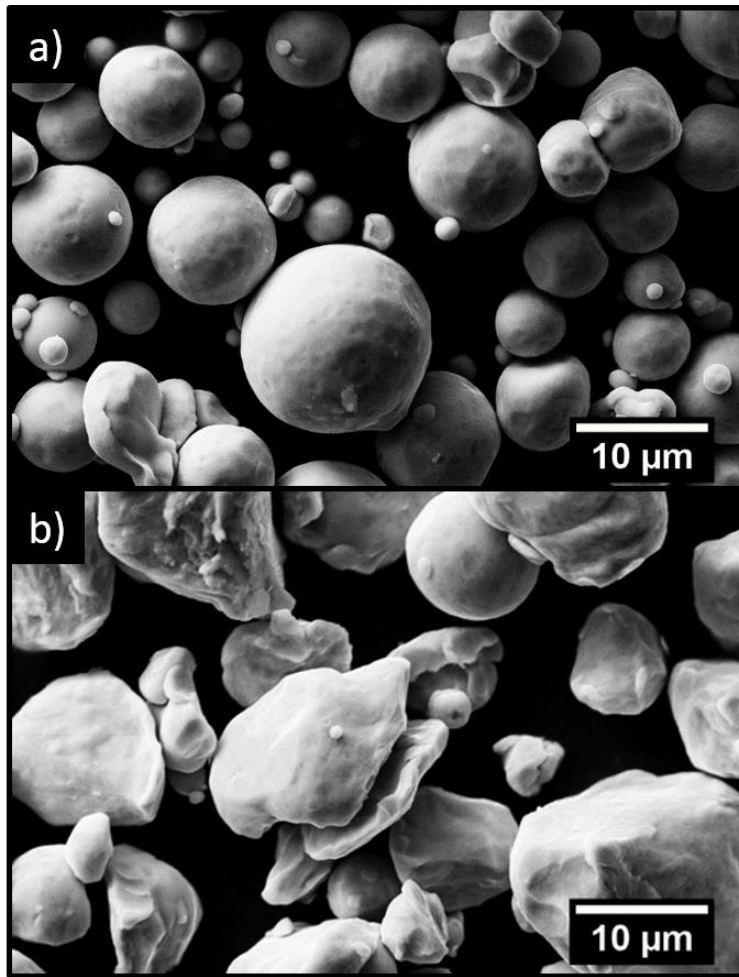


Figure 5.14: a) As-received Cu-159 copper powder; b) Reclaimed Cu-159 copper powder.

The reclaimed powder particles show evidence of fracture and plastic deformation. Some particles are still mostly spherical, which indicates that they have experienced minimal deformation during the initial spray process. Other particles are highly deformed: one can observe completely flattened particles which likely experienced high impact velocity, while others have jagged edges indicating that they fractured under the forces exerted upon them.

By examining the reclaimed particles at higher magnification (Figure 5.15), it is possible to see evidence of ductile rupture by the presence of “cup-and-cone” features. This could indicate that metallurgical bonding took place at one time, but that the particles were then shorn from the substrate or underlying coating by either rebound or subsequent impacts [102].

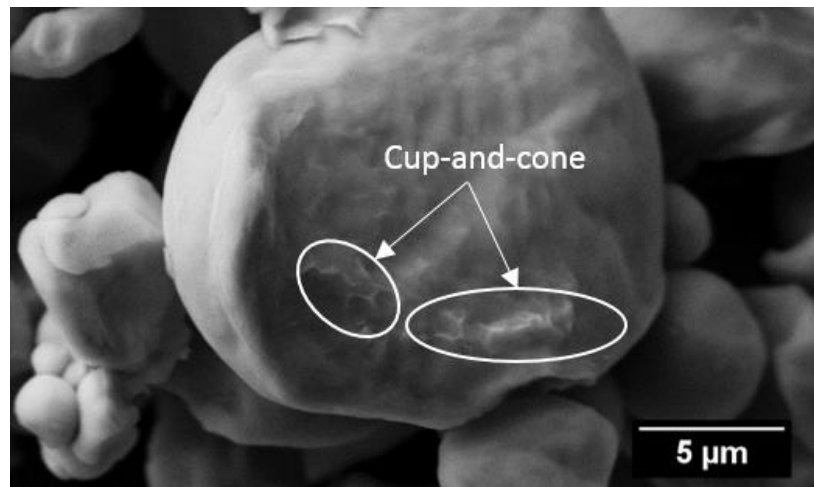


Figure 5.15: Cup-and-cone on reclaimed Cu-159 particle.

5.2.2 Single Particle Impacts

An examination of single particle impacts was done to compare the deposition behavior of the as-received particles to that of the reclaimed particles. These tests were done using the previously determined spray parameters of 400°C and 3.45 MPa. Figure 5.16 shows the result of a single as-received particle impacted on the substrate. One can see the clear spherical shape of the top of the particle, and evidence of jetting on the periphery of the impact zone [103].

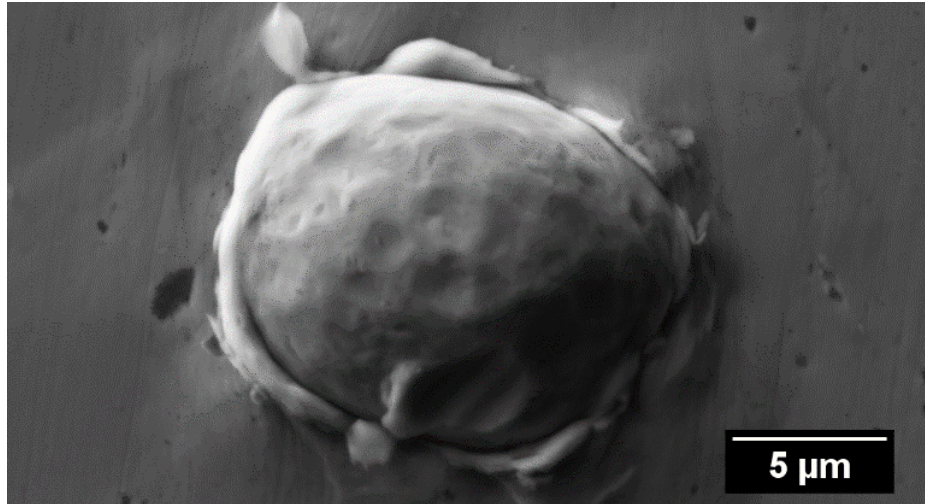


Figure 5.16: Impact of a single as-received particle.

Figure 5.17, which shows a reclaimed powder particle that impacted on the substrate, illustrates that the top of the impacted reclaimed powder is severely deformed and has retained very little of its initial spherical shape. Regardless of the shape of the particle, it seems to have experienced localized jetting along its periphery, similarly to the as-received particle.

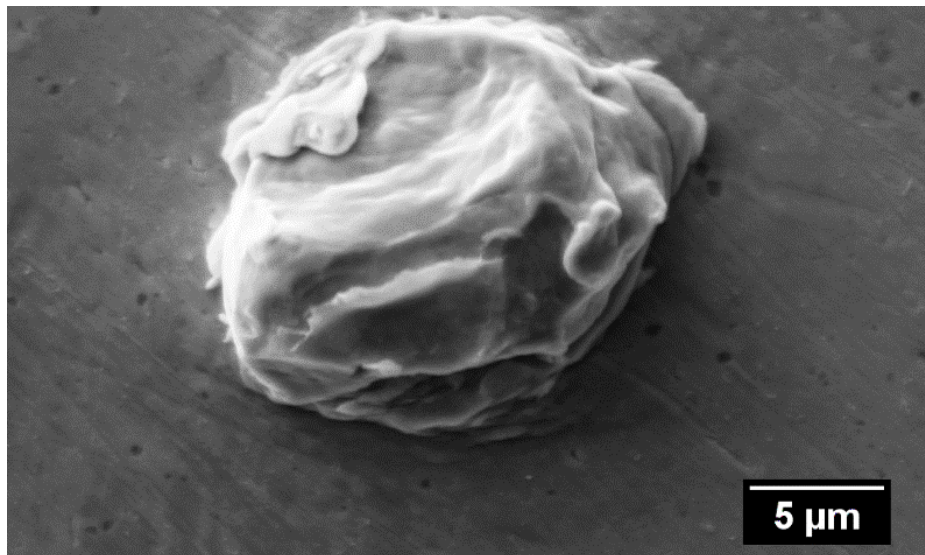


Figure 5.17: Impact of a single reclaimed particle.

5.2.3 Coating Comparison

In order to verify the feasibility of depositing reclaimed powder, various coatings were sprayed using: (1) 100% as-received copper powder, (2) a 50wt% as-received - 50wt% reclaimed powder blend, and (3) 100% reclaimed powder. All three coatings were sprayed using the 400°C-3.45 MPa spray parameters, while maintaining the same powder feeder setting. However, due to the lower flowability of the reclaimed powder, the traverse speed of the gun was varied in order to guarantee that all the feedstock powder was sprayed to impact the substrate for the purpose of DE measurements. The resulting coating porosities and DEs are shown in Table 5.7.

Table 5.7: DE and porosity of as-received & reclaimed powder blends.

Powder Blend	DE (%)	Porosity (%)
100% As-received	68	<0.1
50% As-received – 50% Reclaimed	64	<0.1
100% Reclaimed	46	<0.1

Since the porosity levels of these coatings are comparable, it can be concluded that it is possible to implement powder recycling without any major detriment to the quality of the coatings, and by extension pin fins produced. An important aspect to note is the change in DE between the different powders. The DE for as-received powder was 68% and that of the reclaimed powder is 46%, a considerable reduction. This difference can be attributed to the fact that, in the reclaimed powder, the particles that were the most likely to deposit had already done so in the original spray trial, leaving the generally larger and less likely to adhere particles to be reclaimed. From the as-sprayed etched coating cross-sections in Figure 5.18, it is seen that the average particle size appears somewhat larger in the coating made of 100% reclaimed powder. This is also true in the images of reclaimed

powder, seen previously in Figure 5.14b. Powder granulometry analysis was considered as a means to confirm this observation with the reclaimed powder, however, the irregular shape of the reclaimed powder would make the results untrustworthy.

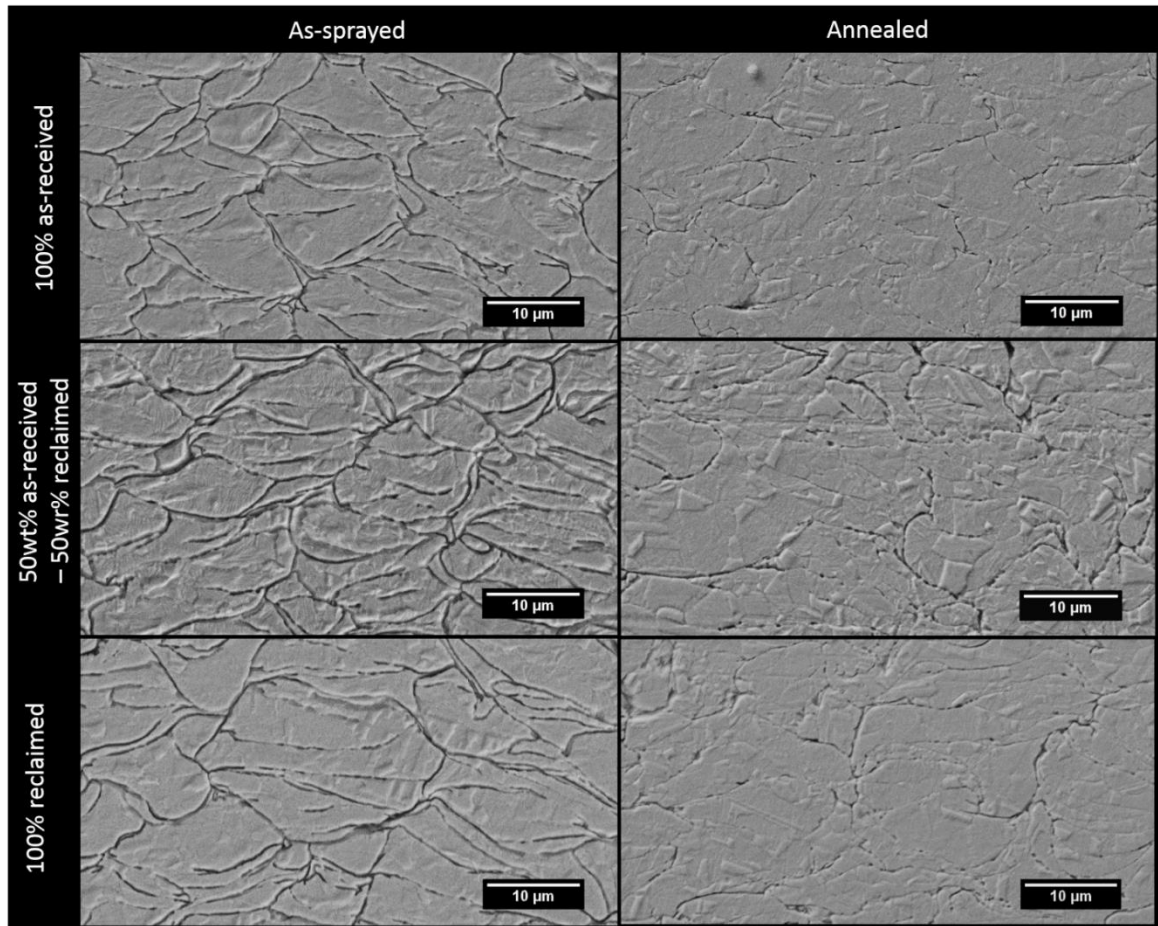


Figure 5.18: Images of (etched) coatings produced using as-received and reclaimed powder blends before and after annealing at 300°C for 4 hours.

Another reason the DE has dropped is due to the fact that the particles have likely been work hardened during their previous impact, and they are now even less likely to bond to the substrate as their critical velocity is potentially larger than in the as-received state [86]. As for the mixed powder, if there were no interaction between the as-received powder and the reclaimed powder, one would expect the DE to be close to the average of

their respective DE's, which would give a value near 57%. However, a DE of 64% was recorded. This can be explained by accounting for particle interactions between the two powders, an effect that has been shown to improve DE in feedstock powders featuring blends of hard particles and soft particles, as seen in the work of Fernandez et al. [94]. This work has shown that the presence of hard particles mixed with an aluminum powder increases the DE of the aluminum. This was determined to be due to both asperity creation and oxide layer removal mechanisms improving the conditions for bonding of aluminum particles. This effect is hypothesized to be present in the mix of reclaimed and as-received particles, where the cold worked, and therefore harder, reclaimed particles act as the hard particles by improving the surface conditions for the bonding of the as-received particles. However, in this case the harder reclaimed particles have the possibility of adhering themselves as well. The lower DE of the reclaimed powder may cause it to become inefficient to reuse for the spraying of subsequent heat sinks, however, in an industrial setting it would probably not be necessary to use 100% reclaimed powder, but rather the reclaimed powder would likely be mixed into the next batch of as-received powder such as a 50-50 mix.

The coatings hardness was also investigated, with the results presented in Table 5.8. There was a noticeable reduction in hardness when reclaimed powder was used as opposed to as-received powder. The reason for this is hypothesized to be due to an in situ annealing effect. The powder feed rate when using reclaimed powder (either 100% reclaimed or 50/50 mix) was lower than for the as-received powder when using the same feeder parameters due to reduced flowability of the reclaimed powder, which stems from the irregular shape of the particles. For this reason, it took longer to spray when using the

same powder feeder setting. As such, the CGDS nozzle was over the substrate for a longer time in the reclaimed and mixed powder sprays, by a factor of 3 and 5, respectively, thus the heated gas stream may have induced in situ thermal softening which would have contributed to the lower hardness values in the reclaimed and mixed coatings. In order to improve the feeding of the reclaimed powder, a feeder wheel with larger holes should be employed, and the rotation speed of the feeder wheel increased, allowing to properly supply powder at the prescribed mass-based feed rate in future sprays.

Table 5.8: Hardness of coatings made with as-received and reclaimed powder.

Powder Blend	Hardness (HV_{0.5g}, no annealing)	Hardness (HV_{0.5g}, with 300°C/4hr annealing)
100% As-received	129.2±7.6	81.8±0.3
50% As-received – 50% Reclaimed	102.1±6.3	84.3±1.6
100% Reclaimed	112.0±1.9	80.7±1.0

Another reason for these lower hardness values is believed to be due to a lowering in the cohesion between particles in the reclaimed and mixed coatings [104]. This could again be due to the previous work hardening of the reclaimed particles, causing them to experience less deformation upon a second impact, and therefore limiting their potential for metallurgical and mechanical bonding [86]. Another explanation for this lowering in cohesive strength is the presence of an oxide layer on the surface of the reclaimed powder. The reclaimed powder has previously been heated through the spray process, and due to this oxide growth may have taken place. The presence of oxides has been shown to lower the DE and cohesive strength of CGDS coatings [105].

Following this, the coatings were annealed at 300°C for 4 hours in air with the intent of improving the cohesion between particles through sintering [106], and in so doing

improve the coatings thermal conductivity by consolidating the particle boundaries [99], [107]. From the annealed coating cross sections in Figure 5.18, it is shown that the intra-particle boundaries have become much less distinct from the annealing process, improving the conditions for conduction between particles as desired. The results shown in Table 5.8 demonstrate that there is much lower variation in the hardness of the coatings after annealing, indicating that the coatings have now become more uniform. The lowering of the overall hardness however is typical of the annealing process, as the work hardening in the coatings that is characteristic of the CGDS process has been relieved [99].

5.2.4 Particle Velocity

The particles velocity was recorded for both the as-received and reclaimed copper particles. The results, shown in Figure 5.19 by box-and-whisker plots, follow a normal distribution that is typically seen in CGDS particle velocity measurements [108]. In this plot, the line in the middle of the box represents the 50% mark of particle velocity distribution, while the upper and lower edges represent 25% and 75%, and the whiskers represent 5% and 95%. The average velocity of the as-received particle was measured to be 538 ± 116 m/s, while the reclaimed particles had an average speed of 487 ± 138 m/s, represented by the small squares in Figure 5.19. The reason for this drop in velocity is likely due to the average particle size of the powders. The physical reclamation process makes it more likely for larger particles to be reclaimed, which artificially skews the particle size distribution. These larger particles are known to be more difficult to accelerate, resulting in a lower average velocity [86]. This reduction in mean velocity may also account for the drop in DE of the reclaimed powder [73].

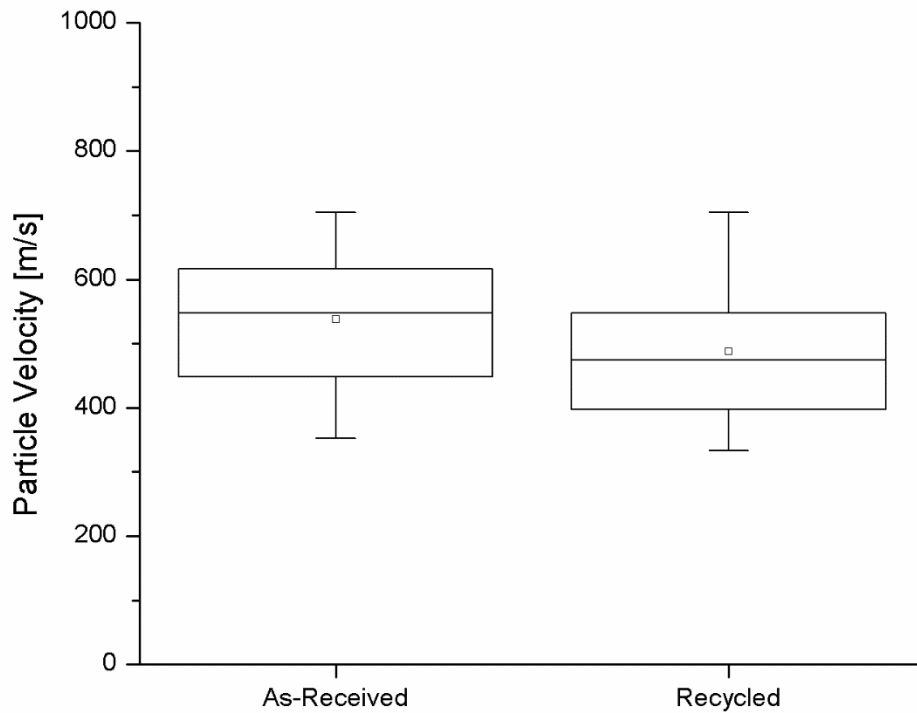


Figure 5.19: Box and whisker plots of the as-received and reclaimed powder velocities.

5.2.5 Cost Analysis for Powder Recycling

With powder recycling proving to be possible without drastically changing the microstructural properties of the coatings produced, the economic model was modified to include powder recycling. Figure 5.20 shows the resulting costs for each set of spray parameters, assuming that all the powder that does not bond to the substrate or mask is reclaimed. For ease of comparison, these values have again been normalized on a unit basis with respect to the cost of the most expensive set of spray parameters previously identified as 200°C-2.07MPa without powder recycling. To allow for direct comparison, the total costs without powder reclamation have been added as black dots on the graph for each set of spray parameters. From this, it is shown that the powder cost no longer dominates the total expense of the pin fin array manufacturing, but rather the gas and

labour costs become more important, each of which are directly proportional to the duration of the spray run. A reduction in spray parameters is associated to a reduction in DE, which means that the spray run must be carried out longer to achieve the same desired pin fin height, which explains the trend of increasing costs at lower spray parameters.

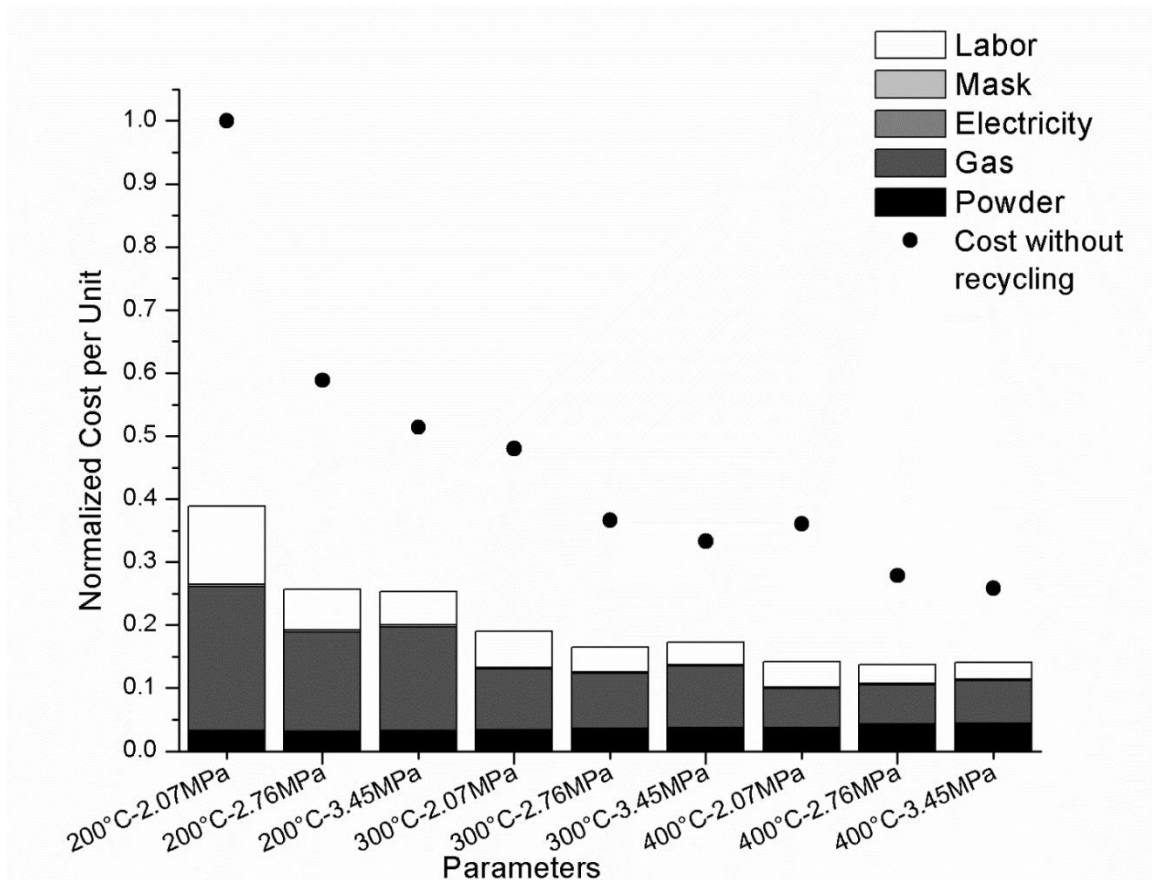


Figure 5.20: Normalized cost associated with producing pin fin arrays with powder recycling at various gas temperatures and pressures compared to cost without powder recycling.

It can also be noted that the powder cost increases at higher spray parameters, which was not the case when there was no powder recycling, as seen previously in Figure 5.6. In order to produce the same fin height, each set of spray parameters must deposit the same

amount of powder on the substrate. Therefore, the variation in total cost in powder for each set of spray parameters without reclamation is reduced to the cost of un-deposited powder and that of the powder adhering to the mask. With powder recycling, the cost of un-deposited powder is eliminated, but not that of the powder on the mask. As previously stated, DE increases with higher temperatures and pressures, but this also leads to a higher DE on the mask. Consequently, at higher parameters, more powder is lost through adhesion to the mask, which accounts for the relatively higher powder costs at high parameters when powder recycling is introduced.

This effect is shown by comparing the ratio of DE on the substrate to DE on the mask, as seen in Figure 5.21. This shows that at higher parameters, the ratio of DEs increases, accounting for the higher relative loss of powder on the mask. In fact, the 400°C-3.45 MPa parameter set is no longer the least costly set when introducing powder recycling, but is slightly more costly than the 400°C-2.76 MPa spray due to this effect. Extrapolating from this result, it is probable that further increasing spray parameters would become less and less cost effective, if powder recycling is implemented. More powder would be lost through adhesion to the mask, and less would be available to reclaim, indicating that there exists an optimal set of parameter where there is a balance between spray time and powder loss.

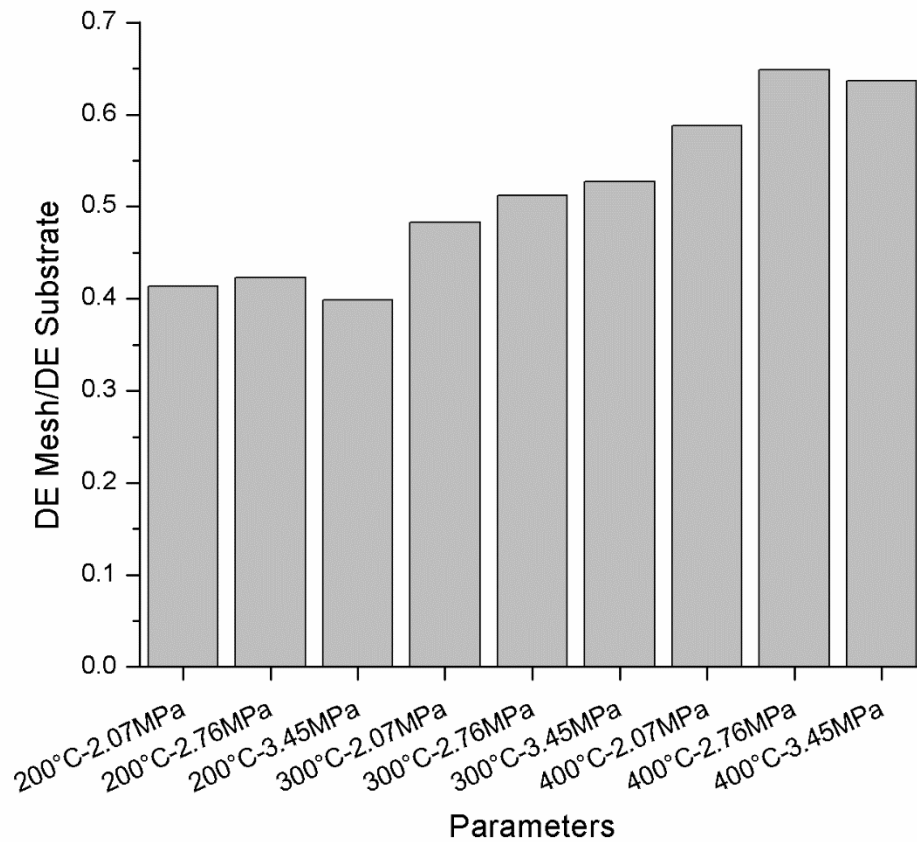


Figure 5.21: Ratio of DE on the mesh to DE on the substrate for production of pin fin arrays

The extra costs in powder are however outweighed by the additional savings obtained from the reduced spray time at higher spray parameters, for the temperatures and pressures tested. Ultimately, the overall cost is reduced by the utilization of powder recycling, with the 400°C-3.45 MPa pin fins being produced at a normalized cost of 0.156 per unit, down from 0.257 per unit, amounting to a 39.3% reduction in production cost if all un-deposited powder could be reclaimed and reused. It is important to note that for this conclusion to hold, subsequent sprays would have to be done with a blend of reclaimed and as-received powder that would result in a minimal DE loss when compared

to the pure as-received sprays. Also important to note is that this assumes that the reclaimed powder could be used multiple times subsequently, and further analysis would need to be done to determine an adequate estimate of reusability. On the other hand, in order to further reduce the powder cost, an important avenue to explore in the future would be the reduction or elimination of mesh buildup. In order to do this, it would be important to limit the ability of feedstock material to build an initial layer on the mesh, possibly by using high hardness metals or ceramics.

5.3 Heat Transfer Performance

The full pin fin arrays, F2.0 and F1.8, were tested for their heat transfer performance, and compared to the non-finned plate, NF. As previously described, it was possible to set the amount of heat input to the copper heater block by a transformer connected to heater cartridges, and it was also possible to control and monitor the water flow rate over the plates using a valve and a flowmeter in conjunction with a constant power pump. The heat test rig could then output inlet and outlet water temperatures, heater block temperatures, and pressure drop across the finned area.

For this analysis, the transformer was set to 90V, as this exhibited high temperature differences between the heater block and the flow of water, without demonstrating any noticeable heat leakage to the environment. To confirm that there was no heat loss from one test to the next, the first thing to verify was that the same amount of heat was being transferred from the heater block into the water for all the plate configurations. According to the first law of thermodynamics, for a given heat input and mass flowrate of water, the difference in temperature between the water inlet and outlet must be the same, regardless of which plate or finned surface is used. This is demonstrated in Figure 5.22 where it is

shown that for the 90V transformer setting, the water temperature difference is the same for a given mass flowrate, regardless of the plate used. Important to note is that the slightest variation in mass flow rate for the lower settings resulted in a large difference in temperature, which accounts for the seemingly large deviations for the left-most data points in Figure 5.22.

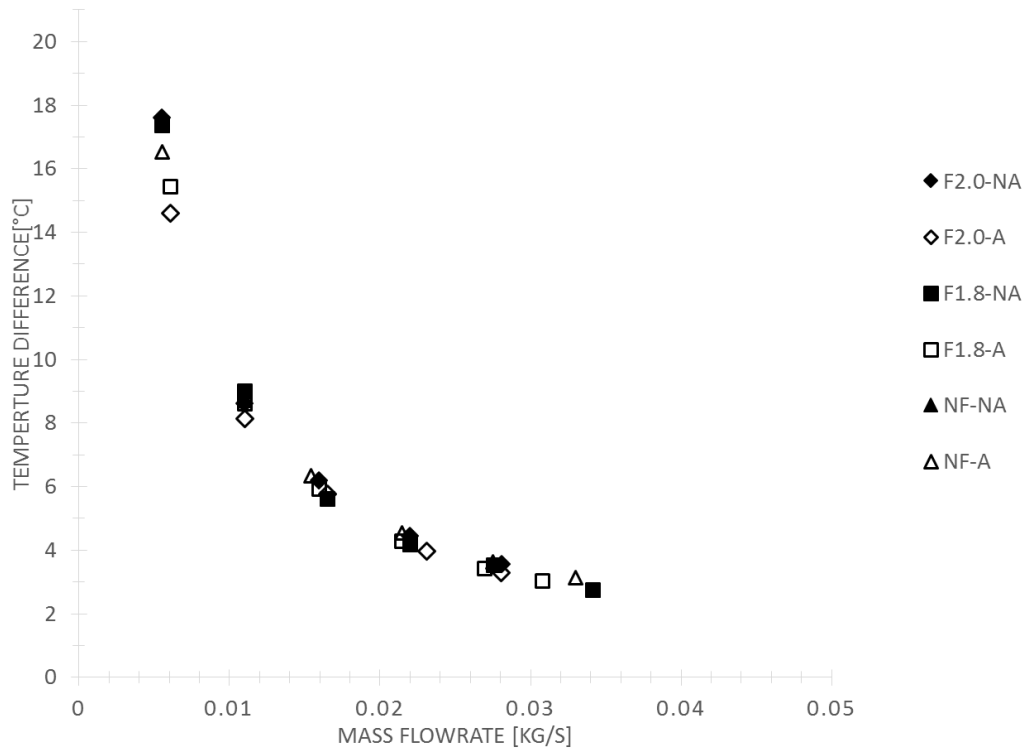


Figure 5.22: Water temperature difference for various water flowrates for various configurations denoted -NA and -A (not annealed and annealed, respectively), of the finned plates F2.0 and F1.8, as well as the bare plate NF.

From the temperature difference, the heat transfer rate into the water can be calculated. This result is shown in Figure 5.23, and reveals that the heat transfer rate into the water matches for every plate configuration, further indicating that there are minimal heat losses to the environment.

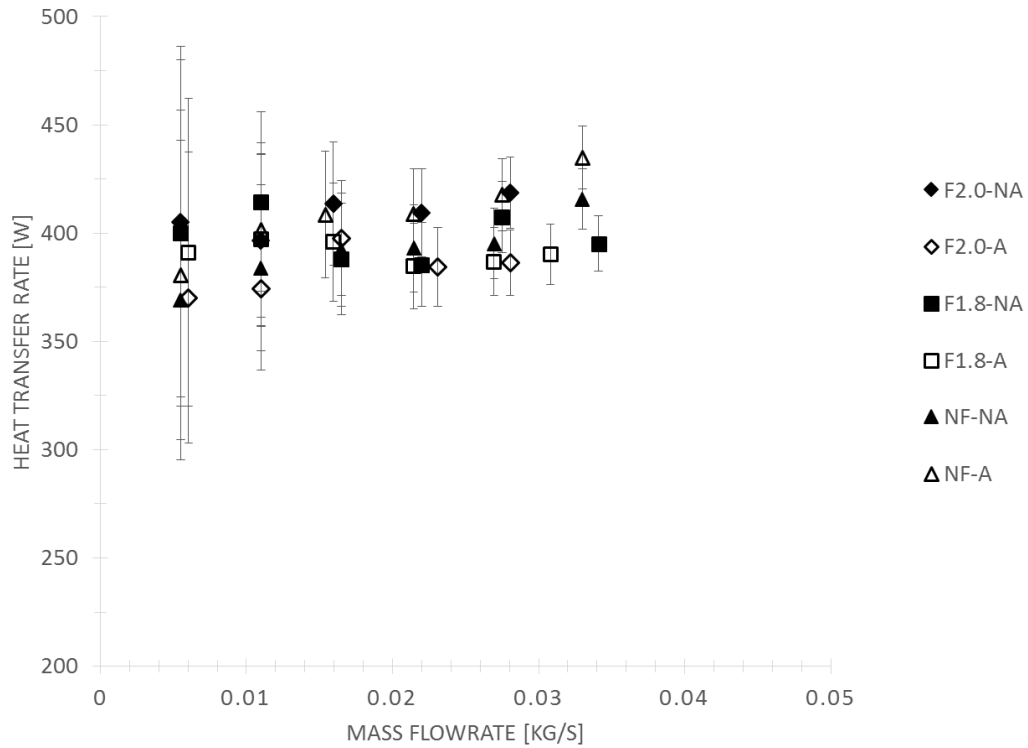


Figure 5.23: Heat transfer rate into the flow of water.

These values were then compared to the heat transfer rate by conduction that was calculated within the heater block itself using the thermocouples therein. Figure 5.24 shows one such example for the F2.0-NA plate, demonstrating that there is slight difference when calculating the heat flux with either method. The difference between the two readings can be accounted for when taking into consideration the fact that the thermocouples set within the heater block may not be reading the temperature exactly at the center of the orifices within which they sit, meaning that the distance between the two readings could be slightly off, resulting in a misrepresentation of the heat flux. Furthermore, these orifices were also filled with thermal past to improve contact between

the block and the thermocouples, but this added resistance which lowers the read temperatures and results in a lower calculated heat transfer rate.

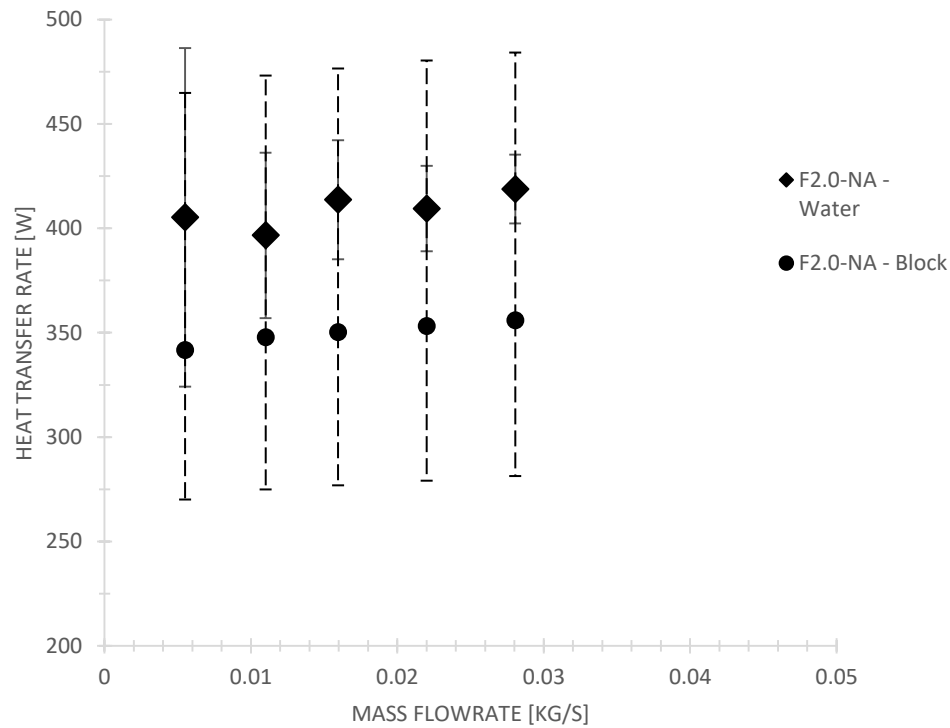


Figure 5.24: Heat flux into water compared to heat flux through heater block.

The pressure drop across every plate configuration was also measured, and is shown in Figure 5.25. This shows that the pressure drop is slightly higher for the F2.0 plate when compared to the F1.8, which was expected since the flow area between the F2.0 fins is smaller than that of the F1.8. This results in a higher level of flow obstruction, leading to slightly higher flow velocities for a given flowrate, which results in higher pressure losses. After annealing there was no large difference between the finned plates' pressure drop. However the slight difference experienced is attributed to the cleaning procedure post-annealing. The plates had a thick oxide layer after annealing, and were therefore put

in a bath of acetic acid. This process most likely resulted in a smoother surface finish than the as sprayed plates, yielding a slightly lower pressure drop. It can also be seen that the pressure drop across the non-finned plate, NF, is practically negligible. This is due to the lack of flow obstruction in the non-finned plates, leading to lower flow velocities and lower pressure losses.

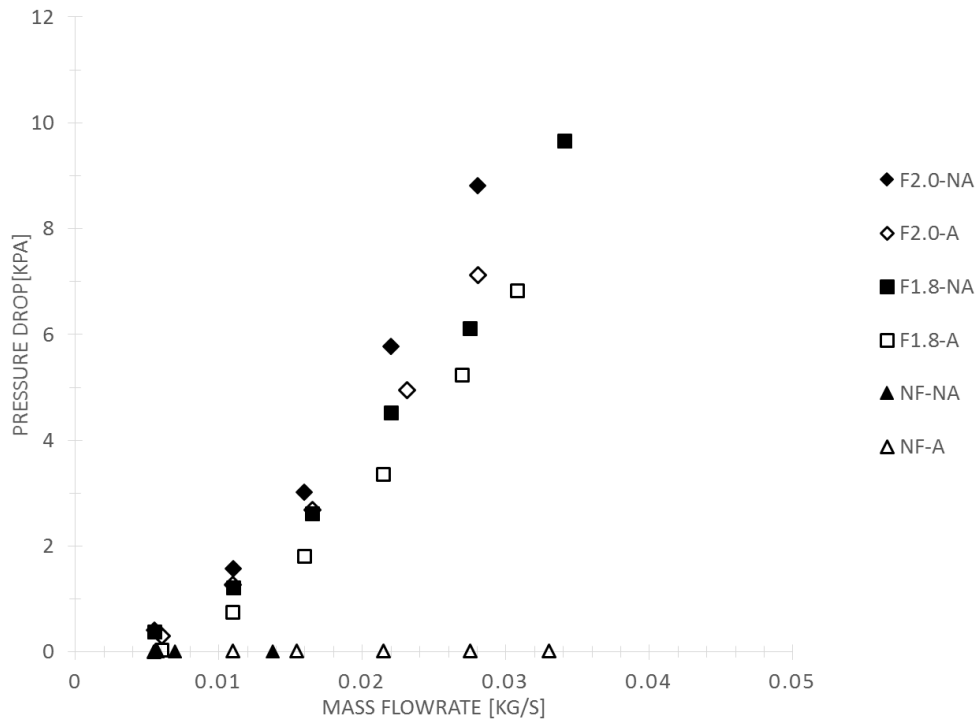


Figure 5.25: Pressure drop across the finned area for each plate configuration.

Before proceeding to the calculations of thermal performance, it was first important to determine which surface heat transfer mode was best suited to the current situation. As previously discussed in experimental procedures Section 4.9.3, the surface can either provide a constant heat flux to the flow of water, or have a constant temperature. In the case of a constant heat flux, the temperature difference from the inlet to the outlet of the

water would be similar to that of the heated surface, whereas for the other case the surface temperature would remain constant. In Figure 5.26 the inlet to outlet temperature differences is shown for both the water and heated surface. These results show that, while the water temperature exhibits a large difference from inlet to outlet, the surface temperature varies little, demonstrating that the approximation of constant surface temperature is better suited to this situation. This is most likely due to the contact region between the water and the surface remaining in the entrance region state, and therefore never becoming fully developed. Therefore, the approximation of a constant surface temperature heat transfer mode was used for the remainder of this work, an assumption that will be verified with CFD modelling further in this chapter.

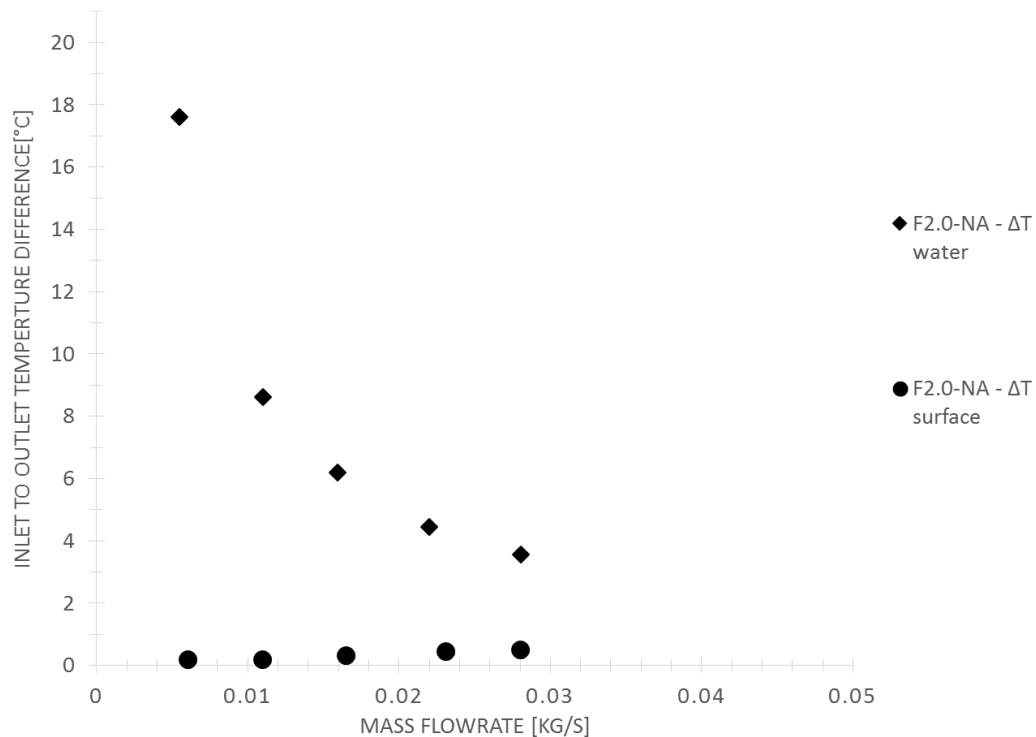


Figure 5.26: Inlet to outlet temperature difference of the water and the heated surface.

The thermal performance of the pin fin arrays was calculated from the data acquired through the heat test rig, and the MatLab code used for data processing can be found in Appendix A. The resulting conductance values for every fin configuration are shown in Figure 5.27. From this it is seen that the finned plates, F2.0 and F1.8, largely outperform the non-finned plates, NF, as expected. All plates showed a rise in conductance with increasing Reynolds number, most likely due to increased convection coefficient from higher levels of turbulence and mixing [17], [18]. The annealing process improved the conductance of both finned plates, as F2.0-A and F1.8-A have a higher conductance than their non-annealed counterparts F2.0-NA and F1.8-NA. This is believed to be due to the particle sintering effect engendered by the annealing process, which enhances the thermal conductivity of the particles boundaries, raising the overall fin efficiency. In both the annealed and non-annealed cases, F2.0 outperformed F1.8. This is most likely due to the higher flow obstruction in F2.0 causing higher flow velocities around the fins, resulting in a higher convection coefficient and therefore more efficient heat transfer. Another explanation is that there is a 17% convective surface area increase from F1.8 to F2.0, a factor which also contributes to a higher conductance value.

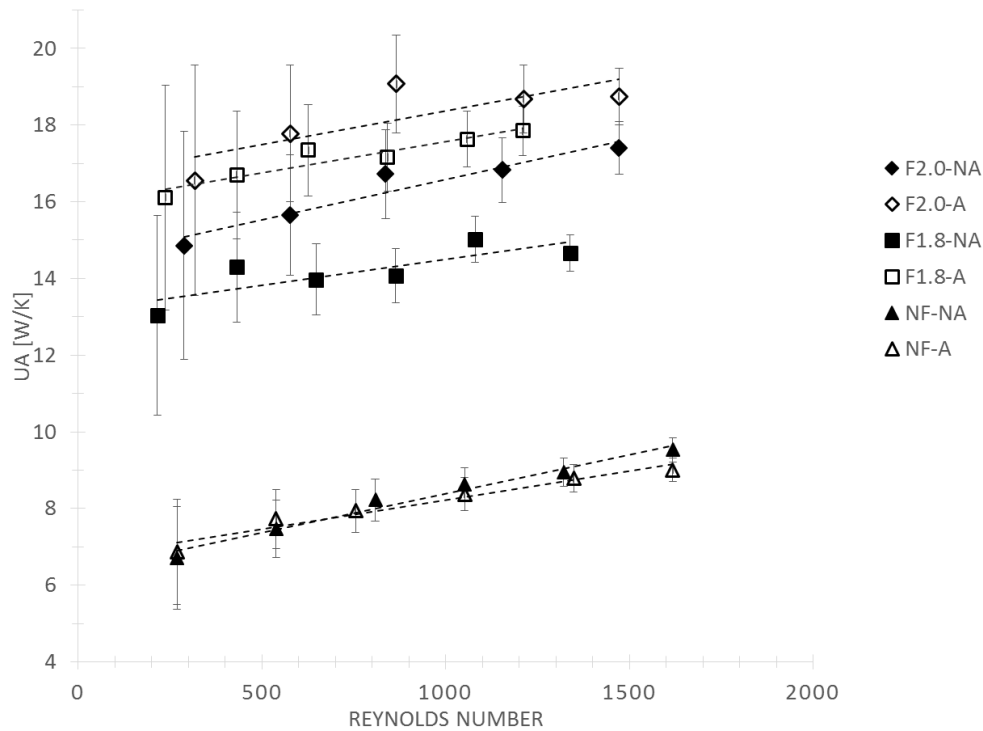


Figure 5.27: Thermal conductance before and after annealing.

In order to compare which fins transfer heat more efficiently at a certain pumping power, the thermal conductance is compared with the effective pump power (on a log scale for ease of visualization) required to flow water through the fin array, as seen in Figure 5.28. Through this representation, it is possible to find the ideal fin configuration, which is one that yields high conductance for low effective pumping power. These curves demonstrate a sharp increase in conductance at low pumping power, while at high pumping power the increase in conductance begins to level off. This shows that there may be little benefit in raising the pumping power for any fin configuration. However this is dependent on the constraints of the pump being used and the desired fin conductance. These results also demonstrate that despite the higher pressure loss, the F2.0 fins were more efficient in heat

transfer than the F1.8. This is most likely due to better conditions for fluid mixing with the higher flow obstruction of the F2.0 fins, resulting in a higher convection coefficient and more efficient heat transfer.

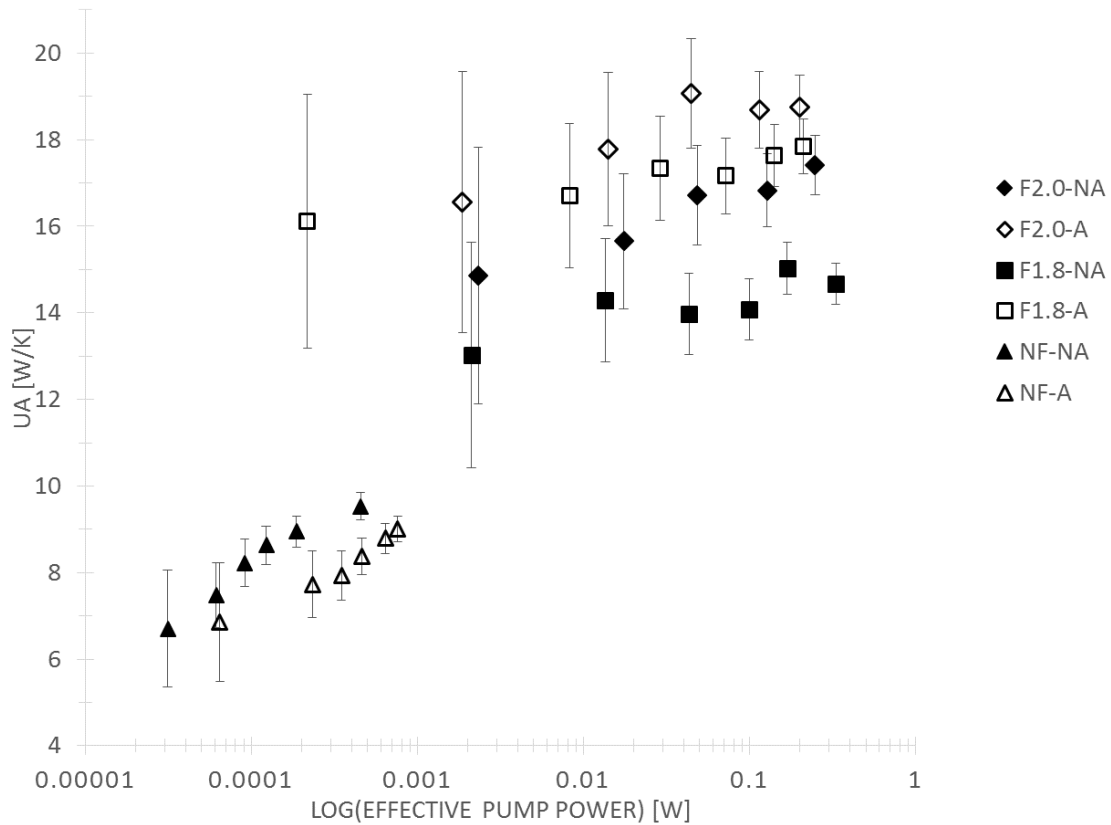


Figure 5.28: Thermal conductance compared to effective pump power.

5.4 Verification of the Constant Surface Temperature

Assumption using CFD Modelling

A CFD model was developed and used with the main goal of determining the validity of using a constant surface temperature assumption in the calculation of the fin conductance (Section 4.9.3). The test case that was investigated was for an inlet water velocity of 0.05

m/s, resulting in a Reynolds number of 292 at the fins, and temperature of 14.7°C (287.85K) with a heat flux of 129898 W/m². These conditions match the lowest flow rate case for the 90V heater settings. The first aspect that was investigated was the difference in water temperature between the inlet and outlet. The temperature distribution in the water is shown in Figure 5.29. It can be observed that the difference in water temperature in the model matched the experimental value. The average outlet water temperature across the exit surface in the model was 305.5K, compared to 305.4K recorded experimentally, constituting a negligible difference. This close result is expected since the inputted value for heat transfer at the base of the model was taken directly from the experimental results.

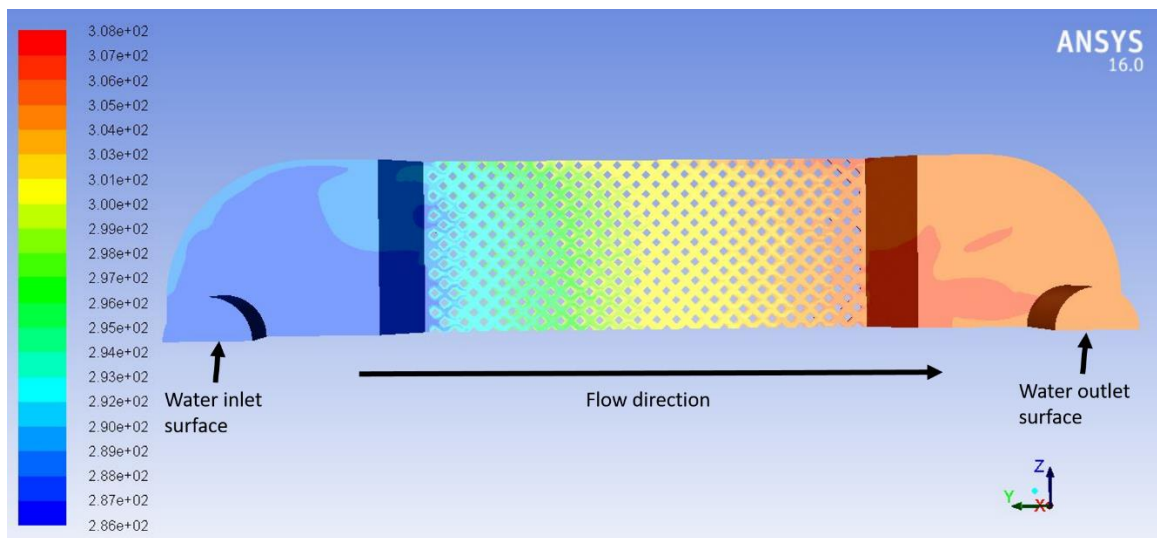


Figure 5.29: Temperature distribution in the water flow.

Having demonstrated that the model matches the experimental set up, it is then used to investigate the validity of an important assumption made in the experimental procedures. The assumption was concerning the nature of the temperature distribution of the

baseplate in the cross flow of water. In order to quantify the fin performance, it was assumed that the baseplate top surface had a constant temperature. This was thought to be a more fitting assumption than a constant heat flux across the plate, based on the initial experimental results. The modelled temperature distribution of the pin fin array and baseplate is found in Figure 5.30.

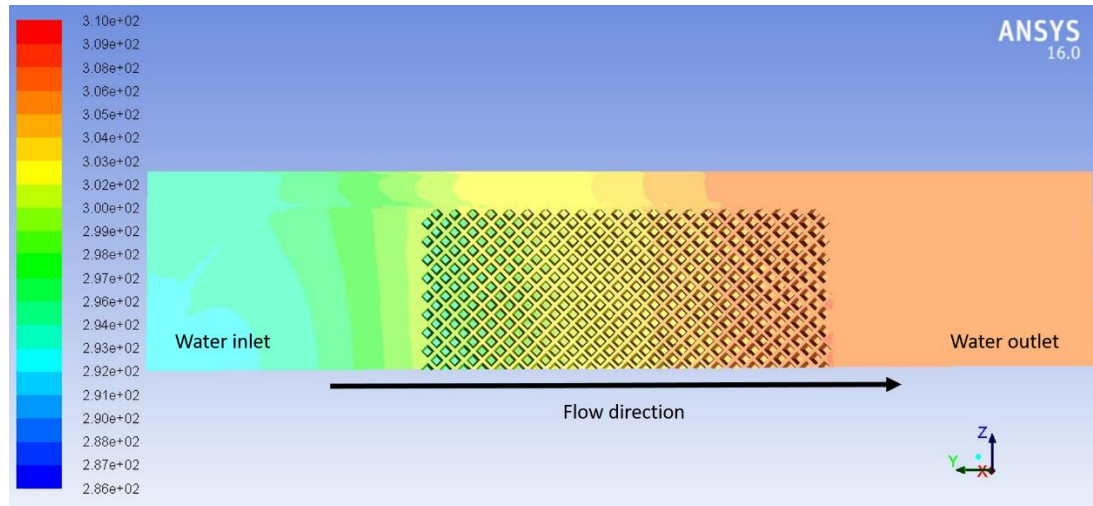


Figure 5.30: Temperature distribution in the pin fin array heat sink.

The modelled temperature rise of the baseplate top surface from the leading edge of the finned area to its exit is 7 K, from 300.5K to 307.5K, which exceeds the experimentally determined temperature rise which was 0.2K. This error can be explained by taking into account the location at which the experimental block temperature was read for interpolation of the surface temperature (see Figure 5.31). It can be observed that the temperature difference across the block is smaller and closer to the heat sources and begins to deviate as it approaches the heat sink baseplate. It follows that by using the one-dimensional conduction approach to interpolate for the surface temperature, this will result in a smaller temperature difference compared to reality. The assumption of one-

dimensional conduction was made in order to be able to determine the surface temperature. However, in reality, the regime in the block is three-dimensional.

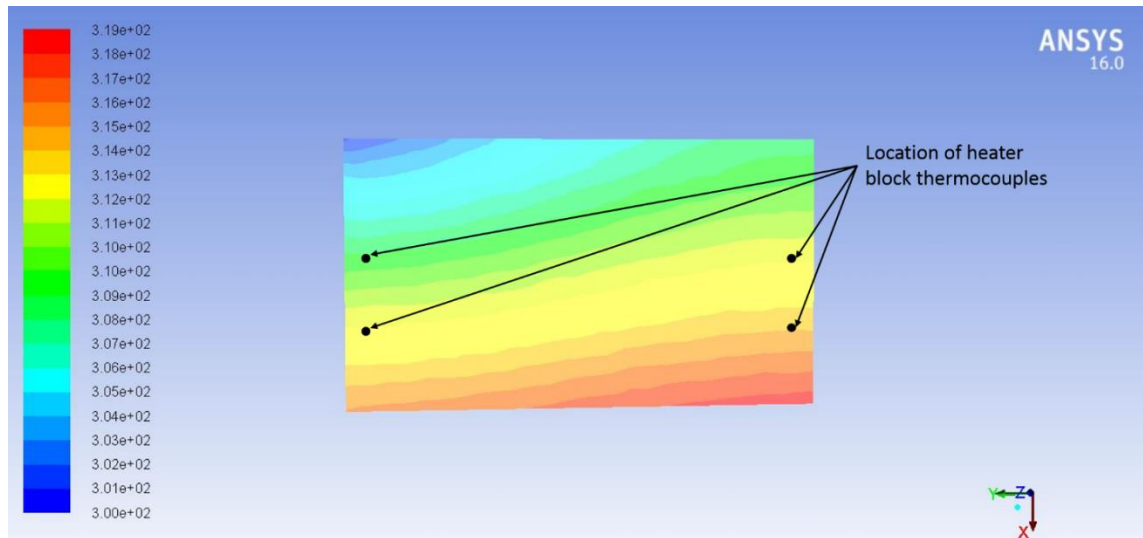


Figure 5.31: Temperature distribution in the heater block, showing thermocouple readout locations.

A plot of the modelled surface temperature and water temperature along the length of the finned area is shown in Figure 5.32. This shows that the water temperature rises swiftly compared to the baseplate surface temperature, and that the slopes of both curves begin to match near the water outlet. This behavior is similar to that seen in constant heat flux cases, and with a longer contact region between the plate and the fluid it is possible that the curves would become parallel once fully developed. However, as the entry region seems to dominate over the actual length of the finned area, it is unclear whether the constant surface temperature assumption is more adequate than the constant heat flux assumption. Both are approximations of the mean temperature difference between the surface and the water. Calculating the true mean temperature difference from the model yields a value of 6.2K. By the constant heat flux assumption, the interpretation of mean

temperature difference would be 2.8K, while by using the constant surface temperature assumption, the mean temperature difference approximation yields 6.4K. Thus, while both assumptions do not fit the situation perfectly, it is concluded that the constant surface temperature assumption yields a better representation of mean temperature difference than the constant heat flux assumption in this situation.

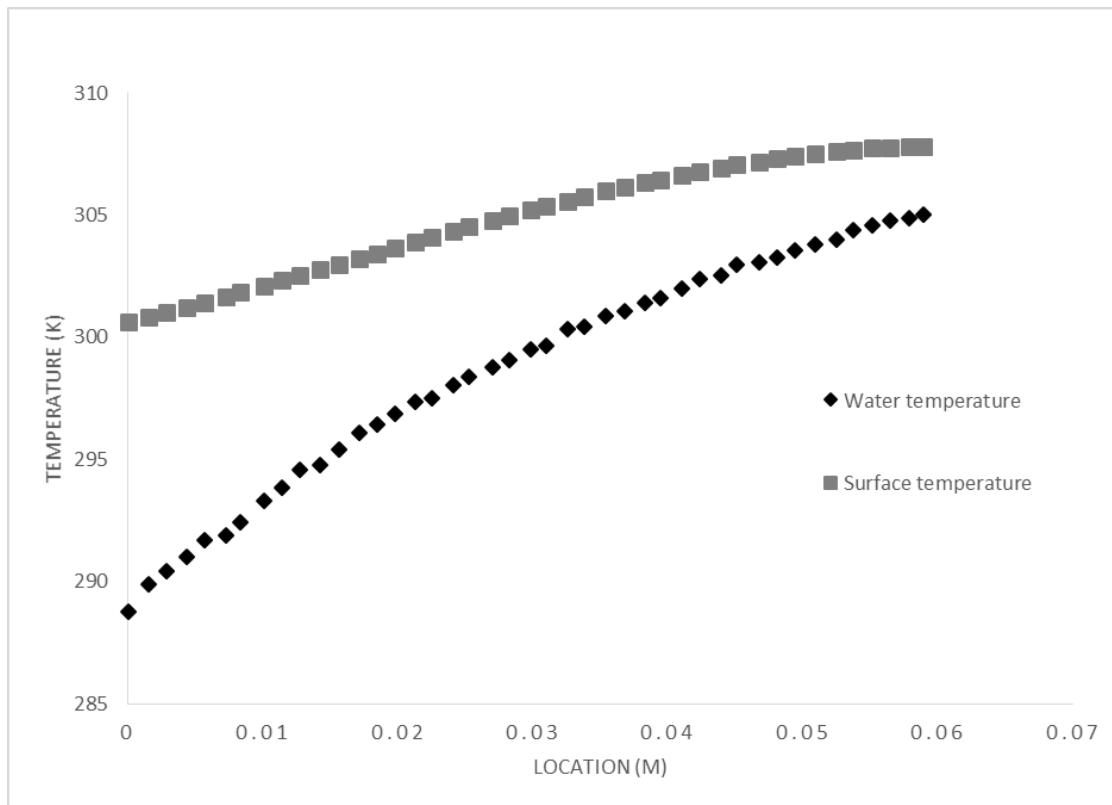


Figure 5.32: Water temperature and surface temperature along the length of the finned area

6 CONCLUSIONS

6.1 Summary of Results

This work explored the feasibility of powder recycling for cold spray additive manufacturing of pyramidal pin fins for water-cooled graphics processing units with the main goal of reducing production costs. An analysis was done to determine the most cost-effective spray parameters while avoiding equipment limitations, showing that higher temperature and pressure were key to lowering costs. Feedstock powder was reclaimed and analyzed in comparison to as-received powder. The reclaimed powder morphology showed that most particles were severely deformed when compared to their as-received counterparts. This however did not have a severe impact on the microstructure of coatings deposited with this reclaimed powder. Although the DE of the pure reclaimed powder was notably lower than the as-received powder, it was found that mixing 50% reclaimed powder with 50% as-received powder results in only a 4% reduction in DE in comparison to the as-received powder. Analyzing in-flight particle velocities revealed that the reclaimed powder particles were on average 60m/s slower than the as-received powder particles, mainly as a result of their large average size. The hardness of the reclaimed coatings was also found to be lower than the as-received, possibly indicating lower cohesion and therefore lower contact between particles. The coatings were annealed, and it was shown that this caused sintering of the particle boundaries, an effect that could improve cohesion. Porosity in all coatings was low, thus demonstrating the viability of powder recycling. These findings suggest that it would be possible to spray pin fin arrays using reclaimed powders with a potential cost reduction of approximately 39% if a suitable reclaimed powder with as-received powder mixture can be found.

Having produced full sized pin fin arrays, the heat transfer performance of the pin fin array heat sinks was tested. Of the two fin configurations tested, it was determined that the fins causing the greater flow obstruction resulted in higher conductance, possibly due to better fluid mixing and higher flow velocity, or the larger surface area available for convection. When comparing conductance to effective pump power, it was found that the higher obstruction fins still had the best performance. Furthermore, annealing of the fins was also found to slightly improve their heat transfer performance. Finally, a CFD model was developed to investigate the soundness of the constant temperature surface assumption for the interpretation of the experimental results. It was concluded that, while the surface temperature variation was larger than initially calculated, the constant temperature surface assumption fit the situation better than the constant heat flux assumption.

6.2 Future Work

While the sprayability of the reclaimed powder was proven, there remain many experiments to be done to determine whether the performance of pin fins made with this powder is still adequate. Further tests must be done in the characterization of the reclaimed powder in order to determine why it experiences such a drop in DE, and whether this can be improved. Thus further beneficial studies that could be undertaken are as follows:

1. Determine a more efficient powder reclamation apparatus as low powder reclamation efficiency was one of the biggest limitations in this work.
2. Determine an optimal reclaimed powder with as-received powder mixing percentage. The 50-50 mix provided an interesting avenue for future work by

demonstrating that there is a benefit in terms of DE from mixing the reclaimed powder with the as-received powder.

3. Further analysis of the reclaimed powder is needed; nano-hardness testing to determine the level of work hardening, TEM analysis to determine the level of oxidation, and size distribution analysis. With this information it may be possible to determine the reason for the lower DE in the reclaimed powder, which could indicate how this may be improved.
4. Spray full pin fin arrays with recycled powder and determine their heat transfer performance. This would confirm whether recycled powder can be used for the production of pin fin arrays. Unfortunately, due to limitation in the availability of reclaimed powder in this study, this step was not undertaken.
5. Testing of the thermal conductivity of the coatings made with as-received, reclaimed, and mixed powder. This would give a good indication of the heat transfer performance of pin fin array sprayed with each respective powder.
6. Compare the thermal performance of pin fins produced by CGDS to fins with the same profile made from bulk copper (machined).
7. Improve the cost effectiveness of producing pin fins by CGDS by reducing the ratio of mesh DE to substrate DE. As stated, the greatest loss in powder if powder recycling can be done effectively, is the buildup on the mesh and mask. To improve on this loss, either increasing the substrate DE, through substrate preheating, or decreasing the mask DE by exploring new masking materials or treatments that result in less buildup could be done.

8. Redesign the heat transfer test rig to place thermocouples closer to the surface of the pin fin array baseplate. This would eliminate the need to interpolate for the surface temperature and eliminate any error therefrom.
9. Expand the simulation to determine the ideal fin shape, height, and spacing. Following this, it would be interesting to test this ideal fin configuration in an experimental apparatus.
10. Determine a fin efficiency equation specific to the fin shapes seen in this work, as opposed to using the modified fin efficiency for triangular pin fins.

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Appendix – A

The MatLab code used to calculate the fin conductance, Reynolds numbers, and effective pump power can be found below:

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%
%Title: HeatTestRigSolver_matrices
%
%Function: To calculate the theermal conductance of the pin fin array
heat
%sinks
%
%Input: Text files containing experimental results
% m_dot: mass flow rate of water (kg/s)
% DP: pressure drop accross finned area (Pa)
% Ti: inlet water temperature (C)
% To: outlet water temperature (C)
% T1_b: bottom inlet block temperature (C)
% T1_t: top inlet block temperature
% T2_b: bottom outlet block temperature (C)
% T2_t: top outlet block temperature (C)
% Qerr: Error of heat transfer measurments (W)
%
%Output: Text files containing thermla conductance, effective pump
power
%and reynolds number
% UA: thermal conductance (W/K)
% Re: Reynold number
% ev: effective pump power (W)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%
function HeatTestRigSolver_matrices

    %Opening files with experimental results
    fileID1 = fopen('A001_90V.txt','r');

    fileID2 = fopen('AA001_90V.txt','r');

    fileID3 = fopen('A002_90V.txt','r');

    fileID4 = fopen('AA002_90V.txt','r');

    fileID5 = fopen('X00_90V.txt','r');

    fileID6 = fopen('AX00_90V.txt','r');

    formatSpec = '%f %f %f %f %f %f %f %f %f';
    sizeA = [9 Inf];
    names = [fileID1, fileID2, fileID3, fileID4, fileID5, fileID6];
```

```

numFiles = 6;

%flag = 1 for F2.0, 2 for F1.8, 3 for NF
flag = [1, 1, 2, 2, 3, 3];

%output matrices
UAOut = zeros(numFiles,6);
ReOut = zeros(numFiles,6);
UAErrOut = zeros(numFiles,6);
evOut = zeros(numFiles,6);

%loop through each file
for i = 1:numFiles
    A = fscanf(names(i) ,formatSpec,sizeA);

    %Water properties
    m_dot = A(1,:);
    DP = A(2,:);
    Ti = A(3,:);
    To = A(4,:);

    x = length(m_dot);

    rho = 1000;           %density
    mu = 7.97*10^-4;      %viscosity
    Cp = 4187;            %Cp of water

    %Copper
    Kc = 387;             %conductivity of copper

    %Thermocouple measurments
    delta_l = 0.012;      %distance between thermocouples
    delta_s = 0.016;      %distance between top thermocouple and
surface

    T1_b = A(5,:);
    T1_t = A(6,:);
    T2_b = A(7,:);
    T2_t = A(8,:);
    Qerr = A(9,:);

    %check what fin configuration
    if flag(i) == 1 || flag(i) == 2
        %Fin dimensions
        B = 1.5*10^-3;      %Base
        H = 1.7*10^-3;      %Height
        S = 0.5*10^-3;      %Passage at base
        if flag(i) == 1
            theta = 9*pi/180; %angle from vertical
        end
        if flag(i) == 2
            theta = 17*pi/180; %angle from vertical
        end
    end
end

```

```

end

%Across dimensions
Bc = sqrt(2*B^2);
Sc = sqrt(2*S^2);

%Area
e1_1 = Bc + Sc;
A1 = e1_1*H;

%Area 2
A2 = (Sc+H*tan(theta))*H;

%fin areas
W = 49.5*10^-3;      %flow channel width
L = 64*10^-3;        %flow channel length
t = 1/8*0.0254;
N = 787;              %Number of fins

%Hydraulic diameter
Aflow = min(A1,A2);
Pflow = 2*Sc + 2*H/cos(theta) + 2*H*tan(theta);
Dh = 4*Aflow/Pflow;

N_obs = W/(Bc+Sc);   %number of fins widthwise

%Fin area calculations
Alfin = 4*(B-H*tan(theta))*H/(cos(theta));
Albase = B^2;
A_baseplate = W*L;
Afin = Alfin*N;
Aunfin = A_baseplate - Albase*N;
At = Afin + Aunfin;
elseif flag(i) ==3
    Aflow = 1.7*10^-3*49.5*10^-3;
    Pflow = 2*1.7*10^-3 + 2*49.5*10^-3;
    Dh = 4*Aflow/Pflow;
end

%outputs
Re = zeros(1,6);
LMDT = zeros(1,6);
Q = zeros(1,6);
nf_sol = zeros(1,6);
n0_sol = zeros(1,6);
solh = zeros(1,6);
UA = zeros(1,6);
UAerr = zeros(1,6);
ev = zeros(1,6);

%loop for different flow rates
for j = 1:x

    %heat flux
    q1 = Kc*(T1_t(j) - T1_b(j))/delta_1;      %heat flux at front

```

```

q2 = Kc*(T2_t(j) - T2_b(j))/delta_l;    %heat flux at back

R_block = (delta_l+delta_s)/Kc;           %block resistance
R_tc = 1/(10*10^4);                     %contact resistance
R_baseplate = t/Kc;                     %baseplate
resistance

%1-D interpolation of surface temperature
T1_s = q1*(R_block + R_tc + R_baseplate) + T1_b(j);
T2_s = q2*(R_block + R_tc + R_baseplate) + T2_b(j);

if flag(i) == 1 || flag(i) == 2
    %velocity
    V1 = m_dot(j)/(A1*N_obs*rho);
    V2 = m_dot(j)/(A2*N_obs*rho);

    %characteristic speed
    Vflow = max(V1,V2);

    %Reynolds number
    Re(j) = rho*Vflow*Dh/mu;

    %effective pump power
    ev(j) = m_dot(j)/rho*DP(j);
elseif flag(i) == 3
    Vflow = m_dot(j)/(Aflow*rho);
    Re(j) = rho*Vflow*Dh/mu;
    ev(j) = m_dot(j)/rho*DP(j);
end

%log mean temperature difference
LMDT(j) = ((T2_s - To(j)) - (T1_s - Ti(j)))/(log((T2_s -
To(j))/(T1_s - Ti(j))));

%Heat into water
Q(j) = m_dot(j)*Cp*(To(j) - Ti(j));

if flag(i) == 1 || flag(i) == 2
    syms h
    %fin efficiency
    m = sqrt(4*h/(Kc*B));
    nf = 2*besseli(2,2*m*H)/(m*H*besseli(1,2*m*H));

    %surface efficiency
    n0 = 1 - Afin*(1 - nf)/At;

    %Convection coefficient
    eqn = Q(j)/(LMDT(j)*At) == h*n0;

    %solve for h
    solh(j) = solve(eqn,h);

    %output solutions

```

```

    %fin parameter
    m = sqrt(4*solh(j)/(Kc*B));

    %trianglular fins efficiency
    nf_sol(j) = 2*besseli(2,2*m*H)/(m*H*besseli(1,2*m*H));

    %straight fins efficiency
    %nf_sol(i) = tanh(m*(H+B/4))/(m*(H+B/4));

    %surface efficiency
    n0_sol(j) = 1 - Afin*(1 - nf_sol(j))/At;

    %conductance
    UA(j) = solh(j)*At*n0_sol(j);

    %errors
    herr = Qerr(j)/(LMDT(j)*At*n0_sol(j));
    UAerr(j) = herr*At*n0_sol(j);
elseif flag(i) == 3
    %bare plate solutions
    solh(j) = Q(j)/(W*L*LMDT(j));
    UA(j) = solh(j)*W*L;
    herr = Qerr(j)/(LMDT(j)*W*L);
    UAerr(j) = herr*W*L;
end
end
%output matrices
UAOut(i,:) = UA;
ReOut(i,:) = Re;
UAErrOut(i,:) = UAerr;
evOut(i,:) = ev;

end

%output to text files for data processing
FileIDUA = fopen('UA_Results.txt', 'wt');
for k=1:numFiles
    fprintf(FileIDUA, '%f\t', UAOut(k,:));
    fprintf(FileIDUA, '\n');
end

FileIDRe = fopen('Re_Results.txt', 'wt');
for k=1:numFiles
    fprintf(FileIDRe, '%f\t', ReOut(k,:));
    fprintf(FileIDRe, '\n');
end

FileIDUAErr = fopen('UAerr_Results.txt', 'wt');
for k=1:numFiles
    fprintf(FileIDUAErr, '%f\t', UAErrOut(k,:));
    fprintf(FileIDUAErr, '\n');
end

FileIDev = fopen('ev_Results.txt', 'wt');

```

```
for k=1:numFiles
    fprintf(FileIDev,'%e\t',evOut(k,:));
    fprintf(FileIDev,'\n');
end

end
```