

# **Performance, Manufacturability and Mechanical Properties of Near-Net Shaped Pyramidal Fin Arrays for Compact Heat Exchangers Produced using Cold Spray as an Additive Manufacturing Technique**

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# Abstract

Significant efforts have been made in the last decades to decrease the world's dependency to fossil fuels. One of the fronts which has shown major improvement is gas turbine efficiency. To this end, components such as recuperators have been developed to recover heat that is usually trapped and wasted in the exhaust gases of combustion processes. Brayton Energy Canada (BEC) has recently developed a promising compact heat exchanger that could be used as a recuperator in gas turbines. Nevertheless, this novel type of Wire Mesh Heat Exchanger (WMHE) still has room for improvement, especially regarding the way that its fin arrays are manufactured due to the fact that the technique presently used is time consuming and consequently costly.

Cold Gas Dynamic Spraying (CGDS or simply cold spray) has improved significantly during the last decade in terms of materials that can be deposited and fundamental understanding of the physics behind this process. This deposition technique relies on the transfer of kinetic energy of a driving gas to solid particles, which deform and adhere onto a substrate upon impact. This work aims to manufacture near-net shaped pin fin arrays using CGDS as an additive manufacturing technique by selectively covering the substrate by the means of a mask.

The theoretical and experimental approaches considered for the development of viable near-net shaped fin arrays will be presented in this thesis. The objectives of this research work are to demonstrate the feasibility of using CGDS as an additive manufacturing technique to produce pin fin arrays, to investigate the cold spray parameters, to evaluate the thermal and hydrodynamic performances of this new type of pin fin created, to find the effect of



geometric parameters such as fin density and height on the performances, to determine the viability of the sprayed pin fins in a real environment by means of finding mechanical properties such as adhesion strength, to prove the possibility of producing a streamwise material anisotropic fin arrays, and finally to understand the different adhesion mechanisms by means of numerical modeling of the relevant impact physics.

Near-net shaped fin arrays were successfully manufactured by the CGDS system available at the University of Ottawa Cold Spray Laboratory as well as by the cold spray system used at BEC. The geometric variables of the pin fin arrays were quantified by three-dimensional optical microscopy while the pin fin microstructure was characterized by two-dimensional optical microscopy and Scanning Electron Microscopy (SEM). The size of the feedstock powders utilized for this work was quantified by laser diffraction analysis while the shape of these powders was characterized by SEM. The thermal and hydrodynamic performance was found by means of an in-house test apparatus developed specifically for the scope of this thesis. Adhesion onto the substrate was evaluated by a standardized shear test: EN15340-2007. Fracture surface analysis of these shear tests was performed using SEM. A finite element analysis of cold sprayed particle impact on substrates and on already deposited particles was conducted in order to help elucidate at a fundamental level the experimental results obtained.

Results of this work show that it is possible to manufacture fin arrays using cold spray as an additive manufacturing technique by using a wire mesh mask to selectively cover some of the substrate area, and then producing pin fins. Spray parameters were found in order to be able to spray without clogging the chosen mask and to build the pin fins on the substrate with a low porosity level. Several geometric parameters, for instance height, fin density and base

angle, were investigated. It was found that the fin base angle has a minor influence on the performance whereas the fin density and height have a significant effect on the performance. The shear strength of the pin fin arrays were found to be sufficiently large to sustain the theoretical drag pressures encountered in typical application scenarios. Streamwise anisotropic materials fin arrays were successfully deposited and tested, and were found to outperform single-material fin arrays made of high efficient service temperature alloys. Different failure mechanisms and shear strengths were identified for the materials tested. It was found that the shear strengths measured seem to correlate qualitatively with the equivalent plastic strain (PEEQ) at the fracture surface determined by numerical simulation, making the model a potential predictive tool for shear strength.

## Preface

This thesis is presented as a series of articles. A statement regarding my contributions and of the co-authors is given here.

I have co-authored (first author) the first paper entitled “Net Shape Fins for Compact Heat Exchanger Produced by Cold Spray” presented in section 5.1 (pages 105 to 118), with Mr. Philippe Dupuis, Dr. Bertrand Jodoin and Mr. Antoine Corbeil. I have contributed directly to this work by performing the design, manufacturing, testing and validation of the performance test fixture described in this work, in collaboration with Mr. Dupuis, after a thorough literature review of the relevant techniques used for such performance testing. I have produced samples for this work, as well as conducted the hydrodynamic and heat transfer tests required for the performance evaluation of the fin arrays manufactured. I have contributed directly to the production of the written document (production of the figures, tables, typesetting, submission and review process) as well as the elaboration of the draft document itself through the interpretation of the performance data results the experimental testing has generated. Mr. Dupuis has produced some samples included in this work, as well as helped me for the hydrodynamic and thermal performance testing. Dr. Jodoin, my thesis supervisor, guided me on the required tests to be performed. He also provided insights and reviewed the manuscript to ensure its quality prior to submission. Mr. Corbeil, listed as the fourth author, is the president of Brayton Energy Canada, industrial sponsor for this work. He also assisted to direct the investigations towards the commercial concerns and set out several technical-engineering requirements. I also used some spray equipment at his facility. I am the first author of the paper since all the co-authors agreed that I have been the one to contribute the most to the technical/engineering content of the paper.

I have co-authored (second author) the second paper entitled “Performance evaluation of near-net pyramidal shaped fin arrays manufactured by cold spray” presented in section 5.2 (pages 119 to 129), with Mr. Philippe Dupuis, Mr. Aslan Farjam, Dr. Bertand Jodoin and Mr. Antoine Corbeil. I have contributed directly to this work by manufacturing the cold sprayed samples and by conducting their hydrodynamic and thermal performance testing. I have contributed directly to the production of the written document (production of the figures, tables, typesetting, submission and review process) as well as the elaboration of the draft document itself through the interpretation of the performance data results that the experimental testing has generated. Mr. Dupuis, the first author, has produced the plain rectangular (straight cut) fin arrays for this work, as well as tested their performance and performed the post-processing grinding operations for the cold sprayed samples that I produced. Mr. Farjam, the third author, has performed the measurements of the fin dimensions reported in this work. Dr. Jodoin, my thesis supervisor, guided me on the required tests to be performed. He also provided insights and reviewed the manuscript to ensure its quality prior to submission. Mr. Corbeil, listed as the fifth author, is the president of Brayton Energy Canada, industrial sponsor for this work. He also assisted to direct the investigations towards the commercial concerns and set out several technical-engineering requirements. I also used some spray equipment at his facility.

I have co-authored (first author) the third paper entitled “Additive manufacturing of pyramidal pin fins: Height and fin density effects under forced convection” presented in section 5.3 (pages 130 to 140), with Mr. Philippe Dupuis, Mr. Aslan Farjam, Dr. Bertrand Jodoin and Mr. Antoine Corbeil. I have contributed directly to this work by producing the different fin arrays, as well as by conducting the hydrodynamic and heat transfer tests

required for the performance evaluation of the fin arrays manufactured. I have also produced the Nusselt number correlations included in this manuscript. I have contributed directly to the production of the written document (production of the figures, tables, typesetting, submission and review process) as well as the elaboration of the draft document itself through the interpretation of the performance data results that the experimental testing has generated. Mr. Dupuis performed the post-processing grinding operations for the samples included in this work. Mr. Farjam, the third author, has performed the measurements of the fin dimensions reported in this work. The fourth author, Dr. Jodoin, my thesis supervisor, guided me on the required tests to be performed. He also provided insights and reviewed the manuscript to ensure its quality prior to submission. Mr. Corbeil, listed as the fifth author, is the president of Brayton Energy Canada, industrial sponsor for this work. He also assisted to direct the investigations towards the commercial concerns and set out several technical-engineering requirements. I also used some spray equipment at his facility. I am the first author of the paper since all the co-authors agreed that I have been the one to contribute the most to the technical/engineering content of the paper.

I have co-authored (first author) the fourth paper entitled “Mechanical Properties of Cold Gas Dynamic-Sprayed Near-Net-Shaped Fin Arrays” presented in section 5.4 (pages 141 to 154), with Mr. Philippe Dupuis, Dr. Bertrand Jodoin and Mr. Antoine Corbeil. I have contributed directly to this work by producing every sample tested. Also, all the results presented were outcomes of experiments that I conducted. In addition, I designed and manufactured the special holder tool to test the shear strength of the pyramidal fins according to the standard used. I have produced the written document (production of the figures, tables, typesetting, submission and review process) as well as elaborated the draft document itself

through the interpretation of the results that the experimental testing has generated. Mr. Dupuis, listed as second author, participated in discussions on this subject, provided useful insights and reviewed the manuscript prior to submission. Dr. Jodoin, my thesis supervisor, guided me on the required tests to be performed. He also provided insights and reviewed the manuscript to ensure its quality prior to submission. Mr. Corbeil, listed as the fourth author, is the president of Brayton Energy Canada, industrial sponsor for this work. He also assisted to direct the investigations towards the commercial concerns and set out several technical-engineering requirements. I also used some spray equipment at his facility. I am the first author of the paper since all the co-authors agreed that I have been the one to contribute the most to the technical/engineering content of the paper.

I have co-authored (first author) the fifth paper entitled “Pyramidal Fin Arrays Performance Using Streamwise Anisotropic Materials by Cold Spray Additive Manufacturing” presented in section 5.5 (pages 155 to 168), with Mr. Philippe Dupuis, Dr. Bertrand Jodoin and Mr. Antoine Corbeil. I have contributed directly to this work by producing every sample tested. Also, all the results presented were outcomes of the experiments that I conducted except for the thermal conductivity tests that were outsourced since I did not have access to a laser flash diffusivity testing apparatus at the University of Ottawa. I have produced the written document (production of the figures, tables, typesetting, submission and review process) as well as elaborated the draft document itself through the interpretation of the results that the experimental testing has generated. Mr. Dupuis, listed as second author, participated in discussions on this subject, provided useful insights and reviewed the manuscript prior to submission. Dr. Jodoin, my thesis supervisor, guided me on the required tests to be performed. He also provided insights and reviewed the manuscript to ensure its quality prior

to submission. Mr. Corbeil, listed as the fourth author, is the president of Brayton Energy Canada, industrial sponsor for this work. He also assisted to direct the investigations towards the commercial concerns and set out several technical-engineering requirements. I am the first author of the paper since all the co-authors agreed that I have been the one to contribute the most to the technical/engineering content of the paper.

I have co-authored (first author) the last paper entitled “Finite Element Analysis and Failure Mode Characterization of Pyramidal Fin Arrays Produced by Masked Cold Gas Dynamic Spray” presented in section 5.6 (pages 169 to 186), with Mr. Philippe Dupuis, Dr. Bertrand Jodoin and Dr. Abbas Ghaei. I have contributed directly to this work by producing every sample tested. Also, all the results presented were outcomes of the experiments that I conducted. In addition, I developed the numerical model used to simulate cold spray impacts. I have produced the written document (production of the figures, tables, typesetting, submission and review process) as well as elaborate the draft document itself through the interpretation of the data results that the experimental testing has generated. Mr. Dupuis, listed as second author, participated in discussions on this subject, provided useful insights and reviewed the manuscript prior to submission. Dr. Jodoin, my thesis supervisor, guided me on the required tests to be performed. He also provided insights and reviewed the manuscript to ensure its quality prior to submission. Dr. Ghaei, the fourth author, provided a subroutine to implement the high strain rate Preston-Tonks-Wallace plasticity model used. I am the first author of the paper since all the co-authors agreed that I have been the one to contribute the most to the technical/engineering content of the paper.

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I would also take this opportunity to thank my colleague and my good friend, Philippe Dupuis for all the valuable discussions we had about heat transfer and cold spray. Your knowledge and your help regarding the English grammar are more than appreciated. All the time you spent reviewing the articles as well as my thesis, to me, is worth its weight in gold. We started working together in second years of bachelor's degree and I still appreciate working with you. It will be a pleasure to work with you again.

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# List of Symbols

## Nomenclature

|                  |                                                         |
|------------------|---------------------------------------------------------|
| $\alpha$         | Thermal diffusivity [m <sup>2</sup> /s]                 |
| $\beta$          | High strain rate exponent                               |
| $\Gamma$         | Grüneisen parameter                                     |
| $\gamma$         | Strain rate dependence constant                         |
| $\Delta P$       | Differential pressure [Pa]                              |
| $\Delta T$       | Overall temperature difference [K]                      |
| $\Delta T_1$     | Inlet temperature difference [K]                        |
| $\Delta T_2$     | Outlet temperature difference [K]                       |
| $\Delta T_{lm}$  | Log-mean temperature difference [K]                     |
| $\Delta m$       | Mass gained by the sample during the spray process [kg] |
| $\delta_1$       | Relative deformation error [%]                          |
| $\delta_2$       | Relative penetration error [%]                          |
| $\delta_3$       | Mean absolute error [m]                                 |
| $\varepsilon$    | Effectiveness                                           |
| $\epsilon_p$     | Plastic strain                                          |
| $\epsilon_d$     | Elastic deviatoric strain                               |
| $\dot{\epsilon}$ | Strain rate [1/s]                                       |
| $\eta$           | Compressive strain                                      |
| $\eta$           | Efficiency                                              |
| $\eta_{fan}$     | Fan efficiency                                          |
| $\eta_o$         | Overall surface efficiency                              |
| $\theta$         | Strain hardening rate                                   |
| $\kappa$         | Temperature dependence constant                         |
| $\mu$            | Dynamic viscosity [Pa·s]                                |
| $\xi$            | Material parameter [1/s]                                |
| $\rho$           | Density [kg/m <sup>3</sup> ]                            |
| $\sigma$         | Flow stress [Pa]                                        |
| $\hat{\tau}_s$   | Normalized work hardening saturation stress [Pa]        |
| $\hat{\tau}_y$   | Normalized yield stress [Pa]                            |
| $A$              | Area [m <sup>2</sup> ]                                  |
| $a$              | Absorptivity                                            |
| $B$              | Material constant [Pa]                                  |
| $C$              | Heat capacity rate [W/K]                                |
| $C_r$            | Capacity ratio                                          |
| $c$              | Speed of sound [m/s]                                    |
| $c_p$            | Specific heat capacity [J/(kg·K)]                       |
| $D$              | Pin fin diameter [m]                                    |
| $DE$             | Deposition efficiency [%]                               |
| $D_E$            | Particle deformation from the experiment [m]            |
| $D_h$            | Hydraulic diameter [m]                                  |
| $D_m$            | Particle deformation from the model [m]                 |

|            |                                                               |
|------------|---------------------------------------------------------------|
| $d$        | Deviatoric stress [Pa]                                        |
| $E$        | Internal energy [J]                                           |
| $e$        | Specific internal energy [J/kg]                               |
| $e$        | Pumping power [kW]                                            |
| $FD$       | Fin density [fin/m]                                           |
| $f$        | Darcy friction factor                                         |
| $G_0$      | Shear modulus at 0 K [Pa]                                     |
| $G_e$      | Elastic shear modulus [Pa]                                    |
| $G_p$      | Plastic shear modulus [Pa]                                    |
| $H$        | Fin height [m]                                                |
| $H_c$      | Corrected fin height [m]                                      |
| $h$        | Convective heat transfer coefficient [W/(m <sup>2</sup> ·K)]  |
| $h$        | Specific enthalpy [J/kg]                                      |
| $I_1$      | Bessel function of first order                                |
| $I_2$      | Bessel function of second order                               |
| $J$        | Temperature material constant [K]                             |
| $K_c$      | Contraction coefficient                                       |
| $K_e$      | Expansion coefficient                                         |
| $k$        | Thermal conductivity [W/(m·K)]                                |
| $L$        | Wall thickness [m]                                            |
| $L_f$      | Length of the heat exchanger [m]                              |
| $\ell$     | Flow length between the major boundary layer disturbances [m] |
| $M$        | Mass [kg/s]                                                   |
| $M_a$      | Atomic mass [kg/atom]                                         |
| $M_0$      | Total mass of powder injected in the system [kg]              |
| $\dot{M}$  | Mass flow rate [kg/s]                                         |
| $m$        | Fin heat transfer parameter [m <sup>-1</sup> ]                |
| $N$        | Number                                                        |
| $N_r$      | Number or row                                                 |
| $Nu$       | Nusselt number                                                |
| $NTU$      | Number of transfer units                                      |
| $n$        | Strain hardening constant                                     |
| $n$        | Number of nodes                                               |
| $P$        | Pressure [Pa]                                                 |
| $P_E$      | Particle penetration from the experiment [m]                  |
| $P_m$      | Particle penetration from the model [m]                       |
| $P_{flow}$ | Flow perimeter [m]                                            |
| $p$        | Energy density [J/m <sup>2</sup> ]                            |
| $q$        | Heat transfer rate [W]                                        |
| $R$        | Thermal resistance [K/W]                                      |
| $Re$       | Reynolds number                                               |
| $R_{eq}$   | Equivalent thermal resistance [K/W]                           |
| $s$        | Hugoniot slope                                                |
| $s_0$      | Saturation stress at 0 K                                      |
| $s_\infty$ | Saturation stress at melting                                  |



|                 |                                                           |
|-----------------|-----------------------------------------------------------|
| $T$             | Temperature [K]                                           |
| $T_m$           | Melting temperature [K]                                   |
| $\hat{T}$       | Temperature ratio                                         |
| $t$             | Fin thickness [m]                                         |
| $t_{1/2}$       | Half of the maximal temperature increase [s]              |
| $U$             | Overall heat transfer coefficient [W/(m <sup>2</sup> ·K)] |
| $UA$            | Thermal conductance [W/K]                                 |
| $V$             | Volume [m <sup>3</sup> ]                                  |
| $\dot{V}$       | Volumetric flow rate [m <sup>3</sup> /s]                  |
| $u$             | Velocity [m/s]                                            |
| $\bar{u}$       | Mean fluid velocity [m/s]                                 |
| $w$             | Fin width [m]                                             |
| $\gamma_0$      | Yield stress constant at 0 K                              |
| $\gamma_1$      | Medium strain rate constant                               |
| $\gamma_2$      | Medium strain rate exponent                               |
| $\gamma_\infty$ | Yield stress constant at melting                          |
| $\gamma_{Ei}$   | $i$ y coordinate of nodes for the experiment [m]          |
| $\gamma_{mi}$   | $i$ y coordinate of nodes for the model [m]               |

## ***Subscripts***

|          |                             |
|----------|-----------------------------|
| $\infty$ | Free stream properties      |
| $0$      | Before the shock wave       |
| $1$      | After the shock wave        |
| $as$     | Arc spray                   |
| $b$      | Base                        |
| $c$      | Cold side                   |
| $cs$     | Cold spray                  |
| $cond$   | Conduction                  |
| $conv$   | Convection                  |
| $D$      | Based on feature diameter   |
| $Dh$     | Based on hydraulic diameter |
| $f$      | Fin properties              |
| $fl$     | Fluid properties            |
| $flow$   | Flow properties             |
| $H$      | Hugoniot                    |
| $i$      | Inlet properties            |
| $h$      | Hot side                    |
| $m$      | Material properties         |
| $max$    | Maximum                     |
| $min$    | Minimum                     |
| $o$      | Outlet properties           |
| $s$      | Shock properties            |
| $smooth$ | Smooth surface properties   |
| $t$      | Total                       |

|       |                            |
|-------|----------------------------|
| $t_c$ | Thermal contact resistance |
| $v$   | Per unit volume            |
| $w$   | Wall properties            |

## List of Abbreviations

|       |                                                  |
|-------|--------------------------------------------------|
| AM    | Additive Manufacturing                           |
| BEC   | Brayton Energy Canada                            |
| CFD   | Computational Fluid Dynamics                     |
| CGDS  | Cold Gas Dynamic Spray                           |
| CSM   | Cold Spray Meter                                 |
| DPG   | Distributed Power Generation                     |
| EBSD  | Electron BackScatter Diffraction                 |
| EOS   | Equation Of State                                |
| FEA   | Finite Element Analysis                          |
| GZ    | Gao-Zhang                                        |
| HRTEM | High Resolution Transmission Electron Microscope |
| JC    | Johnson-Cook                                     |
| MKHL  | Modified Khan-Huand-Liang                        |
| MT    | MicroTurbines                                    |
| MTS   | Mechanical Threshold Stress                      |
| MZA   | Modified Zerilli-Armstrong                       |
| PIV   | Particle Image Velocimetry                       |
| PTW   | Preston-Tonks-Wallace                            |
| SEM   | Scanning Electron Microscopy                     |
| TEM   | Transmission Electron Microscope                 |
| VA    | Voyiadjis-Abed                                   |
| WMHE  | Wire Mesh Heat Exchanger                         |

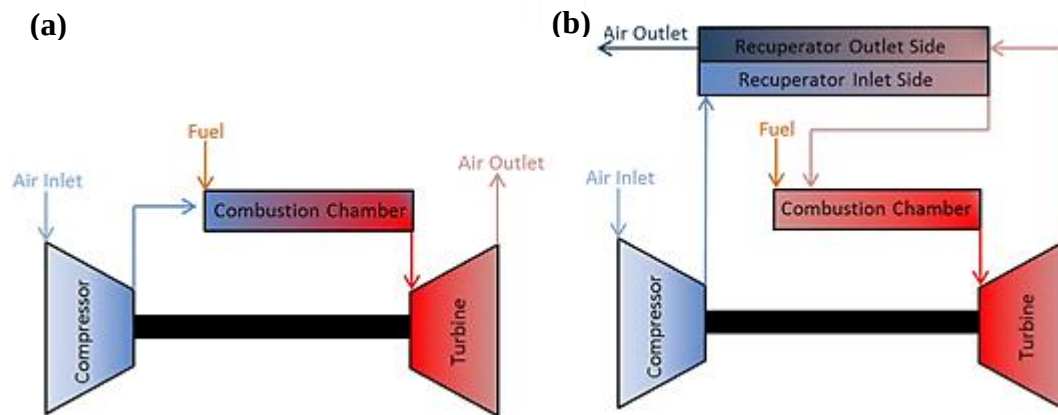
# Chapter 1 - Introduction

## 1.1 Background

Reducing fuel consumption and greenhouse gas emissions are critical design concerns in the 21<sup>st</sup> century. A significant worldwide effort is being made to produce designs with better energy efficiencies. The power generation industry presents a significant potential for the reduction of emissions and fuel consumption. A solution for the reduction of power losses is to use Distributed (decentralized) Power Generation (DPG). This approach does not have long transmission lines because the power plant is close to the end users and the risk of a general electrical failure is minimized because the power is generated for smaller populations [1–4]. However, to make this option competitive and viable, the waste energy from the combustion process used to produce power should be recuperated. Therefore, recuperators have a great potential to become essential in applications where there is waste heat from power production. Recuperators are heat exchangers that increase the energy efficiency of this type of process since they recover the waste heat from the exhaust.

The viability of DPG is linked with the development of high efficiency MicroTurbines (MT) for applications that produce 25 kW to 500 kW. MT are compact gas turbines that produce the power for the DPG as shown in Figure 1.1 (a). It should be noted that in this figure, the colors represent the gas temperature. They transform the thermal energy of the fuel to electricity, where a part of the heat produced is usually wasted at the outlet of the MT since the air at that point is still much hotter than the ambient air temperature. Therefore, the overall energy efficiency can be increased with the use of a recuperator to recover the waste energy from the exhaust stream, and use it to preheat the gas that goes in the turbine, as shown in Figure

1.1 (b). The principal rivals of the MT for the production of electricity for a DPG system are diesel engines. The thermal efficiencies for these engines are approximately 30-35% for the same power production capacity (25 kW to 500 kW) [5]. The thermal efficiency is the ratio of work produced over the total energy input to the system. MT equipped with a recuperator have the potential to reach more than 50% thermal efficiency [5].

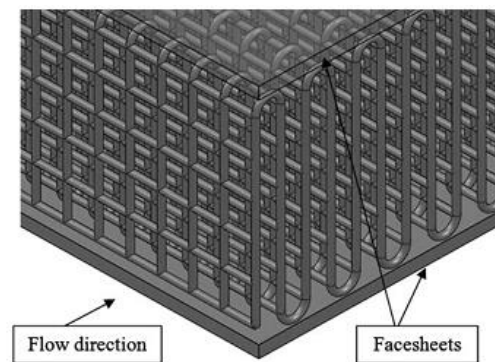


**Figure 1.1: Typical gas turbine diagram with a colored gradient representing air temperature: (a) Simple cycle (b) Regenerative cycle**

Presently, most commercially available recuperators are large, expensive and have an effectiveness of 80-85%. The effectiveness of a heat exchanger is defined as the ratio of the heat transfer rate from one fluid to another fluid over the maximum possible heat transfer rate. For the MT to be an excellent solution for the DPG, the recuperators require an additional effectiveness of 5-10% and need to be more compact [5]. Traditionally, they were manufactured using packed beds/tubes, microchannels or metallic foams. However, these solutions had problems such as leakage, high pressure losses, axial conduction losses and high manufacturing cost [6–8].

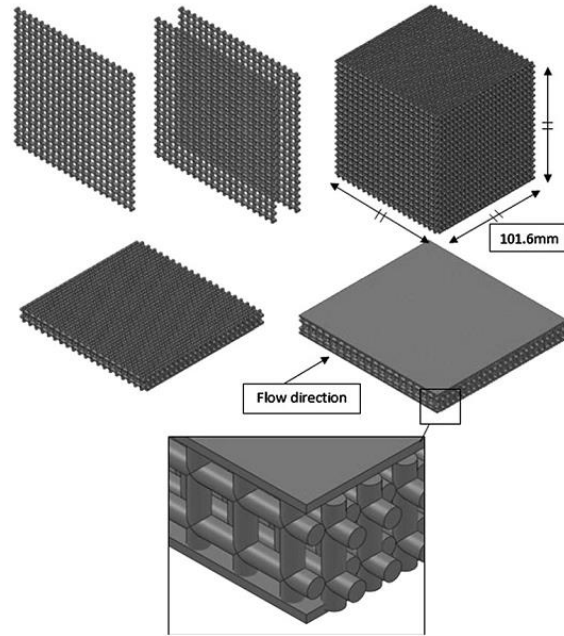
A new design of a high efficiency compact recuperator that could be used in MT has been recently developed by Brayton Energy Canada (BEC). Their design is classified as a Wire

Mesh Heat Exchanger (WMHE). Since the introduction of the WMHE in the late 1980s, many mesh configurations were tested [6, 9, 10]. The wire mesh is used as a conductive material in the heat exchanger. WMHE are more effective, easier to manufacture and relatively inexpensive compared to other types of heat exchanger [11]. Before BEC's improvements, the WMHE was produced with folded wire meshes, by brazing thin sheets to the tips of the folds to act as parting plates, as shown in Figure 1.2.



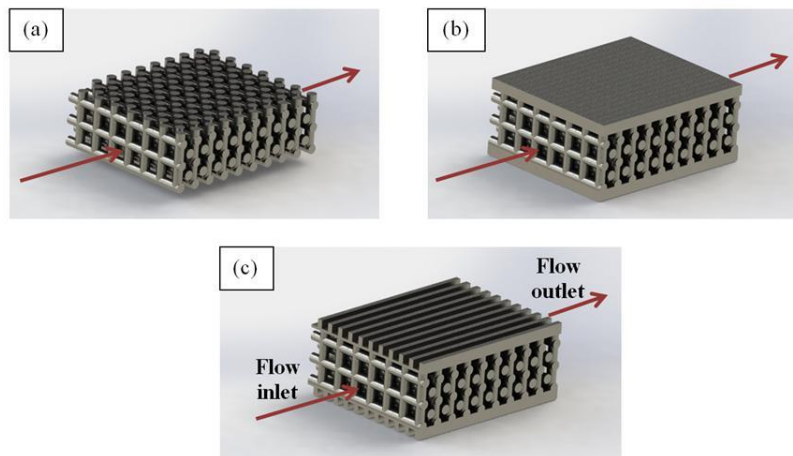
**Figure 1.2: WMHE with folded wire screen and brazed metallic sheets [12]**

This approach is reasonably simple to manufacture. However, the small contact surface between the brazed sheet and the mesh screen due to the large radius of curvature of the wire mesh, limits the conduction to the outer walls by limiting the number of folds per unit length. An advancement of this method is to use a wire mesh brick manufactured by means of sintering a stack of woven wire textiles, cutting this stack into thin wafers and then sealing them by brazing metallic sheets on each side as shown in Figure 1.3 [11].



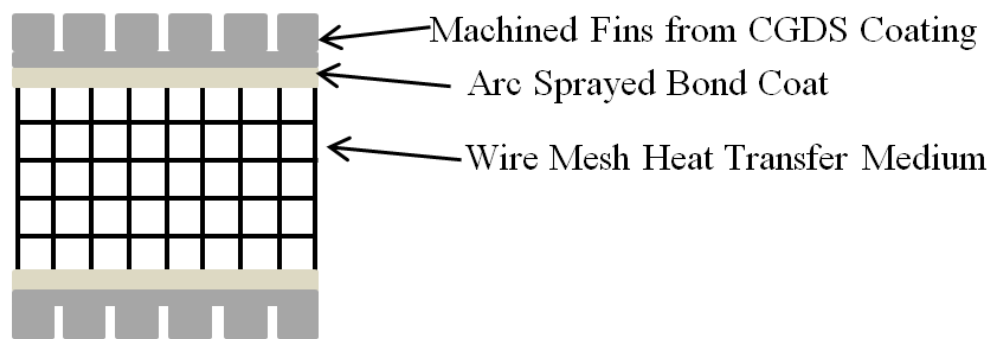
**Figure 1.3: WMHE with stack wire mesh and brazed metallic sheets [12]**

The inconvenience of this method is the cost of the brazing process. BEC's improved design uses this procedure except that the brazing operation is replaced by an innovative method for sealing the thin wafers, by applying a dense thermal spray coating on the outer surfaces of the wafers. This dense coating is afterward machined into straight rectangular fins as it can be seen in Figure 1.4.



**Figure 1.4: Successive steps to manufacture WMHE: (a) Stacked wire mesh (b) Deposited thermal spray coating (c) Machined external fins**

The coating of the outer surfaces is deposited with a thermal and a kinetic spray process. As seen in Figure 1.5, the first layer of deposition is made with an electrical arc spray process and the second layer is produced with a kinetic spray process. The first layer acts as a bond coat between the wire mesh while the second layer is produced in order to create an extended heat transfer surface by machining fins into this layer. Two thermal spray processes are used since the deposition of a layer of Cold Gas Dynamic Spray (CGDS or simply cold spray) on the wire mesh directly is difficult to accomplish. Also, it is problematic to build a thick coating with the electrical arc spray process. Furthermore, the presence of oxides in the arc spray coating makes it difficult to machine fins inside the coating. Consequently, a thick layer of coating is produced with the CGDS process in order to machine the fins on an arc sprayed bond coat. Therefore, according to the three facts above, it is necessary to have two types of coating on the WMHE.



**Figure 1.5: Schematic of a cross-section of the WMHE presently manufactured by BEC**

The electrical arc spray deposition process uses molten materials that are deposited to form a coating on a substrate. Instead of using the thermal energy to deposit particles, kinetic spray processes use the velocity of the particles. The particles stay in solid-state; however they are usually sprayed at higher velocity. In the latter spray process, the particles are plastically



deformed when they hit the substrate [13]. Different types of material were successfully deposited on a substrate with kinetic and thermal processes.

## **1.2 Motivation of Research and General Objectives**

The present study was motivated by the potential of the kinetic spray process to produce the external fins of the recuperators by additive manufacturing with little or no machining. The machining step in the manufacturing processes of the recuperators represents a significant portion of production costs, according to BEC. For that reason, if a manufacturing method without the machining step is feasible, the DPG will be more a competitive product. It should be noted that the fins studied could also be used as external fins of a WMHE intercooler. The principle is the same as for recuperators, however, the state points will not be comparable, in particular the application temperature, which is much lower for intercoolers.

The current most popular method to deposit a coating using kinetic spray methods is CGDS. Therefore, the primary objective of this thesis is to develop viable and performant near-net shape fins arrays using CGDS in order to improve the manufacturing of BEC's WMHE. In order to achieve the desired objective, several studies are required to be performed:

- 1) Exploration of the potential of CGDS to successfully produce near-net shape fins.
- 2) Investigation of cold spray parameters by means of coating characterization.
- 3) Evaluation of the thermal and hydrodynamic performance:
  - a. Comparison with traditional rectangular fins
  - b. Spray angle effect
  - c. Fin density effect
  - d. Height effect
- 4) Examination of the fins adhesion strength.

- 5) Exploration of the potential of producing a heat exchanger with several materials as external fins (multi-material fin arrays):
  - a. Comparison of thermal and hydrodynamic performance with a single material heat exchanger
  - b. Adhesion strength of each materials
- 6) Understand some experimental results using a numerical model.

### **1.3 Thesis Outline**

The content of the present thesis has been organized into six chapters. Chapter 1 provides an introduction to the relevant background for the research project as well as the motivation and the general objectives for this study.

Chapter 2 presents a detailed review of the relevant theory for this project. A review of heat exchangers, existing fin array configurations and heat transfer/fluid mechanics theory useful to the study is provided. Also, an overview of the type of deposition techniques is presented followed by a detailed description of the deposition technique used in this research, the CGDS process. Finally, recent advances in additive manufacturing techniques are covered.

Chapter 3 describes in detail the specific objectives of this study in order to provide the reader with a clear idea and sense of direction for the development of near-net shaped fin arrays manufactured by CGDS for compact heat exchangers.

Chapter 4 provides a detailed description of the experimental and numerical research approach utilized in this thesis in order to fulfill the objectives mentioned in the third chapter. An overview of the CGDS experimental apparatus is presented, followed by a description of the experimental test apparatus used in order to find the thermal and hydrodynamic performance. Moreover, the equipment employed to characterize the fin arrays and to

determine the mechanical properties are also presented. Finally, a description of the numerical model developed to simulate impacts of cold sprayed particles on a substrate is provided.

Chapter 5 presents the experimental results and their discussion in the form of papers published in peer-reviewed journals. Six papers result from the work in this thesis, each addressing different objectives detailed in the third chapter. A brief introduction of the content is also provided before each paper.

Finally, the last chapter (Chapter 6) summarizes the major results of this work, as well as providing several suggestions for future work on the near-net shaped pyramidal fin arrays produced with the CGDS as an additive manufacturing technique.

## **Chapter 2 - Review of Relevant Literature**

In this chapter, a detailed review of the relevant literature for this study is provided. First, a review of heat exchangers and the associated heat transfer/fluid mechanics concepts are presented, as well as a review of the existing research on fin arrays and their manufacturing methods. Second, an overview of the thermal and kinetic spray processes are presented, with a detailed examination of the main coating technique used for this research, CGDS. Finally, a brief introduction on additive manufacturing as well as on the previous researches that use thermal spray processes and masking techniques are presented.

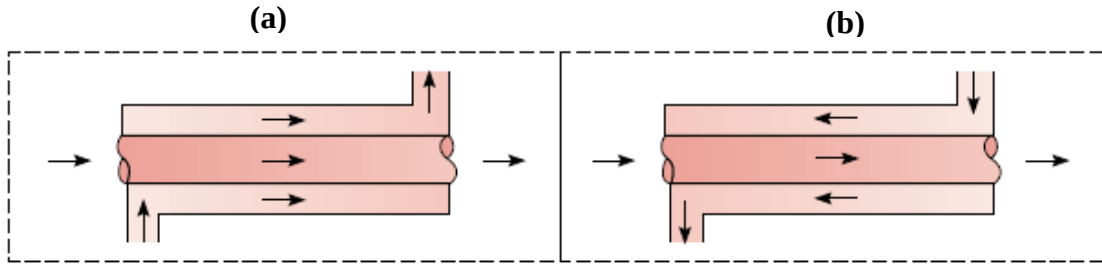
### **2.1 Heat Exchanger**

The current section will present a review of heat exchangers. More specifically, information will be provided on the types of configuration of a heat exchanger, advancement in the fin array development and production as well as the materials typically used in heat exchangers.

#### **2.1.1 Heat Exchanger Overview**

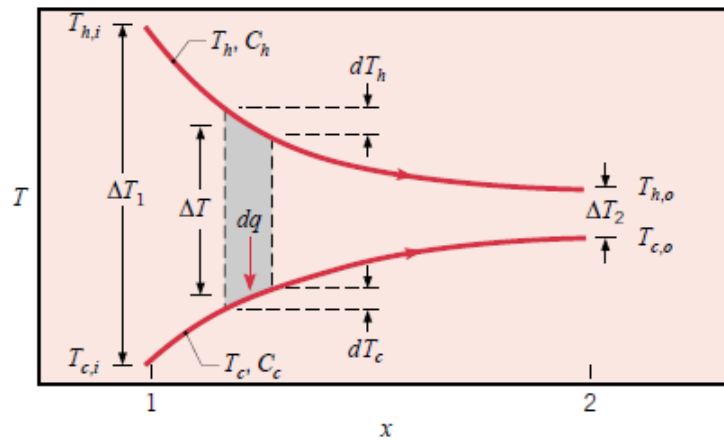
The important engineering parameters for a compact heat exchanger are the heat transfer, the pressure loss and the size. When designing a compact heat exchanger, the heat transfer needs to be maximized while the head loss and the dimensions need to be kept as small as possible.

The heat exchanger flow arrangement configuration could be parallel-flow, counter-flow or cross-flow. In a parallel-flow configuration, both fluids enter at the same end of the heat exchanger and both exit at the opposite end as it is shown in Figure 2.1 (a).



**Figure 2.1: Popular flow arrangement: (a) Parallel-flow (b) Counter-flow [14]**

Consequently, the hot and cold fluids flow in the same direction. This configuration creates a greater temperature difference at the inlet and a smaller temperature difference at the outlet. The temperature gradient decreases quickly along the length of the heat exchanger due to the configuration (Figure 2.2).



**Figure 2.2: Temperature profile for a parallel-flow heat exchanger [14]**

In a counter-flow arrangement the fluids enter at opposite ends and flow in opposite directions, as shown in Figure 2.1 (b). Consequently, the temperature difference will be close to constant along the length as shown in Figure 2.3. The counter-flow heat exchanger configuration is more effective than the parallel-flow configuration due to the more efficient exchange of thermal energy [14]. As such, this type of configuration needs less surface area to exchange the same amount of thermal energy. Consequently, a heat exchanger using a

counter-flow arrangement is more viable than the parallel-flow arrangement in order to manufacture a compact heat exchanger with the highest amount of heat transfer possible.

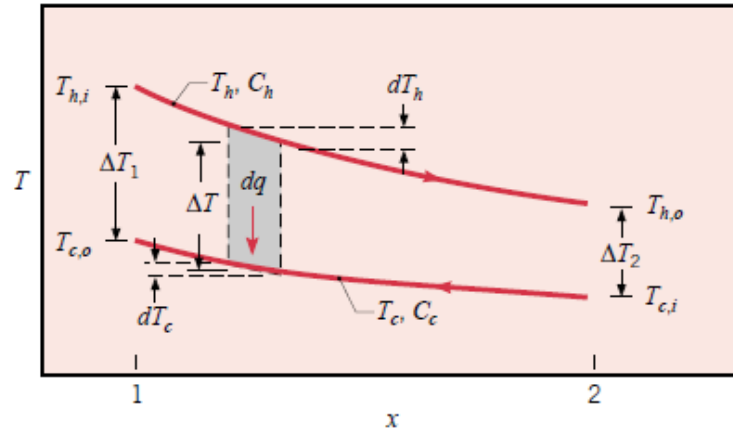


Figure 2.3: Temperature profile for a counter-flow heat exchanger [14]

A third type of flow arrangement to exchange thermal energy between two fluids is the cross-flow configuration. In this configuration, the two fluids flow perpendicularly as shown in Figure 2.4. This type of flow arrangement is not the most efficient but it is used when the manufacturability is a concern since this type of flow arrangement is easy to produce [15].

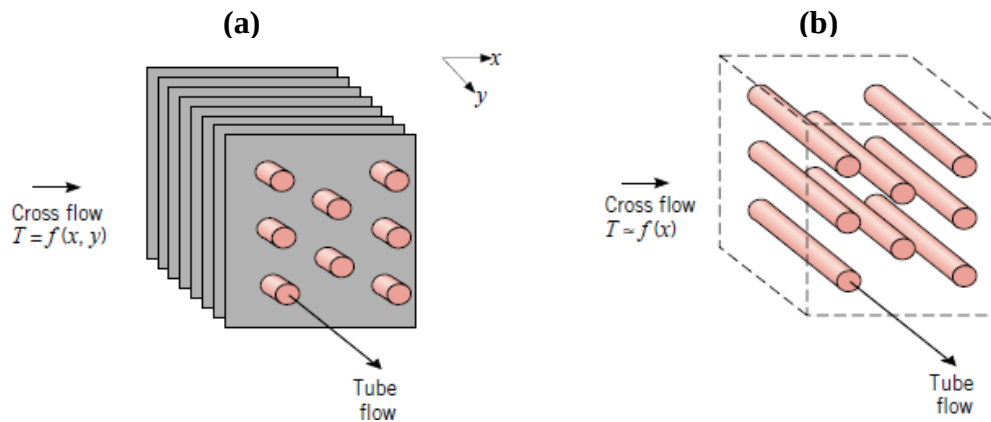
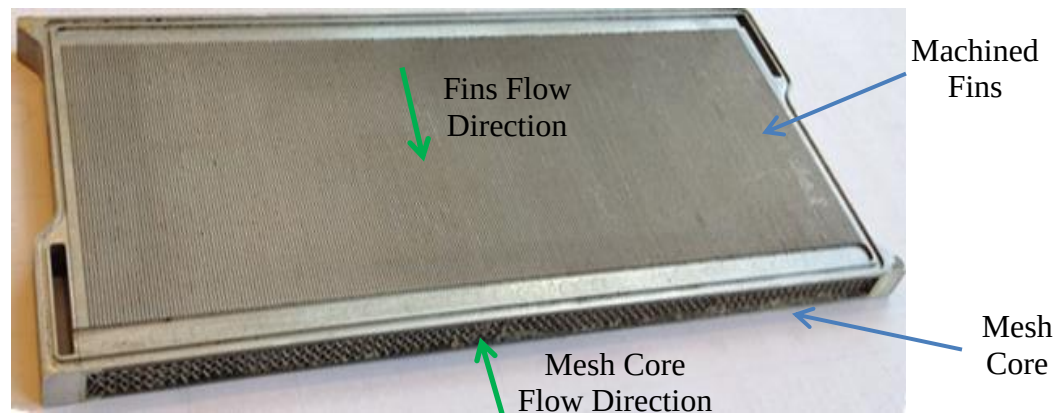


Figure 2.4: Cross-flow arrangement heat exchanger: (a) Finned (b) Unfinned [14]

BEC's WMHE has a counter-flow arrangement equipped with machined fins as shown in Figure 2.5. The latest figure represents one unit cell whereas in the real application there will be several unit cells stacked (fin to fin) in order to produce the whole heat exchanger.



**Figure 2.5: Prototype of one WMHE cell produced by BEC**

As previously mentioned the prototype is built using machined straight rectangular fins. Despite their lower thermal performance compared to other types of fin array, this type of fin remains popular due to their relatively easy manufacturability at low fin density [16]. However, as the fin density increases the manufacturability of this type of fin arrays becomes more difficult due to the associated tolerance [17]. Another drawback of producing high density machined straight rectangular fins is the production speed decreases with tool size [17].

## 2.1.2 Modeling of BEC's Heat Exchanger

The following section gives the important equations to be able to represent mathematically the BEC's Heat Exchanger and to understand the important parameters of a heat exchanger.

### 2.1.2.1 Heat Exchanger Modeled as Thermal Resistance

A widely utilized method to model a heat exchanger is to use the thermal resistance analogy. The principle is comparable to the electrical circuits where there is resistance to the current. The assumptions made to use this model are that heat transfer occurs through one dimension and steady-state conduction [14]. Figure 2.6 shows typical thermal resistances for a heat exchanger manufactured by BEC.

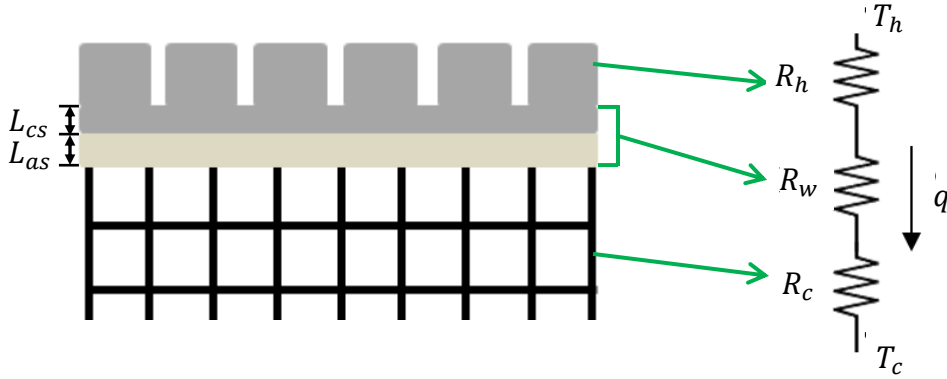


Figure 2.6: Typical thermal resistances for a heat exchanger with a schematic of the BEC's WMHE

where  $R_h$  and  $R_c$  are the convective thermal resistance for the hot and cold side respectively,  $R_w$  is the conduction thermal resistance due to the wall between the two fluids,  $q$  is the heat rate while  $T_h$  is the temperature at the hot side and  $T_c$  at the cold side.

The thermal resistance can be added together the same way as in an electrical circuit. In order to find the total thermal resistance, the following equation can be used:

$$\sum R = R_h + R_w + R_c \quad (2.1)$$



The thermal resistance for conduction in a plane wall is defined as:

$$R_{cond} = \frac{L}{k_m A_w} \quad (2.2)$$

where  $L$  is the wall thickness,  $k_m$  is the thermal conductivity and  $A_w$  is the area of the surface where there is thermal energy transfer for this resistance.

For the particular case presented in Figure 2.6 (the currently manufactured BEC's WMHE), there are two layers of coating and therefore, the conduction resistance ( $R_w$ ) is expressed as:

$$R_w = \frac{L_{as}}{k_{mas} A_w} + \frac{L_{cs}}{k_{mcs} A_w} + R_{t,c} \quad (2.3)$$

where  $L_{as}$  is the wall thickness for the arc sprayed bond coat,  $L_{cs}$  is the wall thickness for the cold spray coating before the base of the machined fins,  $k_{mas}$  is the thermal conductivity of the arc sprayed bond coat,  $k_{mcs}$  is the thermal conductivity of the cold spray coating and  $R_{t,c}$  is the thermal contact resistance between the two layers.

While the thermal resistance for convection (in this case  $R_h$  and  $R_c$ ) are defined as:

$$R_{conv} = \frac{1}{\eta_o h A} \quad (2.4)$$

In the last equation, the variable  $h$  and  $\eta_o$  are respectively the convective heat transfer coefficient and the overall surface efficiency.  $A$  is the area of the surface where there is thermal energy transfer for this resistance.

The heat rate can be described as:

$$q = \frac{\Delta T}{\sum R} = \frac{(T_h - T_c)}{\sum R} \quad (2.5)$$

where  $\Delta T$  is the overall temperature difference.

A useful variable for the heat exchanger is the overall heat transfer coefficient ( $U$ ). A relation exists between the overall heat transfer coefficient and the total thermal resistance, which is:

$$UA = \frac{1}{\sum R} \quad (2.6)$$

where  $UA$  is thermal conductance.

#### **2.1.2.2 Convective Heat Transfer Coefficient**

The values of the convective heat transfer coefficient associated with  $R_c$  and  $R_h$  are determined using two different approaches. Since this study is directed toward the external flow because the near-net shape fins are on the external surface of the heat exchanger and the calculation to find the convective heat transfer coefficient for the internal flow is complex and tedious, the development for the internal flow equations is not done in this work. This demonstration is performed in Corbeil's thesis [11], and is suggested for further reading.

The convective heat transfer coefficients for the external flow are found using typical empirical relations determined by simulation or experiment, and depend on the flow conditions and the geometry of the external surface. In this thesis, experimental approach will be preferred. Therefore, an apparatus was designed at the University of Ottawa and the details about this experiment are given in section 4.3.

#### **2.1.2.3 Heat Transfer Performance of Extended Surfaces**

The thermal resistance diagram for an extended surface is shown in Figure 2.7. As it can be seen there is an exchange of heat between the extended surface and the fluid as well as with the base area where there is no fin.

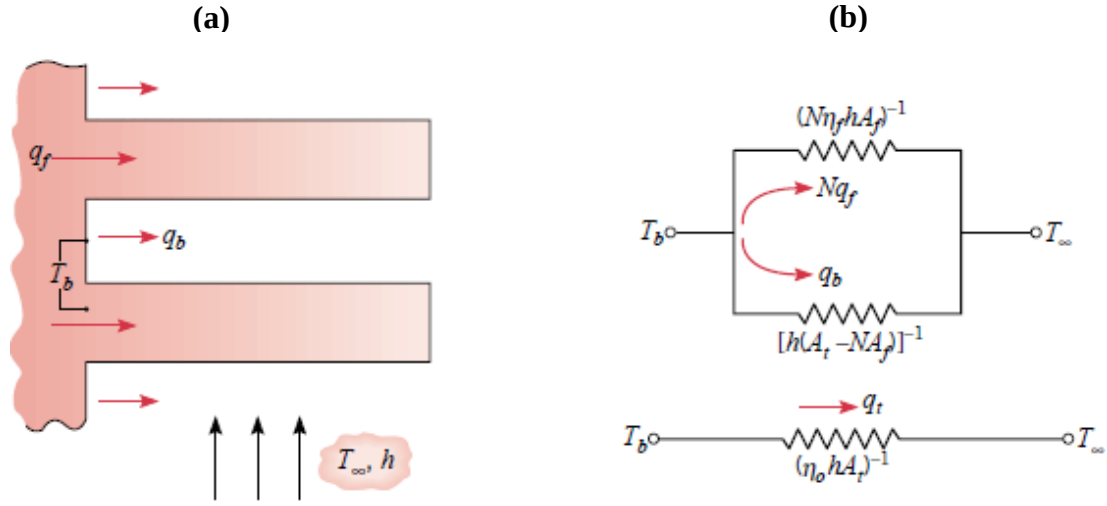


Figure 2.7: (a) Extended surfaces (b) Thermal circuit [14]

The thermal resistance  $R_h$  can then be represented by:

$$R_h = \frac{1}{\eta_o A_t h} \quad (2.7)$$

which includes the heat transfer by the base area and the fins ( $A_t$ ).

The overall surface efficiency is an indicator of the performance of the extended surface and is given by [14]:

$$\eta_o = 1 - \frac{N_f A_f}{A_t} (1 - \eta_f) \quad (2.8)$$

where  $N_f$ ,  $A_f$  and  $\eta_f$  are the number of fin, the area of one fin and individual fin efficiency, respectively.

$A_t$  is given by:

$$A_t = N_f A_f + A_b \quad (2.9)$$

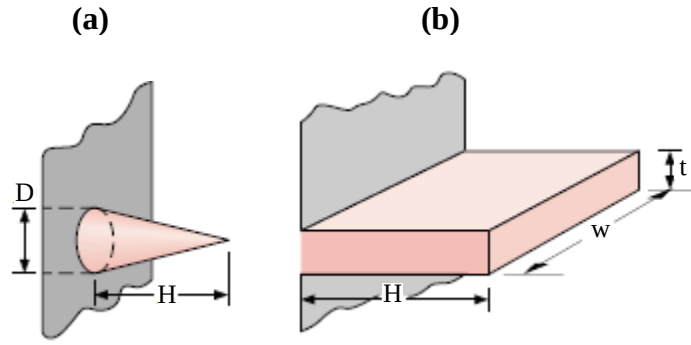
where  $A_b$  is the total base area exposed to the flow.

The individual fin efficiency ( $\eta_f$ ) is defined as the maximum heat transferred by a fin over the maximum amount of heat that can be transferred by the fin:

$$\eta_f = \frac{q_f}{q_{max}} \quad (2.10)$$

where  $q_f$  is defined as the heat transferred by the fin while  $q_{max}$  is the maximum heat that the fin could exchange.

The fin efficiency depends on the fin geometry and the convective heat transfer coefficient. In this study, the geometries encountered are the triangular pin fins and the straight rectangular fins shown in Figure 2.8 (a) and (b), respectively.



**Figure 2.8: Pin fin geometry: (a) Triangular pin fins (b) Rectangular straight fins (adapted from [14])**

The fin efficiency for these two geometries is shown in Table 2.1, based on Incropera [14].

**Table 2.1: Equations associated with different pin fin geometry**

| Geometry                  | Fin Efficiency                                                 |                                            |
|---------------------------|----------------------------------------------------------------|--------------------------------------------|
| Triangular pin fins       | $\eta_f = \frac{2}{mH} \frac{I_2(2mH)}{I_1(2mH)} \quad (2.11)$ | $m = \sqrt{\frac{4h}{k_m D}} \quad (2.12)$ |
| Straight rectangular fins | $\eta_f = \frac{\tanh(mH_c)}{mH_c} \quad (2.13)$               | $m = \sqrt{\frac{2h}{k_m t}} \quad (2.14)$ |

The fin efficiency equation for triangular pin fins are used for the sprayed pin fin arrays in this research work while the definition for rectangular straight fins will be used for the traditional straight cut utilized for comparison purposes. The expressions  $I_1$  and  $I_2$  in Table 2.1 are the

Bessel functions of order 1 and 2, respectively. At this point the thermal resistance diagram is fully defined and the heat transfer through the heat exchanger could be found by:

$$UA = \left[ \left( \frac{1}{\eta_o h A} \right)_h + \left( \frac{L_{as}}{k_{mas} A} + \frac{L_{cs}}{k_{mcs} A} + R_{t,c} \right)_w + \left( \frac{1}{\eta_o h A} \right)_c \right]^{-1} \quad (2.15)$$

#### 2.1.2.4 Heat Exchanger Performance

A technique to determine a heat exchanger performance is using the  $\varepsilon$ - $NTU$  method. It should be noted that other techniques exist but they are not shown in this thesis. This method is based on the effectiveness of the heat exchanger ( $\varepsilon$ ) which is only dependent on the  $NTU$  (Number of Transfer Units), the fluids' thermal capacity ratio ( $C_r$ ) and the heat exchanger configuration. The effectiveness is a parameter to measure the performance of a heat exchanger which is the heat transfer rate done by the heat exchanger over the maximal possible heat transfer rate between the two fluids, if perfect heat transfer is achieved:

$$\varepsilon = \frac{q}{q_{max}} \quad (2.16)$$

The capacity ratio is given by:

$$C_r = \frac{C_{min}}{C_{max}} = \frac{(\dot{M} c_{p_{fl}})_{min}}{(\dot{M} c_{p_{fl}})_{max}} \quad (2.17)$$

where  $C_{min}$  is the minimal value for the heat capacity of the fluids,  $C_{max}$  is the maximal value for the heat capacity of the fluids,  $\dot{M}$  is the mass flow rate and  $c_{p_{fl}}$  is the specific heat capacity. For this study, the type of heat exchanger is a counter-flow and therefore, the  $\varepsilon$  -  $NTU$  equation for that type of configuration is given below [14]:

$$\varepsilon = \frac{1 - \exp[-NTU \cdot (1 + C_r)]}{1 - C_r \cdot \exp[-NTU \cdot (1 + C_r)]} \quad (2.18)$$

As it is mentioned in Incropera work [14], as the  $NTU$  and/or the  $C_r$  rise, the  $\varepsilon$  increase for this type of configuration.

The  $NTU$  is defined as:

$$NTU = \frac{UA}{C_{min}} \quad (2.19)$$

Therefore, since a higher  $NTU$  means a better performance of the heat exchanger, the thermal conductance ( $UA$ ) is an important parameter to monitor and it is defined with the equation 2.15 for this study. Consequently, an increase in heat transfer area, convective heat transfer coefficient, overall surface efficiency and/or thermal conductivity of the wall will mean an enhancement in performance of the heat exchanger whereas a thicker wall and an higher thermal contact resistance will decrease the effectiveness.

#### 2.1.2.5 Fluid Mechanics Associated with Fin Array

The fluid mechanics aspect is also a main part of the design of the fin array. Figure 2.9 shows the different types of pressure change for a typical heat exchanger.

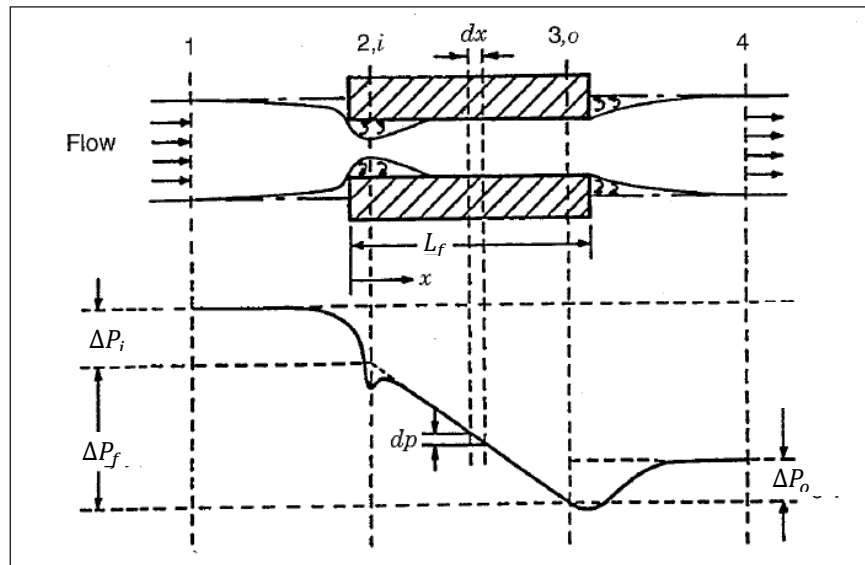


Figure 2.9: Different pressure change across a typical heat exchanger (adapted from [18])

The total pressure drop across a fin array is then determined by the following equation:

$$\Delta P = \Delta P_i + \Delta P_f + \Delta P_o \quad (2.20)$$

The first term ( $\Delta P_i$ ) is the pressure drop due to the contraction of the flow at the entrance of the fin array while the last term ( $\Delta P_o$ ) is due to the expansion of the flow at the outlet. The  $\Delta P_i$  is given by the following equation:

$$\Delta P_i = K_c \frac{\rho_{fl} \bar{u}^2}{2} \quad (2.21)$$

where  $\rho_{fl}$  is the fluid's density,  $K_c$  is the contraction coefficient and  $\bar{u}$  is the mean velocity.

Similarly, the  $\Delta P_o$  is given by the following equation:

$$\Delta P_o = K_e \frac{\rho_{fl} \bar{u}^2}{2} \quad (2.22)$$

where  $K_e$  is the expansion coefficient. It should be noted that the value of  $K_c$  and  $K_e$  are usually determined experimentally.

The  $\Delta P_f$  term is explained by the fact that inside the array, there are frictional losses for the fins as well as flow modifications, such as flow separation. The frictional pressure drop through a longitudinal continuous surface, for example straight rectangular fins, could be expressed using the Darcy-Weisbach equation as follows [15]:

$$\Delta P_f = f \cdot \frac{L_f}{D_h} \cdot \frac{\rho_{fl} \bar{u}^2}{2} \quad (2.23)$$

where  $f$  is the Darcy friction factor,  $L_f$  is the length of the heat exchanger,  $\rho_{fl}$  is the fluid's density and  $D_h$  is the hydraulic diameter.

More specifically, the frictional pressure loss for non-continuous surfaces such as fin arrays is given by [15]:

$$\Delta P_f = f \cdot L_f \cdot FD_L \cdot \frac{\rho_{fl} u_{max}^2}{2} \quad (2.24)$$

where  $FD_L$  is the fin density in the streamwise direction and  $u_{max}$  is the maximum velocity in the fin array.

## **2.1.3 Advancement in Fin Arrays**

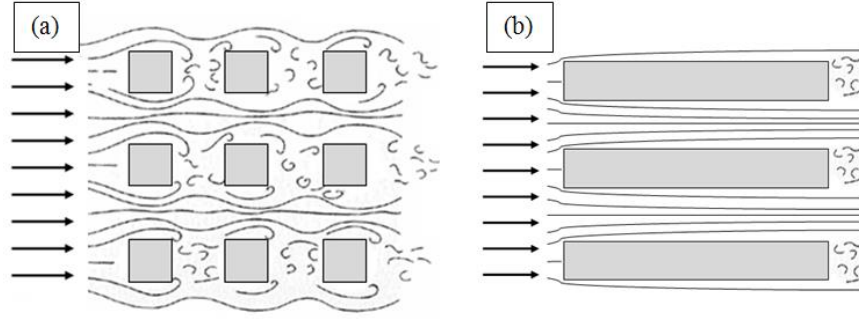
### **2.1.3.1 Kays and London**

Kays and London have studied different types of fins for compact heat exchangers, with their results first published in 1958 [15]. They studied types of fins varying from the traditional rectangular straight fins to more complex fin geometries. Most types of fins studied by Kays and London need extensive machining processes, which can be tedious and expensive.

### **2.1.3.2 Pin Fins**

The development of discontinuous (pin) fin arrays (Figure 2.10 (a)) has been inspired by the research work regarding tube banks in cross-flow, accomplished in the 1970's by Žukauskas [19]. His major contribution is the discovery of turbulence in the wake of tubes. After the publication of these results, the development of pin fins has become a major study subject since using discontinuous features instead of continuous features, such as those traditionally used, increases the mixing because of the beneficial turbulence behind the fins, as shown in Figure 2.10. This turbulence promotes a higher heat transfer rate, at the expense of moderately higher pressure losses.





**Figure 2.10: Top view for: (a) Discontinuous fin arrays (b) Straight cut fin arrays**

In Žukauskas' work, three heat transfer regimes are found for discontinuous features when varying the Reynolds number: laminar, sub-critical and critical flow regimes [19]. The first regime occurs from  $Re_D = 0$  to about  $Re_D = 1000$ , where the fluid flows around the tube, with its velocity low enough to stay attached to the fin surface. When the transition occurs in the sub-critical regime, the flow still enters in a laminar fashion, but flow separation occurs along the fin's length. This creates a turbulent wake behind the fin, increasing fluid mixing and consequently the heat transfer rate for  $Re_D$  numbers between 500 and 200 000. The overlap in flow regimes is due to the transition between the laminar and sub-critical regimes. The critical flow regime consists of a turbulence dominated regime, which is found at  $Re_D$  larger than 200 000. Also, Žukauskas stated that within the domain of a regime, the correlations between Nusselt number and Reynolds number are linear and therefore, can be extrapolated linearly [19]. The same regimes found by Žukauskas in banks of tubes were observed in pin fin arrays by several scientists [20]. Even if the head losses are higher in discontinuous fin arrays, the improvement in heat transfer overcomes this disadvantage [20].

The beginnings of this discontinuous fin array research area of heat transfer can be traced to the study of adding circular pin fins to the cooling system of gas turbine blades during the 80's. At that time, the study of pin fins was restricted to those with circular cross-sections in

the inline and staggered configurations. Numerous researchers found interesting results, which have established the basis of understanding for pin fin arrays [21- 23]. Brigham and Van Fossen [21] studied the relative height parameter ( $H/D$ , where  $H$  is the fin height and  $D$  is the pin fin diameter). Their results show that longer pin fin ( $H/D > 4$ ) tend to have a better heat transfer than shorter pin fin ( $H/D < 4$ ). Sparrow et al. [22] and Metzger et al. [23] established experimentally that circular pin fin arrays have a higher heat transfer coefficient than the traditional continuous straight rectangular fin array.

Numerous new cross-sections for pin fin arrays could be potentially produced due to the improvements of manufacturing technologies in the 90's. Square pin fin arrays were studied experimentally by few authors [24, 25]. Chapman et al. [24] originally tested experimentally the square pin fin arrays. Jeng et al. [25] found the influence of the transverse and the longitudinal dimensionless pitch on the thermal and hydraulic performance of pin fin with square cross-section for in-line and staggered configurations. The influence of this parameter could be applied to the other type of cross-section in a similar way. Jeng and coworkers found an optimal inter-fin pitch by finding the largest Nusselt number at a given pumping power [25]. Meanwhile, Grannis and Sparrow [26] explored the possibility of using diamond shaped pin fin by providing correlation between the friction factor and the Reynolds number. Their research was motivated by the use of this type of pin fin array in a heat exchanger utilized in the space shuttle Atlantis. Later, Tanda et al. [27] and Chyu et al. [28] studied experimentally diamond shaped pin fins. They found that the diamond cross-section fin arrays has a higher pressure drop and lower heat transfer compared to the square pin fin arrays which means the latter is better than the diamond cross-section pin fins. Several other cross-sections were encountered during the literature survey such as elliptic [24, 29, 30], drop shaped [31] and

oblong shaped [23]. The main goal of these studies was to find the cross-section which has the highest heat transfer while keeping a reasonable head loss. To this end, several scientists did simulations instead of experiments in order to find the cross-section shape that offers the best performance even if the manufacturing method to obtain that cross-section does not exist yet. Sahiti et al. [20] and Zhou et al. [32] did several numerical simulations of various cross-sections: circular, drop, elliptic, NACA (airfoil), lancet and square. Their results shows that all cross-sections tested have approximately the same efficiency, except square that is less efficient. However, no literature was encountered on pin fin arrays that have a varying cross-section along the fin height, probably due to the difficulty of manufacturing such arrays.

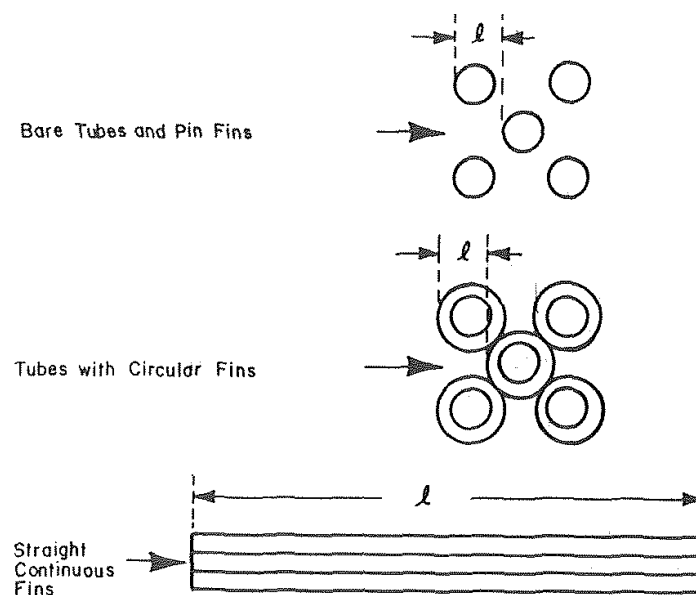
It should be noted that usually the authors give an equation for the heat transfer and an equation for the pressure losses for the experimental conditions they have tested. The equation for heat transfer is usually in terms of the dimensionless Nusselt as a function of Reynolds number while the equation for the head loss is in terms of the dimensionless friction factor also as a function of the Reynolds number.

#### **2.1.3.3 Efficiency Index**

As it is possible to see through a review of the work published in this area, usually the papers show the thermal performance and the hydraulic performance independently, by giving one equation for each performance, related to the Reynolds number. However, in an industrial application, it is important to know if it is better to increase the thermal performance in spite of the associated cost of higher pressure losses and to know at which point it is not efficient to increase the thermal performance due to the increased cost in pressure drop. Since the beginnings of the study of pin fin arrays and compact heat exchangers, several authors have

shown interest in finding an efficiency parameter that compares fin arrays in terms of their thermo-hydraulic performance.

The earliest comparison method found in the literature survey is a direct use of Colburn factor in 1942 [33]. The Colburn factor is a dimensionless number which links the heat, the momentum and the mass transfer. In 1949, London and Ferguson [34] suggested a modified Colburn factor by plotting the heat transfer coefficient versus the flow friction power supplied normalized by the total heat transfer area. In this method, when the heat transfer coefficient is better for a given normalized pumping power, the fin array is more efficient. After several years of using this method, LaHaye et al. [35] proposed a different approach. Instead of using experimental data such as heat transfer coefficients, they calculated the ratio of the flow length between two the major boundary layer disturbances to the hydraulic diameter. Figure 2.11 shows several flow lengths between the major boundary layer disturbances for different types of fin.



**Figure 2.11: Example of the flow length ( $l$ ) between the major boundary layer disturbances for several types of fin [35]**

They proposed that the lower this ratio; the higher is the heat transfer. The major drawbacks of this technique are that it does not take the influence of fin efficiency, fin thickness, volume occupied by the fin array and it is only valid for the turbulent regime [35]. Furthermore, these earlier methods of comparison do not take into account the effect of the driving temperature difference on the performance [36]. Therefore, these methods can only be used when the fin arrays to be compared have the same temperature difference between the fins and the fluid, which is not practical.

In order to take into account the influence of this parameter, Soland et al. [37] introduced the comparison using the  $NTU$ . They compared the  $NTU$  per unit volume versus the power input required to move the flow across the fin array per unit volume. Meanwhile, Shah [38] developed a method of comparison using a volume goodness factor. This technique compares the heat transfer per unit volume and the temperature difference versus the power required by unit volume. Lately, Tian et al. [6] suggested a method of comparison by plotting the ratio of Nusselt divided by the friction factor at the exponent  $1/3$  versus the Reynolds number as shown in Figure 2.12.

For a given Reynolds number, the higher the ratio is, better is the efficiency of the fin array [6]. Another interesting method was introduced by Chang et al. [39] in 2008. Their method of comparing pin fins is by the performance ratio demonstrated in the following equation:

$$\eta = \frac{\frac{Nu}{Nu_{smooth}}}{\left(\frac{f}{f_{smooth}}\right)^{1/3}} \quad (2.25)$$

where  $Nu$  is the Nusselt number and  $f$  is the friction factor. The subscript smooth means the properties for a smooth, unfinned equivalent channel.

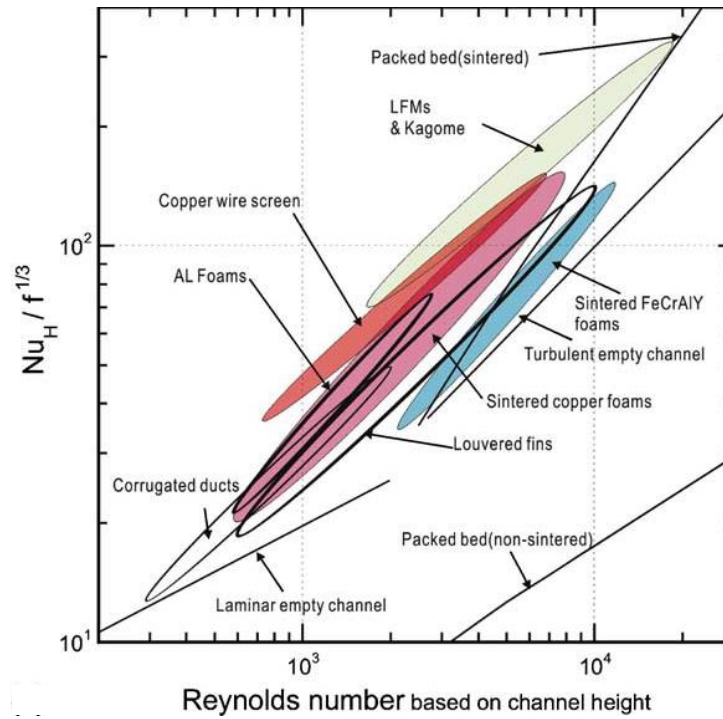


Figure 2.12: Comparison graph from Tian et al. method [6]

Therefore, the latter method finds the increase in heat transfer compared to an equivalent smooth channel and then divides this ratio by the ratio of increase in friction factor at the exponent 1/3. As such, for a given Reynolds number, a higher performance ratio means more efficient pin fin array as represented in the following figure:

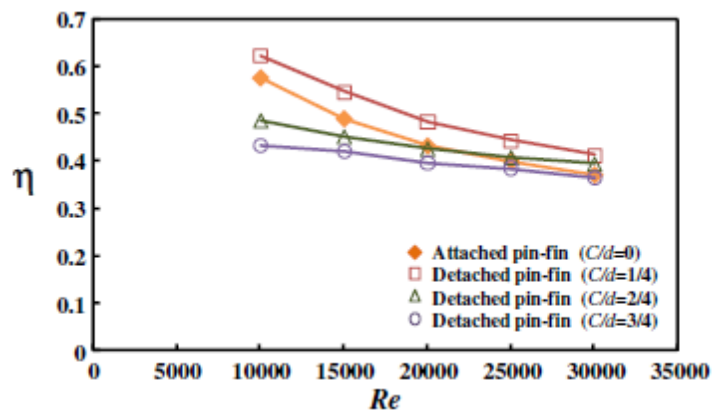


Figure 2.13: Comparison graph from Chang et al. method [39]

This method is close to the technique proposed by Tian et al. but they added the comparison factor to an equivalent smooth channel. However, the methods proposed by Tian et al. and Chang et al. do not take into consideration the volume of the fin array or the footprint area [6, 39]. Another difficulty with the methods proposed above is the comparison with literature since the values obtained are sample geometry dependent and the values required to be adapted for comparison are often not provided [36].

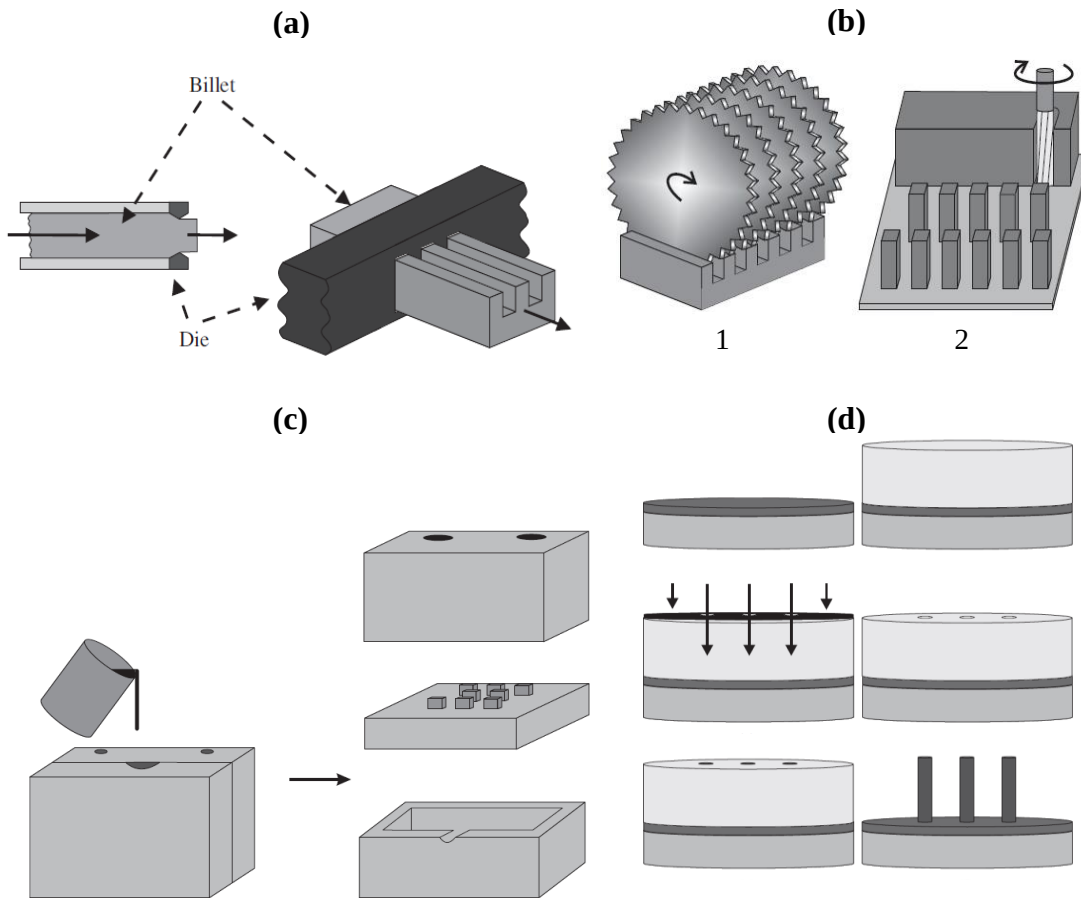
More recently, Sahiti et al. [40] demonstrated that the most accurate method to compare fin arrays is by a direct comparison of heat transfer rate and power input required per unit volume or per unit of footprint area depending on the application. To this end, a plot of the thermal and hydrodynamic data as a curve of thermal conductance per unit of fin array volume as a function of the required pumping power per unit volume is used to compare different pin fin cross-sections. The most efficient fin array has a higher thermal conductance at a given pumping power. Furthermore, another advantage of the technique proposed by Sahiti et al. is that this method of comparison allows performing evaluation of fin arrays without the details of the geometry and of the calculations of the surface in comparison. When other techniques are used, this information is needed to be able to do an accurate comparison between two works.

### **2.1.4 Manufacturing Techniques for Micro-Fin Arrays**

In the fin arrays domain, the one BEC currently manufactured is considered as micro-fin arrays due to their height (thousands of microns). Consequently, this subsection presents a review of the manufacturing techniques that exist to produce these kinds of fin.

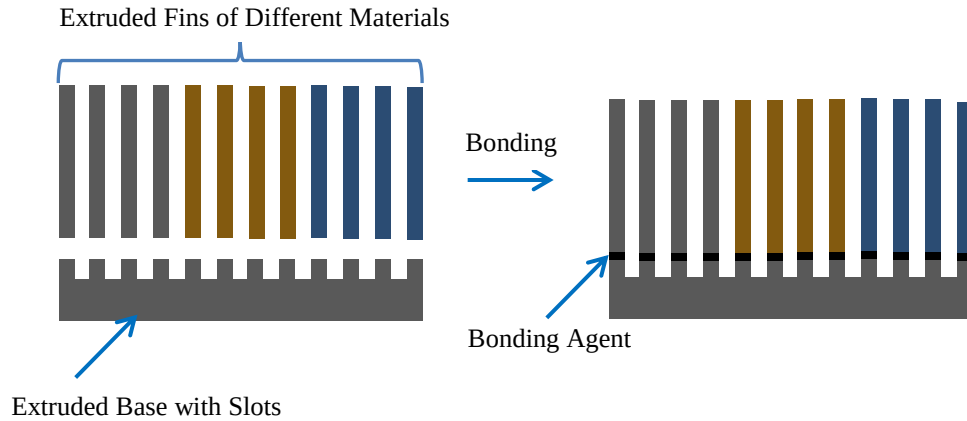
Extrusion (Figure 2.14 (a)) and milling (Figure 2.14 (b)) have typically been used to produce both continuous fin and pin fin arrays, respectively [41]. It should be noted that the technique is used at BEC is the milling technique for continuous fin as shown in Figure 2.14 (b)-1. Casting (Figure 2.14 (c)) has also been used as an alternative production method for these arrays [41]. All these manufacturing techniques suffer from the same drawback of increased cost as the complexity and the density of the fin array increases [41]. The trend in the last decades is to increase the ratio of heat transfer area per unit volume, and as such fin arrays have tended to present higher fin densities (amount of fins per unit base area) [41]. The cost of casting and extrusion increases rapidly due to the precision required for the production of the mold or die [17, 42], while for machining, the production speed decreases with tool size, which is the limiting factor for feature size and the associated tolerances [17, 43]. Metal mold casting is further restrained as the design must accommodate features with taper angles and without concave details so that the part can be removed from the rigid mold after casting [17, 44]. When the fin array features are too small for conventional manufacturing techniques, lithographic techniques (Figure 2.14 (d)) can be used to produce fin arrays but require a significant equipment investment [17].





**Figure 2.14: Manufacturing methods for micro-fin arrays: (a) Extrusion (b) Milling (c) Casting (d) Lithography [17]**

Another limitation of conventional fin array manufacturing methods is that they typically restrict a fin array to sections made of one homogeneous material. These sections can be joined together to create mixed material fin arrays as shown in Figure 2.15, but this rapidly becomes uneconomical or impractical due to reliability issues with the bond between the sections. Additionally, the bonding agent itself represents a thermal barrier for the heat flow. This type of process is usually limited to plain rectangular fin geometries due to manufacturing concerns [41].

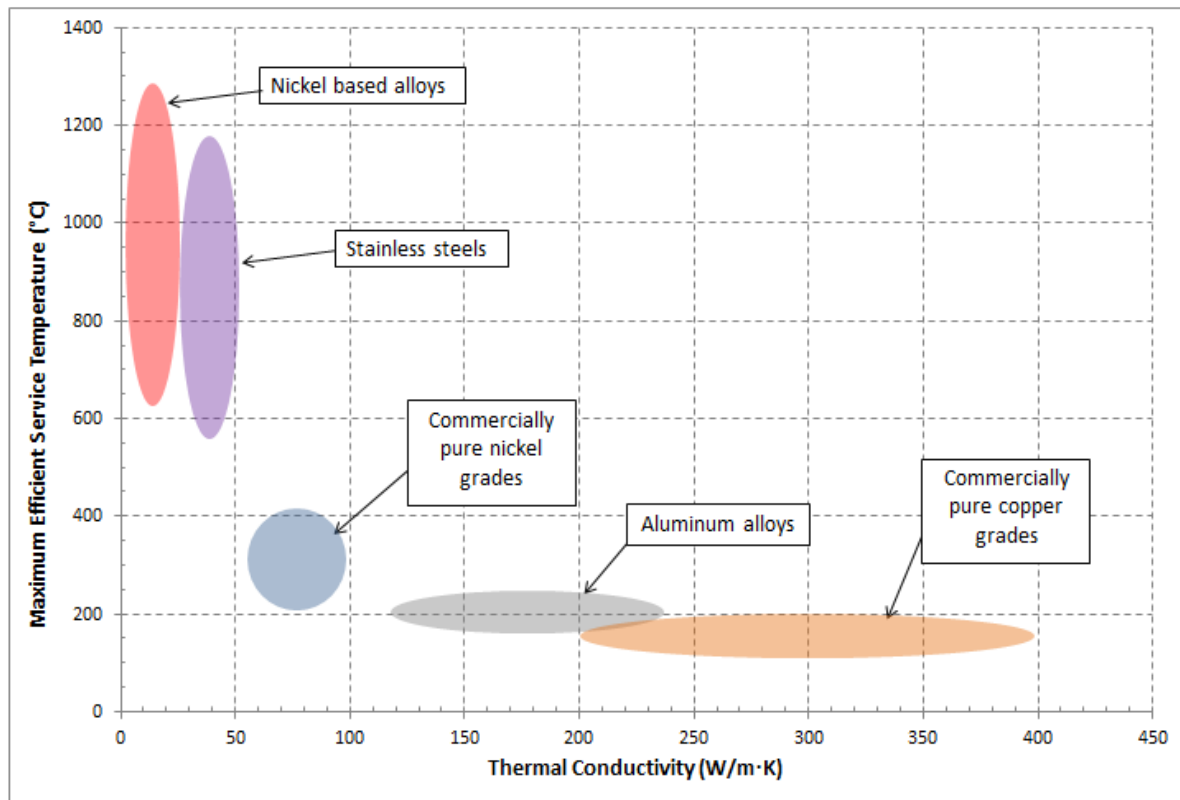


**Figure 2.15: Fin array produced by bonding**

### 2.1.5 Materials in Heat Exchanger

An important parameter to measure how efficient a material is to transfer thermal energy is its thermal conductivity,  $k_m$ . Several materials such as pure aluminum ( $k_m = 237 \text{ W/m}\cdot\text{K}$ ) and pure copper ( $k_m = 401 \text{ W/m}\cdot\text{K}$ ) have a high thermal conductivity making these materials good candidates for heat exchanger applications. However, there are other factors to take into consideration when selecting a material for a heat exchanger, in particular the application temperature. The low melting point of aluminum (933 K) makes it an irrelevant material for heat exchangers in high temperature applications. Even copper that has a relatively high melting point (1358 K), cannot be used in high temperature applications due to the formation of a non-conductive oxide layer at high temperatures [45]. As seen in Figure 2.16, the maximum efficient service temperature of copper is around 200 °C (473 K). For extremely high temperatures, stainless steel and nickel alloys are good choices since their maximum efficient service temperature is higher than the other materials presented in Figure 2.16. Nevertheless, their thermal conductivities are lower than those of copper and aluminum. For intermediate temperatures, commercially pure nickel grades show a compromise between the high maximum efficient service temperature and high thermal conductivity, as shown in

Figure 2.16. Commercially pure nickel grades have a lower maximum efficient service temperature (673 K) than its melting point (1713 K) for the same reason as pure copper. However, the detrimental effect of the oxide layer on this material's thermal conductivity is less important than it is for copper [45].



**Figure 2.16: Maximum efficient service temperature versus thermal conductivity for several materials [adapted from 45]**

Another factor to think about when choosing a material for a heat exchanger is the cost and the availability of the materials. For example, the cost to produce copper fins is higher than aluminum fins due to the price of the feedstock material.

## 2.2 Thermal Spray Processes

Thermal spray deposition processes operate on the principle of using a heat (thermal spray) source combined with an accelerated carrier gas to project molten particles onto a substrate in order to form a dense coating, as depicted in Figure 2.17.

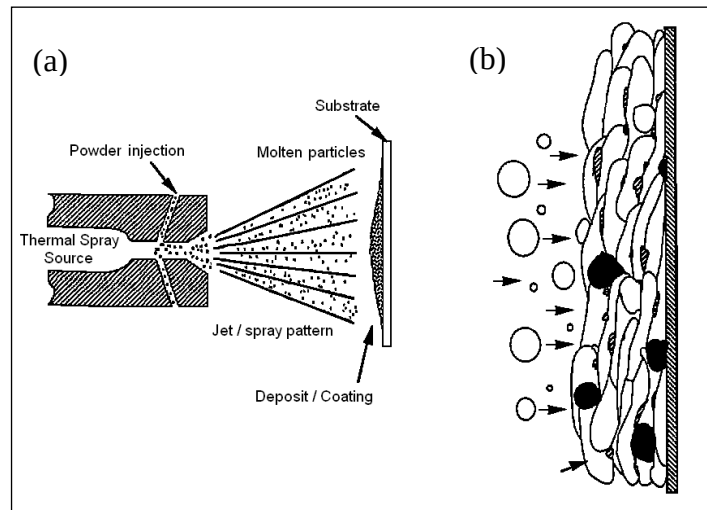


Figure 2.17: (a) Thermal spray process (b) Resulting coating [46]

Thermal spray processes are divided into four groups (plasma spraying, electric arc spraying, combustion spraying and kinetic spraying) depending on the type of heat/energy source used to deposit the particles on a substrate, as shown in Figure 2.18.

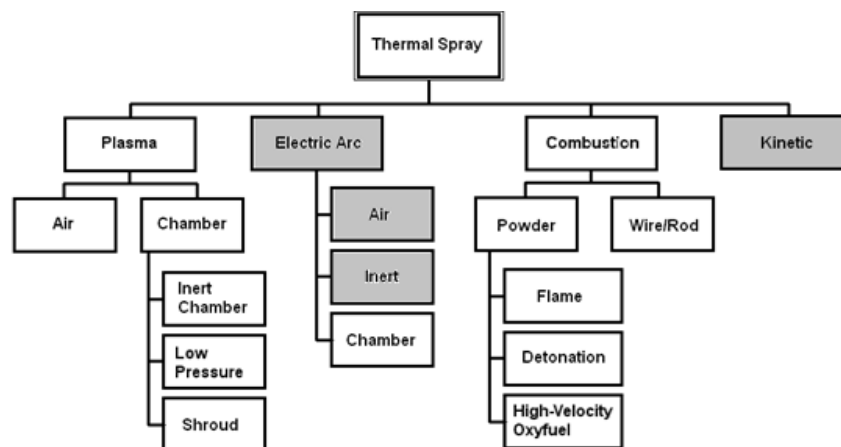


Figure 2.18: Different groups of thermal spray (adapted from [47])

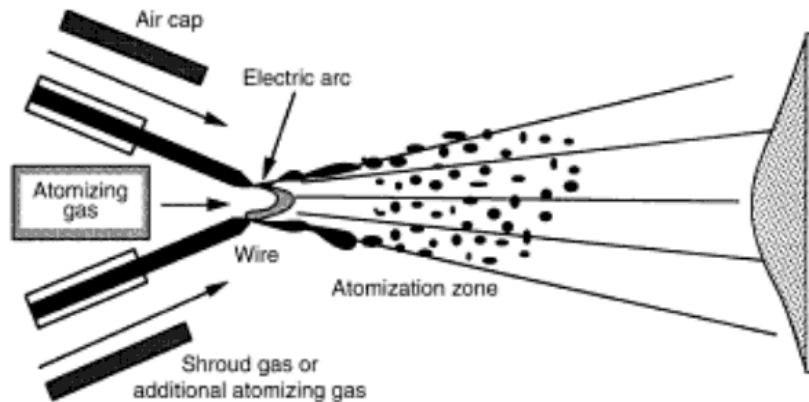
Combustion groups use chemical energy as source to create a high temperature gas stream that melts and carries the particles. The electric arc spray uses the electricity to melt the materials and a carrier gas stream. For the plasma spray, the electricity is used to create a high temperature plasma that melts and carries the particles. The adhesion strength of the coating (deposited particles) onto the substrate (adhesion) and itself (cohesion) for these three groups is sustained by the melted particles solidifying on the substrate surface and onto each other. Kinetic spray processes such as CGDS, rely on momentum transferred from an inert gas, usually helium or nitrogen, to relatively cold particles which are projected onto a substrate's surface. It should be noted that the processes filled in gray in Figure 2.18 are of specific interest to this study as they are thermal spray processes currently used to manufacture BEC's WMHE. As such, the use of these spray processes will be explored in this thesis to improve the fin arrays presently produced at BEC. Thus, more details will be given in the following subsections on the two thermal processes that are going to be used for this study.

### **2.2.1 Electric Arc Spray**

In the industrial application, the fin arrays produced by CGDS will be deposited on a layer produced by electric arc spray. As such, it is important to do a brief review of this coating technique.

This deposition method was first developed by Schoop in the 1910's [47]. Nevertheless, this thermal spray process was barely used until the 1960's. The technique consists of two consumable conductive wires that are brought together, with an electric arc struck between them using a DC current, as represented in Figure 2.19. Consequently, the tip of the junction of the two wires is melted. A gas flow around the tip is used to shear the molten material

resulting in fine, molten, metallic particles. These atomized metallic particles are propelled by the gas onto the substrate, where they deform under the impact force and subsequently solidify to form a coating.



**Figure 2.19: Schematic of a typical electric arc spray nozzle [47]**

The temperature of this thermal spray process is between 3000 °C to 6000 °C [48]. The particle velocity is in the low range of the thermal spray processes, ranging from 50 m/s to 150 m/s [48]. When air is used as carrier gas, there is a high oxide content in the coating since the molten particles react with the oxygen from the air.

As the wires need to be conductive and ductile, this thermal process is generally limited to aluminum alloys, brass, zinc, copper, titanium, nickel and stainless steels. However, non-conductive wires coated with a conductive material were successfully deposited [47].

## **2.2.2 Kinetic Spray Processes**

Since the main part of this research work will be done using kinetic spray, a more detailed review is presented in this subsection. CGDS is a kinetic spray process that was originally developed by a Russian group at the Institute of Theoretical and Applied Mechanics of the Siberian Branch of the Russian Academy of Sciences in Novosibirsk, Russia in the 1980's

[13, 50]. As the name suggests, cold spray operates at temperatures lower than typical thermal spray processes.

Figure 2.20 shows a schematic of typical CGDS equipment. CGDS rely on the transfer of momentum from a high pressure driving gas, usually nitrogen or helium, to particles which are accelerated toward a substrate surface. The primary flow goes through a gas heater that can preheat the gas, below the particle material's melting point, in order to increase the gas velocity, through an increase in the local speed of sound. Afterward, the gas is accelerated to supersonic speeds using a converging-diverging (DeLaval) nozzle. Meanwhile, a secondary high pressure supply is used to push the particles to be sprayed from the powder feeder into the nozzle. When the particles reach the nozzle, they are accelerated by the driving flow. Consequently, when the particles hit the substrate, they have a high amount of kinetic energy. This kinetic energy is converted to plastically deform the particles onto the substrate.

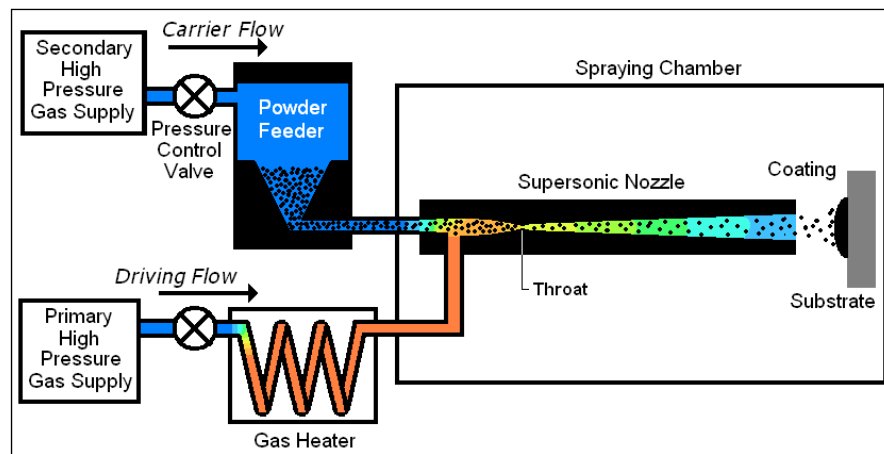


Figure 2.20: Schematic of the typical CGDS process [51]

#### 2.2.2.1 Bonding Mechanisms

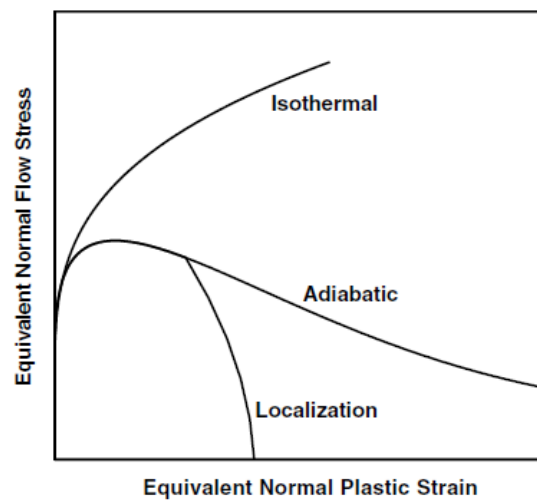
The bonding mechanisms involved in the cold spray process are still a popular topic since they are not fully understood and explained. However, several theories have been proposed

after many experimental and numerical observations were made over the last decade. Two main mechanisms are believed to be encountered during the cold spray deposition: metallurgical bonding and mechanical anchoring.

The first theories published on the cold spray bonding mechanism were numerical investigations made by Assadi et al. [52] and Grujicic et al. [53] in the early 2000's. It was proposed that the process involved in cold spray is similar to other processes involving high speed dynamic deformation, such as explosive welding, where adhesion is created by the formation of adiabatic shear instabilities upon impact. To understand the effect of the adiabatic shear instability, typical dynamic stress-strain curves are depicted in Figure 2.21. The adiabatic shear instability plays a role on how the resistance to flow (flow stress) varies with the plastic strain. As it can be seen, the flow stress increases for a typical work hardening material (isothermal) as the plastic strain rises. Under adiabatic conditions, the flow stress decreases at a certain value of the plastic strain. Indeed, in an adiabatic condition, material softening is engendered by the increase in temperature generated by the heat from the dissipation of the plastic strain energy. The first part of the adiabatic curve in Figure 2.21 shows an increase since at low plastic strain, the energy that generates a rise in temperature is insufficient to overcome the work hardening generated by the plastic strain. However, at a certain point, the temperature generated is sufficient to overcome the work hardening induced by the plastic strain. Thus, the flow stress of the material decreases, which is also called material softening. Therefore, this fact explains the reduction in flow stress of the second part of this adiabatic curve. The word adiabatic is used since the heat generation is so rapid that the heat does not have time to dissipate into the material due the impact time involved in cold spray deformation. Furthermore, since the strain, stress, temperature and inherent instability

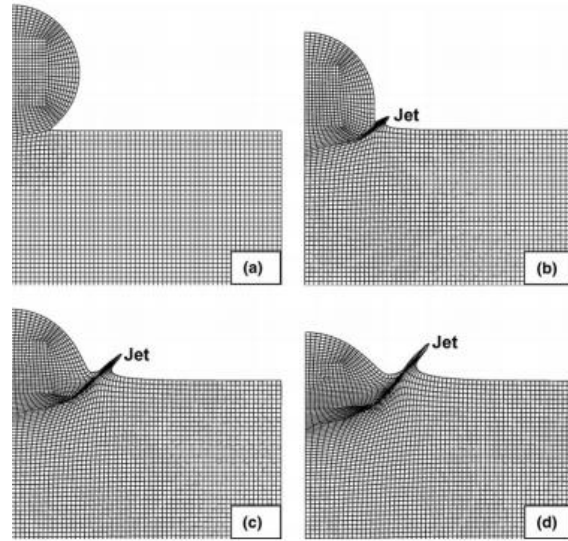


of strain softening are fluctuating within the real materials, the shearing and heating become highly localized and the flow stress quickly drops to zero. If the plastic zone is retained in a localized zone, its effect on the material softening is more pronounced since the heat generation is more concentrated. Furthermore, the high strain rates involved in the cold spray process bring a sufficient amount of plastic energy to be in the softening regime. To sum up, the theory proposed by Assadi et al. [52] and Grujicic et al. [53] is a competition between the work hardening and the material softening in order to have material flowing since the work hardening increases the flow stress whereas the material softening due to the adiabatic and localization phenomena involved in cold spray decreases the flow stress.



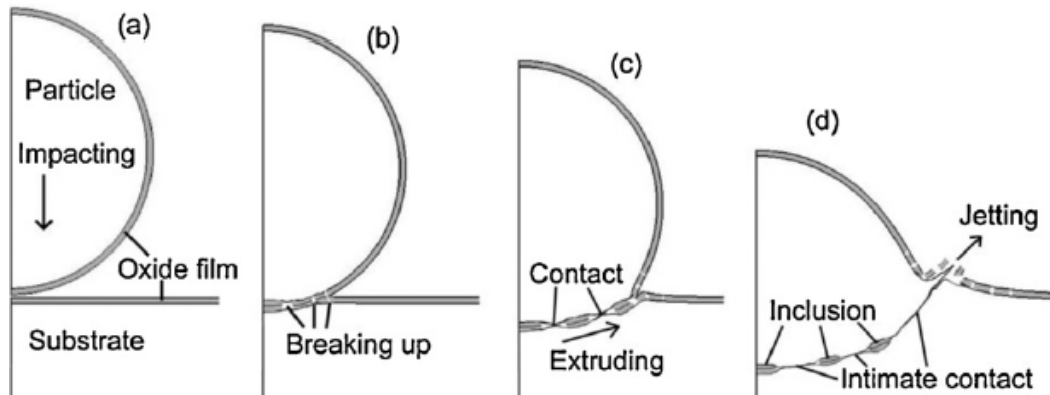
**Figure 2.21: Typical flow stress vs. plastic strain for isothermal, adiabatic and localization conditions [53]**

The combination of the high plastic strain encountered during the cold spray process and the adiabatic shear instability phenomenon result in the material flowing. This material movement is known as jetting, where the material viscously flows outwards from the center of the particle while creating a crater in the substrate, as shown in Figure 2.22 [53].



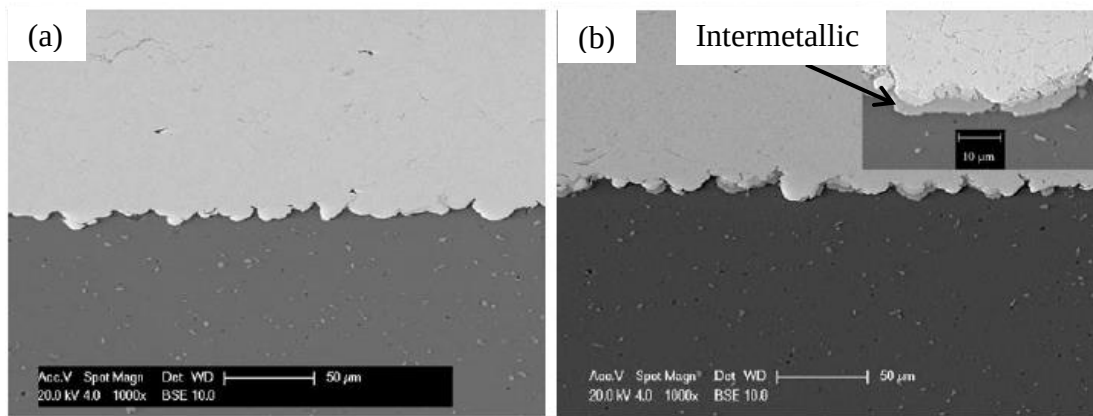
**Figure 2.22: Evolution of a copper particle hitting a copper substrate in CGDS: (a) 4.4 ns (b) 13.2 ns (c) 22.0 ns (d) 30.8 ns [53]**

Jetting is believed to (partially) break the oxide layer usually present at the surface of the particles and the substrate, as seen in Figure 2.23. Indeed, when a particle impinges upon a substrate that has naturally formed oxide layer as seen in Figure 2.23 (a), the oxide layers are believed to start breaking up due to the high plastic strains involved as shown in Figure 2.23 (b). The high plastic strains are also hypothesized to move outward the materials which include parts of the oxide layers and also, since the oxide layers (ceramic) will not plastically deform at the same rate (higher flow stress) than the metals involved, the latter will be potentially in contact as seen in Figure 2.23 (c). However, it is believed that not all of the oxides are removed and therefore some inclusions will remain at the end of the deformation as schematized in Figure 2.23 (d). Finally, by removing a portion of the oxide layers, the two materials are brought into intimate contact (Figure 2.23). A high contact pressure and a metal-to-metal contact are the two requirements in order to have metallurgical bonding. The pressure encounters during the cold spray process are high [53]. Therefore, metallurgical bonding could potentially occur during the cold spray process.



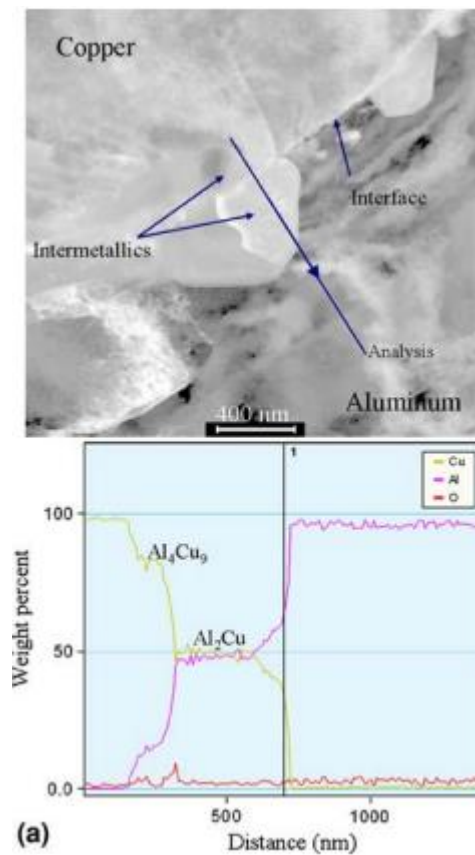
**Figure 2.23: Evolution of the bonding process involving the oxide removal for the cold spray process [54]**

In order to prove that metallurgical bonding could be, at least partially, the mechanism behind the adhesion in cold spray, an interesting method was developed by Price et al. [55] and Hussain et al. [56]. To characterize if metallurgical bonding could be present during the cold spray process of copper on aluminum, the deposition of copper powder on an aluminum substrate was accomplished with the coating and substrate subsequently annealed at 400 °C. The intermetallics formed during the annealing process are used as an indicator of the degree of metallic bonding [56]. The aluminum-copper diffusion (measured by the presence of intermetallics) indicates that the contact surfaces are clean of the oxide layer [57–59]. Otherwise, if the oxide layer is not broken, it is sufficiently thick to block the diffusion and no intermetallics will be found. On Figure 2.24, the intermetallics formed during annealing are seen as the grey zones at the interface. Since there are zones where there are not any intermetallic; it is presumed that the cold spray process did not remove the entire oxide layer. Inclusions of oxide layer are still presents at the interface. Price et al. observed an increasing degree of metal-metal contact as the particle's velocity increase, which means a better surface removal of the oxide layer at higher velocity [55].



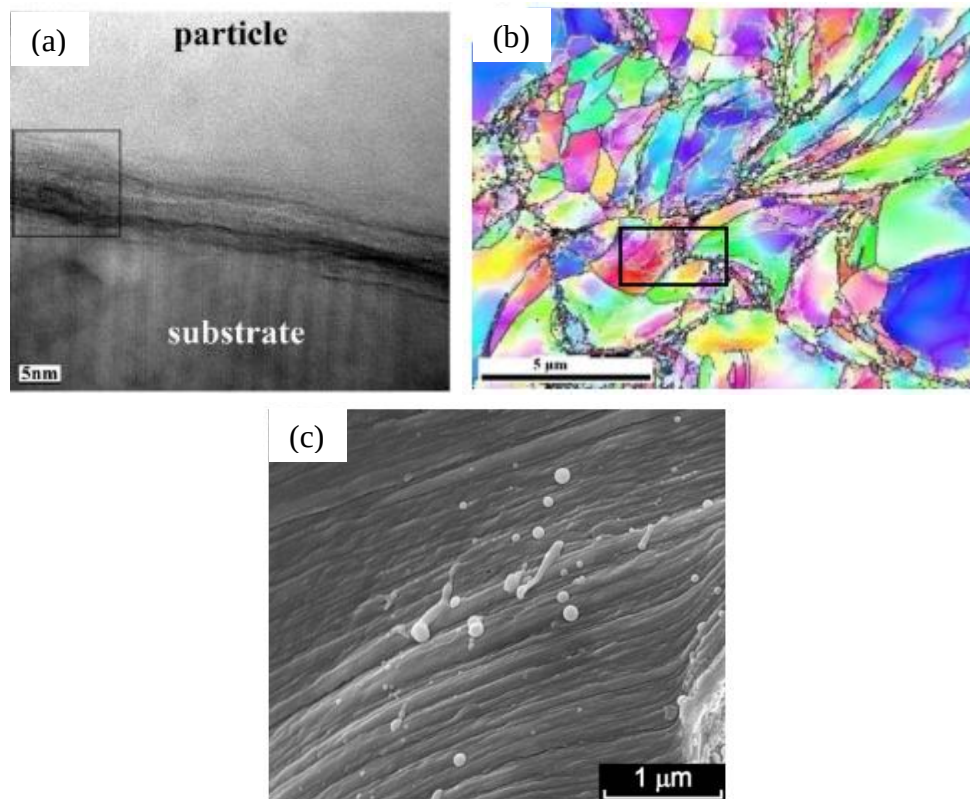
**Figure 2.24: Copper coating on aluminum substrate: (a) As-sprayed, no intermetallic found (b) Annealed at 400 °C, intermetallics present at the interface (adapted from [56])**

In the same vein, intermetallics have been observed at high magnification (Figure 2.25), at the interface between two different metals, on the as-sprayed coating.



**Figure 2.25: Intermetallics observed for a copper coating on an aluminum substrate [60]**

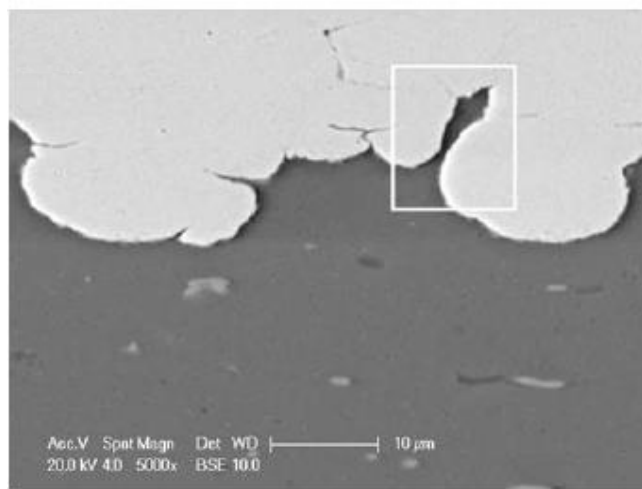
The formation of these intermetallics should occur following local melting since solid state diffusion process will produce a thinner layer than what is observed experimentally. Due to the short impact times, the diffusion distance for typical paired cold sprayed materials are in the order 0.1 nm whereas the intermetallics observed are in the order of several nanometers. Several authors reported an evidence of interfacial melting by amorphous regions, recrystallized grains or spheroidised particulate ejecta, as seen in Figure 2.26 [61–63]. It should be noted that the amorphous zones created by the local melting is retained down to room temperature due the rapid cooling at the interface by the abrupt thermal gradient after the localized deformation zone's heating [64].



**Figure 2.26: Evidence of interfacial melting: (a) High Resolution Transmission Electron Microscope (HRTEM) image of an amorphous region for aluminum [61] (b) Electron BackScatter Diffraction (EBSD) image of recrystallized grains of as-sprayed nickel coating [62] (c) SEM image of aluminum spheroidised particulate ejecta outside a crater [63]**

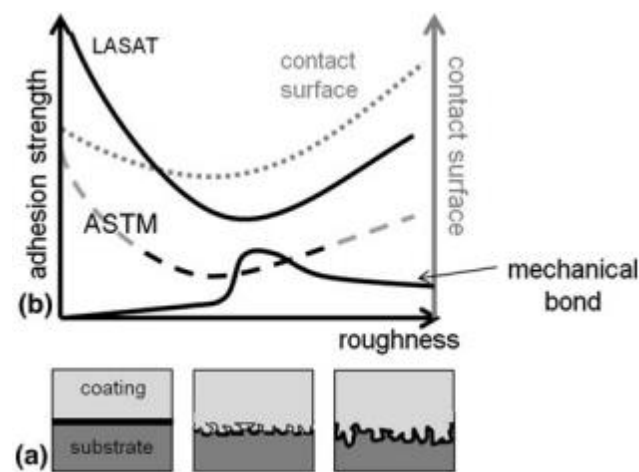
These results show that metallurgical bonding is a possible adhesion mechanism for the cold spray process, at least for some combinations of materials and parameters. Since it is believed that the formation of intermetallics on as-sprayed coating is due to local melting instead of solid state diffusion, the local temperature is an important variable which is dependent on the particles, the substrate and the cold spray parameters. Therefore, the combination of parameters is very important in order to have to presence of an intermetallic on the as-sprayed coating. Consequently, the lack of an intermetallic on the as-sprayed coating is not necessary meaning an absence of metallurgical bond as show by Hussain et al. [56] since the oxide layer is removed but the temperatures encountered do not promote the formation of intermetallic compounds.

Another bonding mechanism involved during the cold spray process is mechanical anchoring. The interlocking of the sprayed particles with the substrate and already deposited particles occurs due to the embedment of the particles in the substrate. Especially for hard particles on a soft substrate, the particles deform significantly the substrate (Figure 2.27), creating a surface profile.



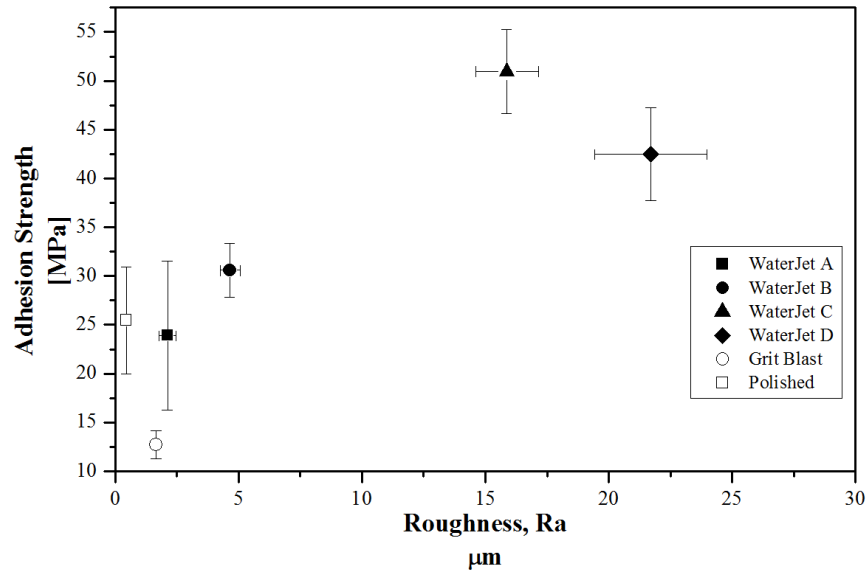
**Figure 2.27: Example of an interfacial locking for copper on an aluminum substrate [56]**

Therefore, the particles could be locked in these crevices created during the spray. Several studies focus on modifying the substrate surface using grit blasting to create, before the spray, a controlled amount of surface roughness, which will promote mechanical anchoring at a certain roughness value [56, 65–67]. Before that roughness value, roughening the surface is detrimental in terms of adhesion strength since the crevice is not large enough and the contact area of the particles on the substrate is lowered, as shown in Figure 2.28 [66].



**Figure 2.28: (a) Schematic of the interfacial contact area (b) Contact surface and adhesion strength as function of the roughness [66]**

The graph presented above was made with values of roughness ranging from 0.05  $\mu\text{m}$  to 5.53  $\mu\text{m}$  (deviation from the arithmetic average) [66]. Recently, a new method of increasing further the surface roughness was developed and preliminary studies were performed. This new technique, the pulsed water jet, uses a high-frequency high pressure jet of water to induce the surface roughness [65]. As shown in Figure 2.29, a higher surface roughness yields to a significant increase in adhesion strength [65].



**Figure 2.29: Adhesion strength vs. substrate roughness [65]**

To date, the relative importance of mechanical anchoring and metallic bonding to the total adhesion strength of cold sprayed coatings is still unclear. However, it is known that the materials involved, the spray conditions and the substrate preparation have an influence on these two bonding mechanisms and on the adhesion strength itself.

#### **2.2.2.2 Critical Velocity**

To be able to get in the adiabatic shear instability regime, the sprayed particles need to reach a so called critical velocity. The velocity of the particles at impact needs to be high enough to induce thermal softening higher than the work hardening, as explained in the bonding mechanisms section.

In order to understand the concept of critical velocity, the notion of deposition efficiency should be defined. The deposition efficiency represents the proportion of the sprayed material that is deposited on the substrate. In fact, in thermal spray processes, a certain amount of the materials is not deposited; they instead rebound from the coating causing a deposition

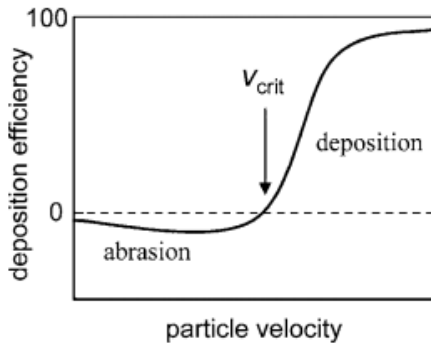


efficiency lower than 100%. Mathematically, the deposition efficiency ( $DE$ ) is defined as [64]:

$$DE = \frac{\Delta m}{M_0} \times 100 \quad (2.26)$$

where  $\Delta m$  is the mass gained by the component during the spray process and  $M_0$  is the total mass of powder injected in the system during the spray of the component.

Therefore, as it can be seen in Figure 2.30, below this critical velocity, the deposition is negative (material is removed from the substrate), if the particles are harder than the substrate, since the velocity is insufficient to deform plastically the particles. The particles are eroding the substrate in a similar way of the grit blasting process. As the particle velocity increases, at a certain point (critical velocity), they have sufficient velocity to deform plastically and some deposition could be noticed on the surface [68]. From that point, the deposition efficiency will increase with the particle velocity. Eventually, if the impact velocity is sufficient, deposition efficiency close to 100% can be reached, depending on the powder.



**Figure 2.30: Deposition efficiency vs. particle velocity [68]**

It should be noted that the critical velocity is dependent on the mechanical properties of the substrate and the particles, the impact temperature as well as the size of the particles [68]. However, if the impact velocity increases further, the deposition will eventually decrease as

shown in Figure 2.31. Ultimately, the particles will erode the substrate. This velocity is named the erosion velocity.

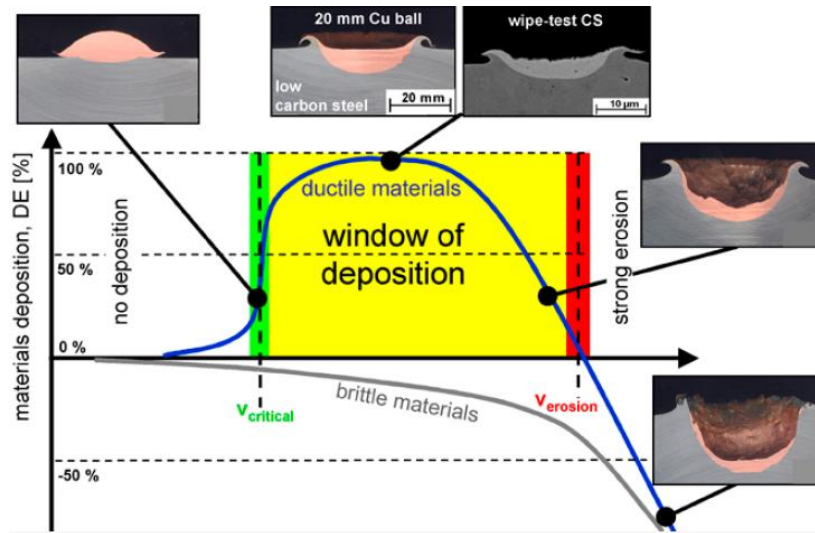


Figure 2.31: Concept of critical velocity and erosion velocity [69]

Consequently, it is important to be in the deposition window for the material in order to form a coating. The deposition window is defined as a range of velocities where the particles will adhere on the substrate. Figure 2.32 shows example of window of deposition for several materials.

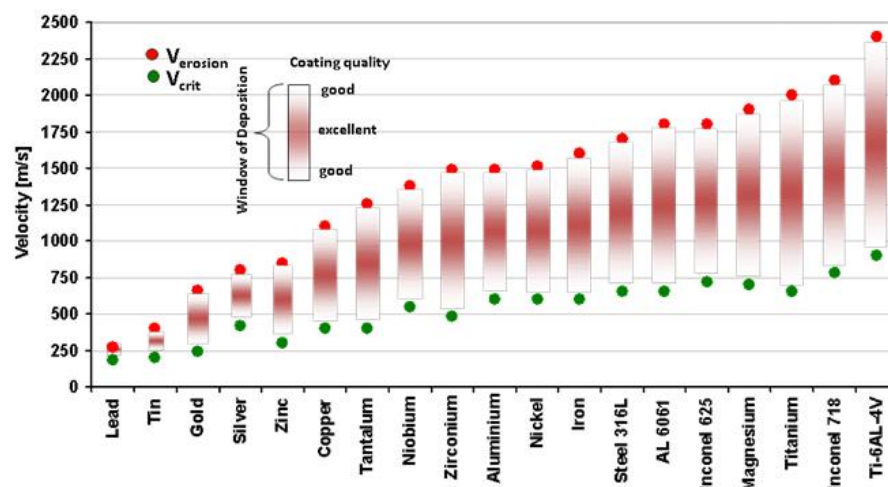
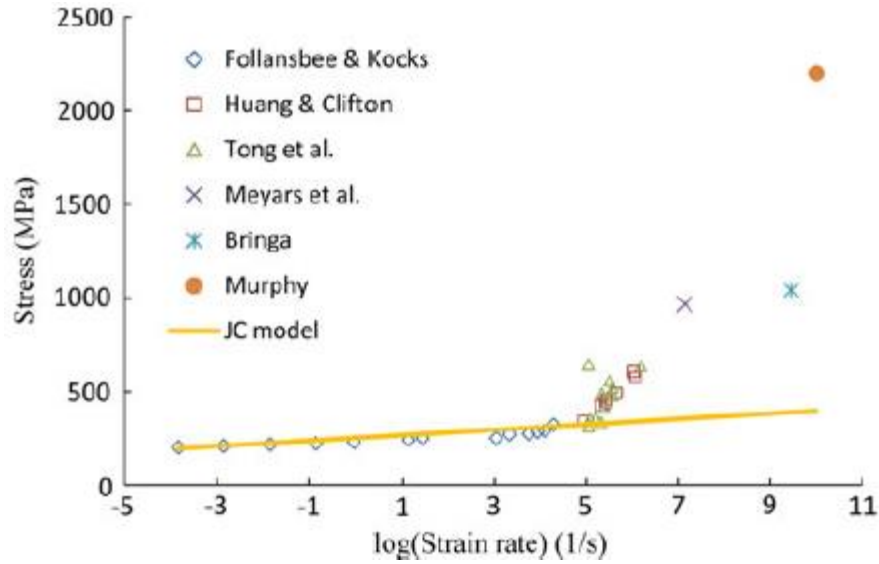


Figure 2.32: Example of window of deposition for various materials deposited by CGDS [70]

### **2.2.2.3 Process Modeling**

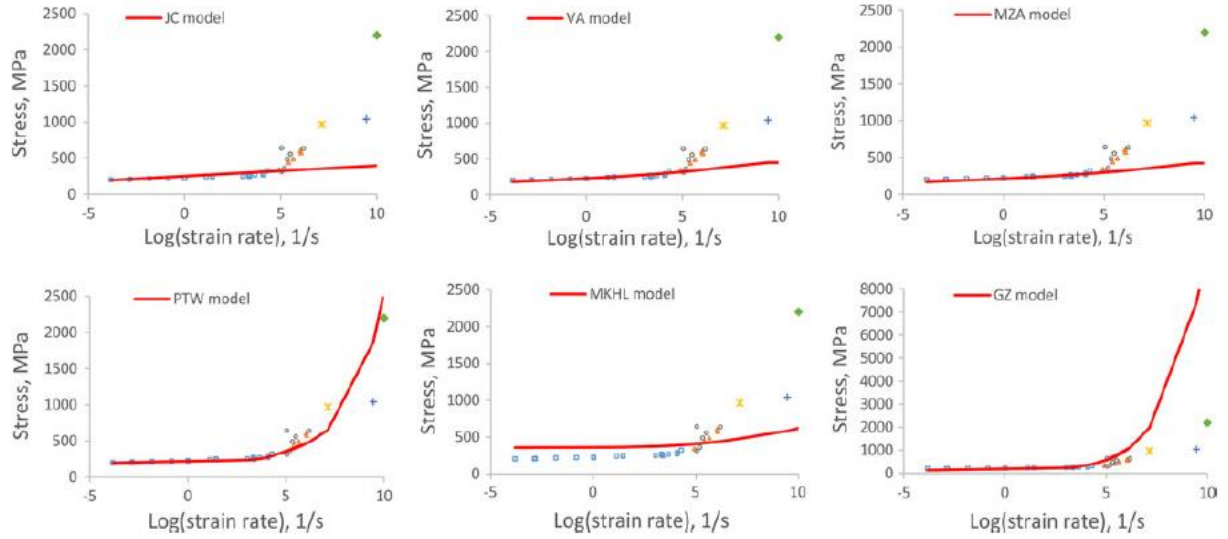
Over the past fifteen years, modeling the cold spray process has become more and more frequent. There are many difficulties involved with the experimental characterization of the bonding mechanisms during the adhesion process itself, as the time resolution required for these measurements are short (on the order of few nanoseconds). These difficulties for the experimental capture of data while the process occurs are not encountered when using numerical models, which make this type of investigation an attractive option to obtain insight into the physics involved in the bonding process. Assadi et al. [52] and Grujicic et al. [53] used such modeling techniques to demonstrate that the adiabatic shear instability and its resultant plastic flow localization contribute significantly to the adhesion of particles onto the substrate. Since these landmark publications, several authors have developed similar numerical models to study the adhesion mechanisms in cold spray in more detail [63, 69, 71–74], with varying degrees of success when it comes to the predictive ability of such models.

In particle impact modeling, the accuracy of a numerical model's predictions heavily lies within the choice of mechanical properties implemented in the model and with the deformation behavior model used for the materials considered. The mechanical properties of several materials for large deformation rates have been determined experimentally under various loading conditions, including for very high strain rate cases, where a sharp increase in the flow stress was noted for strain rates near  $10^5 \text{ s}^{-1}$ , as depicted in Figure 2.33, for pure copper.



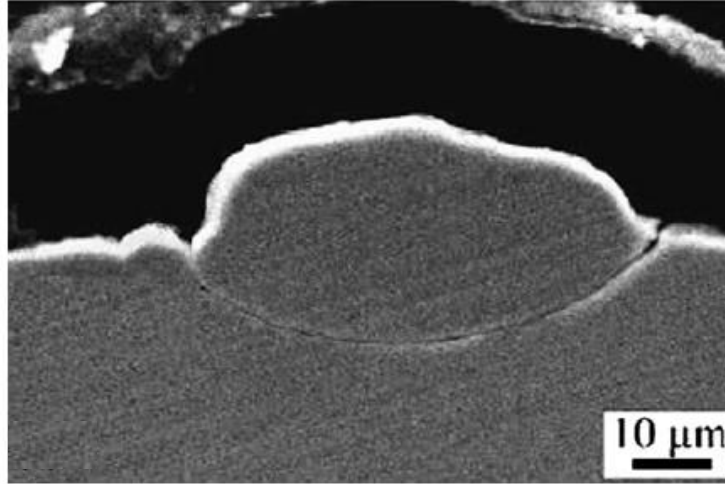
**Figure 2.33: Flow stress dependence on the strain rate for pure copper (data taken from [75–79]) with the predicted Johnson-Cook (JC) curve [80]**

Most numerical studies dealing with particle impact in cold spray have used the empirical Johnson-Cook (JC) model of material deformation, most likely due to its success in other high strain rate applications and since it is a rather simple model that is included in many commercially available finite element software packages. It was noted by several authors [73, 74, 80–83] that this model fails to accurately predict material deformation at strain rates higher than  $10^5 \text{ s}^{-1}$ , due to the sharp increase in flow stress at strain rates typically found in cold spray (Figure 2.33). These are often greater than  $10^7 \text{ s}^{-1}$  [73, 80, 84], which may explain the limitations of the JC model in successfully describing the particle-substrate behaviour and predicting the adhesion properties. It has been demonstrated by Rahmati and Ghaei [80] that the particle impacts at strain rates typically found during the cold spray process are more accurately predicted when using the Preston-Tonks-Wallace (PTW) model than by using other models such as the JC, Voyiadjis-Abed (VA), Modified Zerilli-Armstrong (MZA), Modified Khan-Huand-Liang (MKHL) and Gao-Zhang model (GZ) as shown in Figure 2.34.



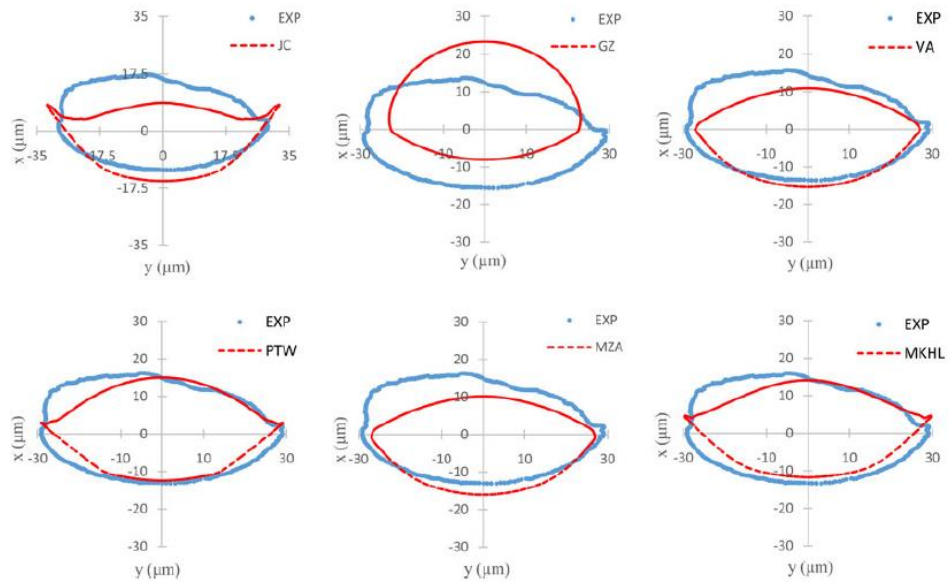
**Figure 2.34: Copper flow stress at various strain rates with the associated models [80]**

The PTW description of material behaviour is a complex constitutive model that was developed by Preston et al. [85] to predict the behaviour of materials at very high strain rates through its description of the dislocation motion during plastic deformation. As such, this model takes into account the sharp increase of the flow stress at strain rates exceeding  $10^5 \text{ s}^{-1}$ , which may explain its success in obtaining realistic splat geometries and substrate deformation in operating conditions such as those found in the cold spray process. The latter has been demonstrated by Rahmati and Ghaei by comparing the experimental cross-section of a copper particle deposited on a copper substrate (Figure 2.35) with the numerical cross-section results obtained for each studied model (Figure 2.36) [80].



**Figure 2.35: Cross-section of a copper particles deposited on a copper substrate [86]**

The comparison method used by Rahmati and Ghaei to determine which model gives a more accurate prediction of the cold spray deformation is the quantification of three errors (deformation, penetration and mean absolute) between the simulation and the experiment.



**Figure 2.36: Numerical cross-section for each model studied by Rahmati and Ghaei, compared to the experimental cross-section for copper on copper impact [80]**

The relative deformation error ( $\delta_1$ ) is defined as [80]:

$$\delta_1 = \left| \frac{D_E - D_m}{D_E} \right| \times 100 \quad (2.27)$$

where  $D_E$  and  $D_m$  are the particle deformation from the experiment and from the model at the centerline, respectively. These variables are schematized in Figure 2.37.

The relative penetration error ( $\delta_2$ ) is defined as [80]:

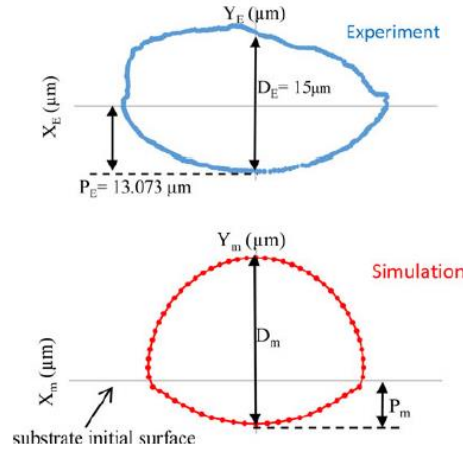
$$\delta_2 = \left| \frac{P_E - P_m}{P_E} \right| \times 100 \quad (2.28)$$

where  $P_E$  and  $P_m$  are the particle penetration from the experiment and from the model at the centerline, respectively. These variables are also schematized in Figure 2.37.

The mean absolute error ( $\delta_3$ ) is defined as [80]:

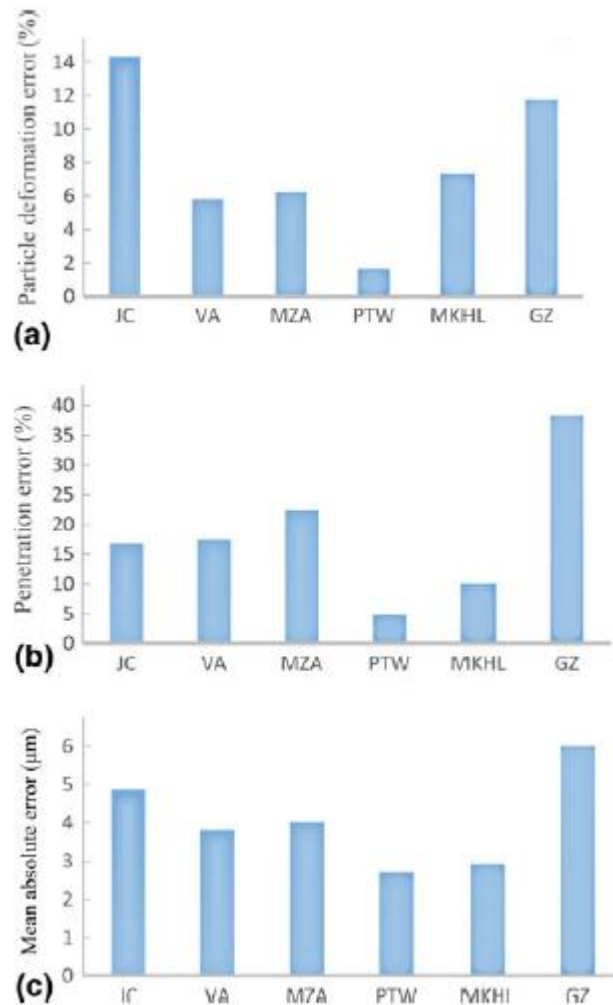
$$\delta_3 = \frac{1}{n} \sum_{i=1}^n |y_{Ei} - y_{mi}| \quad (2.29)$$

where  $n$  is the number of nodes located in the circumference of the particle for the model, and  $y_{Ei}$  is the  $i$  y coordinate of nodes for the experiment and  $y_{mi}$  is the  $i$  y coordinate of nodes for the simulation. These variables are also schematized in Figure 2.37.



**Figure 2.37: Schematic of the parameters used for comparison [80]**

By using the mathematical definitions of the three errors shown previously, the results from the simulations for each model and the experiment picture (Figure 2.35), Rahmati and Ghaei were able to compare quantitatively each model with an experimental result. Their findings are summarized in several histograms as shown in Figure 2.38



**Figure 2.38: Error histograms for: (a) Particle deformation (b) Penetration (c) Mean absolute [80]**

Therefore, it is possible to conclude from the histograms that the PTW model leads to the more accurate model since this model shows the lowest value for the three errors defined by Rahmati and Ghaei.



### **2.2.3 Application Range of Thermal Spray Processes**

The various thermal spray processes use a variety of temperature and particle impact velocity depending on the physical principles behind each, as shown in Figure 2.39. The electrical arc spray (wire arc in this figure) operates at higher temperatures than the CGDS (cold spray in this figure) and in the low range of particle velocity of the CGDS while plasma spray has extremely high process temperatures. Therefore, the application range for each thermal spray group is not similar. The deposition of certain material, sometimes, is not possible with a technique while it is possible with another. For example, CGDS is recognized to be able to produce low oxide content coatings with ductile materials but is still not a recognized process for the deposition of ceramic materials [47]. Plasma spray produces coatings with higher oxide contents but ceramic materials are easily deposited [47]. Knowing the application, the choice of the right thermal spray technique is then crucial.

In this study, CGDS, which has a low temperature and a high particle velocity, has an enormous advantage to manufacture fin arrays since the temperatures are lower than the melting point of metallic masks that can be used to form shaped deposits. The potential use of a mask to produce fin arrays will not be possible using other thermal spray processes since the mask will melt or molten particles will clog the mask by solidifying onto it.

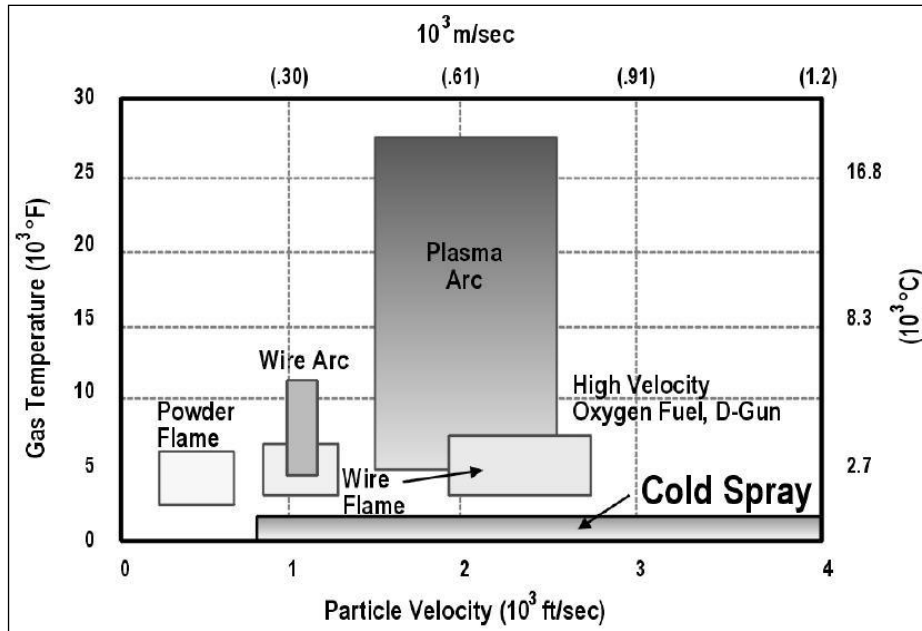


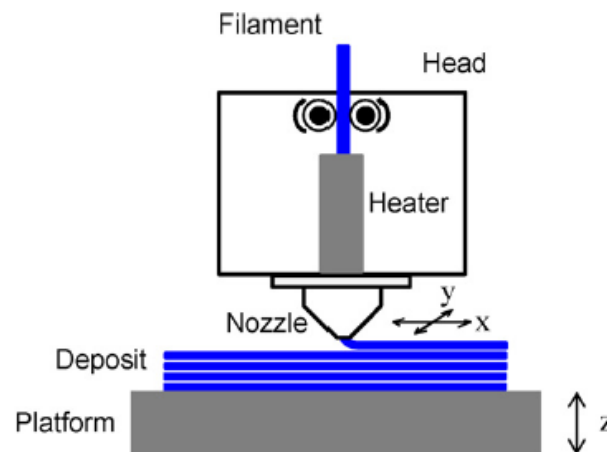
Figure 2.39: Different thermal spray processes with their typical range of gas temperature and particle velocity [87]

## 2.3 Additive Manufacturing

### 2.3.1 Introduction

Additive Manufacturing (AM) is defined as processes used to build three-dimensional (3D) features layer by layer. The major advantage of AM is to achieve satisfactory geometric accuracy while having the ability to use the raw materials in an efficient way by producing minimal waste [88–90]. Indeed, unlike conventional manufacturing methods that create components by removing materials from bulk or sheet metal, additive manufacturing produces the final feature by adding material [91]. Also, producing complex geometries that were not previously attainable by material removal processes is another interesting advantage about the AM techniques [91]. AM is also considered as an environmentally friendly manufacturing method with the reduction of wastes achievable [91].

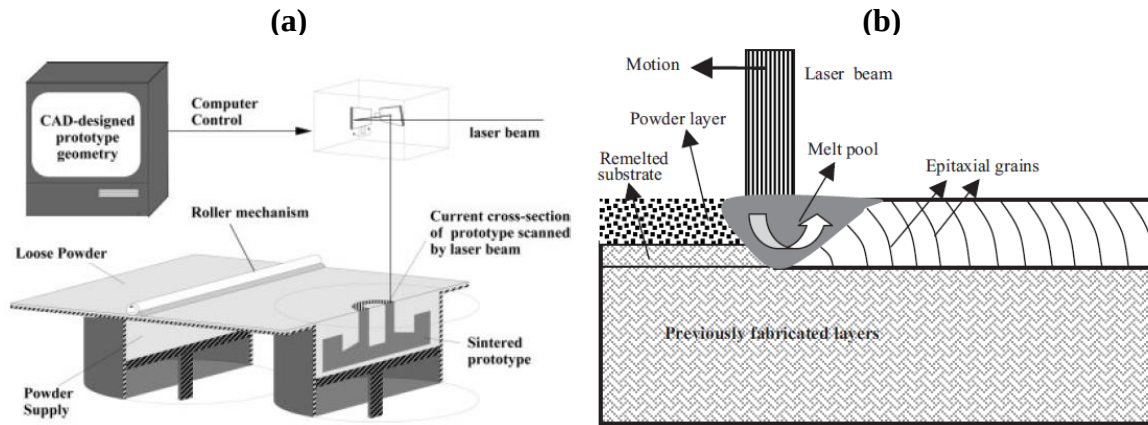
First introduced in the 1980's, AM has obtained a growing amount of attention with the advancement accomplished in the 3D printing processes over the last decades [92]. The most common 3D printing process nowadays is the fused deposition technique which consists of using a computer controlled nozzle which lays down layers of extruded molten polymers as shown in Figure 2.40.



**Figure 2.40: Schematic of the fused deposition technique [93]**

Eventually, a component is built from the deposited layers [94]. The robots controlling the nozzle follow a pattern based on a computerized 3D solid model in order to produce the desired features. The build-up speed is limited by the nozzle capacity [94]. The pieces produced with this method have usually good integrity but restricted structural strength [94]. Metals can also be used as feedstock for this type of technique. However, consolidation of the component should be obtained afterward by heat treatments such as sintering [94]. This type of AM technique is still under development [94]. Other techniques to obtain near-net shaped metallic components have been developed and successfully used; nevertheless they are less often utilized in the industry. For example, selective laser sintering is a technique that sinters powdered materials, usually metallic particles with a high power laser. The laser is aimed

according to the 3D model drawn on the computer in order to fuse and bind materials at a defined location to create a solid component as shown in Figure 2.41 [91].



**Figure 2.41: (a) Schematic of the selective laser sintering process (b) Laser material interaction [95]**

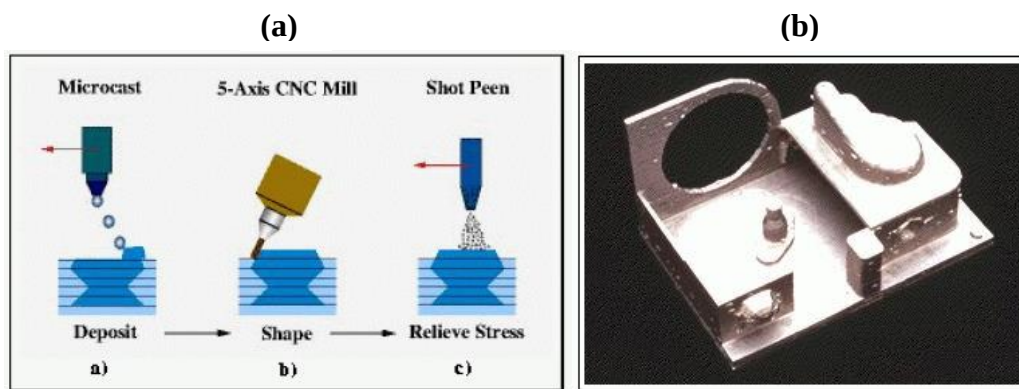
Techniques such as stereolithography, laser engineered net shaping, laminated object manufacturing and electron beam melting have been also proved to successfully build 3D metallic part [91].

Some drawbacks have been noticed for AM techniques. First, the size limitation is a concern since large sized components are often unrealistic to be produced because of the low material building rate [91]. Second, the cost is still an issue even if the price dropped significantly in the current decade. In particular, the cost of AM equipment for manufacturing metallic components is a high investment [91].

### **2.3.2 Additive Manufacturing and Thermal Spray Processes**

The use of thermal spray processes as AM processes is fairly new. Conventionally, thermal spray techniques are used to deposit coatings instead of building near-net shape features. The first notable use of thermal spray as an AM process has been reported in the mid 90's. Several

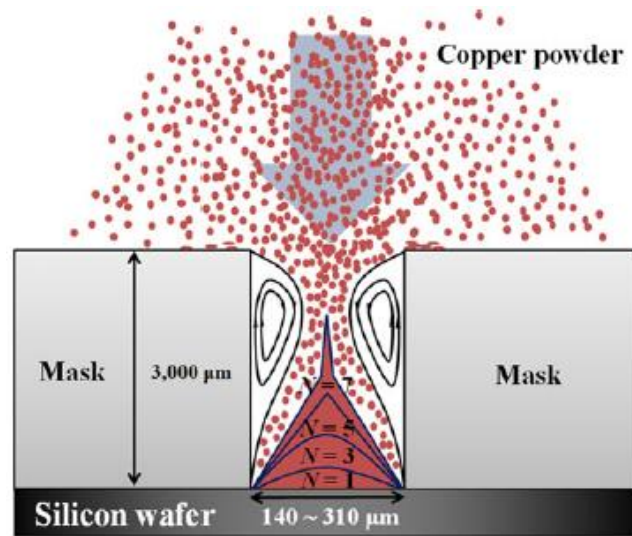
studies have been performed by a group led by Prinz and Weiss concerning the deposition of molten material using the plasma spray process, shaped with the use of a stackable mask, which resulted in a U.S. patent [96]. This process consists of using a disposable mask to create different shapes using plasma spray deposition, which is named microcast in their paper, followed by machining with a computer numerical control mill as shown in Figure 2.42 (a) [97]. In their process, there is also a step that includes shot peening in order to relieve some of the stresses developed during manufacturing. The three steps shown Figure 2.42 (a) are repeated for each deposited layer until the final part is completed [97]. An example of a part produced with this technique is shown in Figure 2.42 (b). This method has several major flaws, such as time consuming, poor mechanical strength, warpage and thermal stress issues related to the nature of plasma spray deposition [97].



**Figure 2.42: (a) The process developed by Prinz and Weiss (b) Example of a stainless steel 308 part produced with this technique [97]**

More recently, Kim et al. [98] studied the possibility of using CGDS with a machined stainless steel mask with rectangular slits in order to produce near-net copper electrode for solar cells application. The results are pyramids with a rectangular base. They explained the pyramidal shape obtained by CGDS is due to the bottleneck effect, as depicted in Figure 2.43. When the stream of gas goes within the slit of the mask, a recirculation zone is created at the

mask's sharp corner. The particles lose their vertical momentum in that recirculation zone. As a result, the particles have a tendency to center in the hole of the slit. After the recirculation zone, the particles still can be deposited outside of the center resulting in a prismatic shape.



**Figure 2.43: Schematic of the bottleneck effect [98]**

However, this method leads to few imperfections such as the mask could be embedded into the silicon wafer (substrate) since there is no spacing between the mask and the substrate as well as the height of the electrode is limited to this thickness of the mask [98].

## **Chapter 3 - Research Objectives**

### **3.1 General Objectives**

The general purpose of this research work is to improve the manufacturing method of fin arrays currently produced at BEC while ensuring a better performance. Investigation of the relevant literature has suggested that the CGDS is a promising process to produce fin arrays through the use of a mask. The literature review has also presented the advantages of producing discontinuous fins instead of the continuous fins currently used at BEC. Following the literature review, to have the optimal fin arrays and a commercially viable production technique, this research work proposes to deposit near-net fin arrays using CGDS deposition through a mask as an additive manufacturing process that could produce well bonded, discontinuous fins. Furthermore, the mask should have no possibility to be embedded into the substrate. The aim of this research project is consequently a) to investigate whether CGDS is a potential solution for the manufacturing of discontinuous near-net shape fin arrays b) to show that the resulting fin arrays have a better performance than the traditional machined fins c) to determine if these fin arrays are viable for use in industrial applications and d) to understand the impact phenomena in order to explain some experimental results.

### **3.2 Feasibility of CGDS to Produce Discontinuous Pin Fin Arrays**

The first specific objective of this research work is to demonstrate the feasibility of manufacturing discontinuous pin fin arrays by CGDS. Several types of mask will be used between the spray nozzle and the substrate in order to create pin fins. The principle behind this idea is shown in Figure 3.1. The selected mask needs to be available commercially, relatively cheap, easily removable from the substrate, reusable for several sprays and allow

the production of near-net shape pin fin arrays. If it is impossible to find an available commercial mask, it needs to require the least machining time/cost as possible. It is expected to get pyramidal shape fin as shown in the literature survey. It should also be noted that the feasibility study will be accomplished with pure aluminum powder because it is well known to be easily deposited with CGDS.

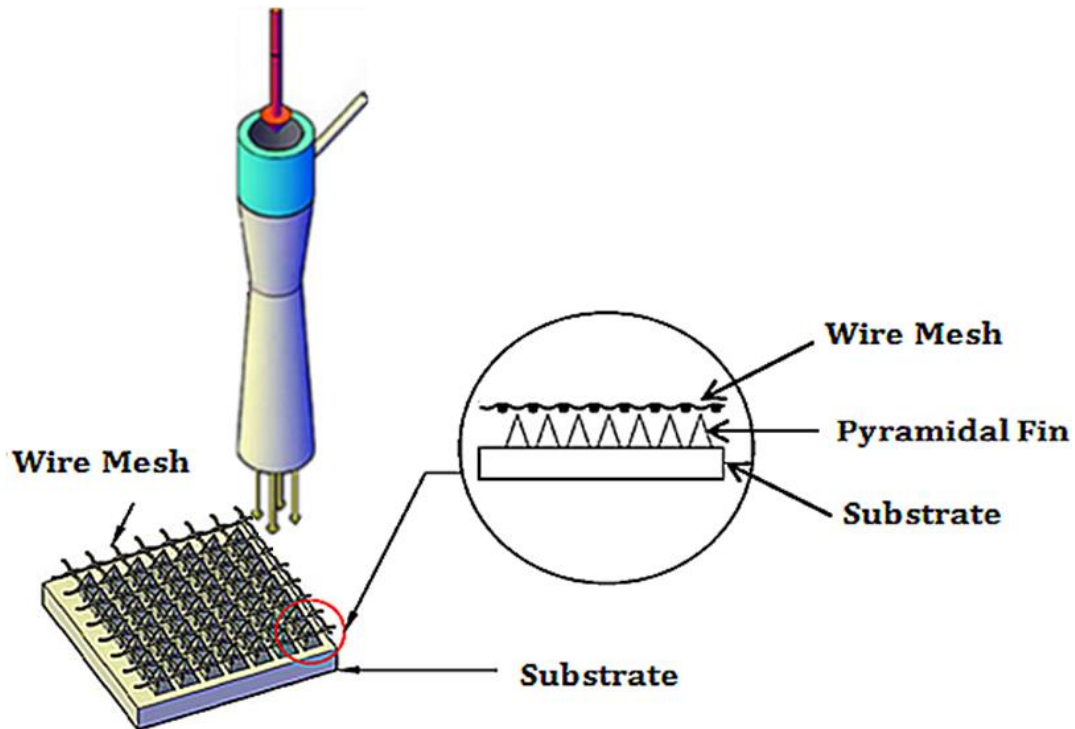


Figure 3.1: Proposed spray configuration to produce near-net fin arrays

### 3.3 Investigation of CGDS Parameters

Once the feasibility of manufacturing pin fin arrays by CGDS is proven and the appropriate mask is selected, the next step will be a qualitative and quantitative investigation of the CGDS parameters in order to achieve high quality pin fin arrays. Parameters such as the distance between the nozzle and the mask as well as between the mask and the substrate will be investigated in order to prevent material deposition between the fins. By varying several spray parameters such as driving pressure, gas temperature and feed rate, the porosity level in the



produced fin will be investigated. Also, the maximum fin density that can be obtained will be examined since at a certain point the mask will possibly be clogged by the particles.

### **3.4 Thermal and Hydrodynamic Performance**

The performance of the new type of pin fins will be also evaluated. Two important parameters for the pin fin arrays are the thermal performance as well as the hydrodynamic performance, as seen in the literature survey. Therefore, these two parameters will be determined experimentally with an in-house test apparatus. Once these two types of performance are found, an appropriate comparison method will be chosen to study the effect of the geometric parameters such as the height, fin density and base angle as well as to allow a comparison with the plain rectangular fins currently used at BEC.

#### **3.4.1 Comparison with Plain Rectangular Fin Arrays**

In order to ensure that fins produced have a better performance than fin arrays currently used at BEC, a comparison is necessary. The thermal conductance, the convective heat transfer coefficient and the pressure drop will be compared. According to the literature survey, it is expected that the thermal conductance, the convective coefficient of heat transfer and the pressure drop will be higher. A higher thermal conductance and convective coefficient mean that the fin array exchange in terms of thermal energy is better; however there is a higher cost in pressure drop. Therefore, it is important to compare these two kinds of fin arrays with a method that includes the pressure drop and the thermal performance in order to have a better overall picture. The selected comparison method will be explained in the section 4.3.3. It should be noted that a comparison with a straight cut fin array machined from a plain coating and machined from bulk aluminum will be performed in order to be able to compare to the

sprayed fin array. A comparison with both type of straight cut fins are necessary since the first type (produced from a coating) is what is currently manufactured at BEC and the second type (produced from bulk material) is what is traditionally manufactured for straight cut micro-fin arrays.

### 3.4.2 Effect of Base Angle

Another parameter to investigate is the effect of the base angle. The base angle is defined as the angle between the edge of the fin and the horizontal as depicted in Figure 3.2 .

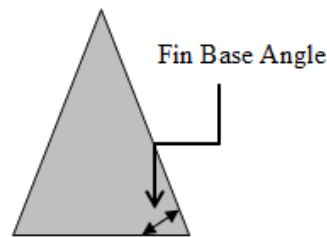


Figure 3.2: Schematic of the fin base angle

The procedure will be to spray several samples at different heights and removing the top part of the fin to get the same height for every sample as shown in Figure 3.3. By this procedure, the base angle will be varied from sample to sample but the height will be the same. The contact surface with the flow will be similar for each sample since the top is shrouded for every test. The effect of the base angle will be quantified using the thermal and hydrodynamic performance measured with an apparatus developed at the University of Ottawa (section 4.3).

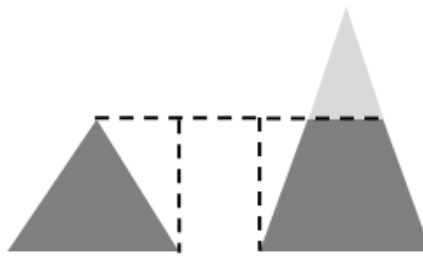


Figure 3.3: Schematic showing the difference in fin base angle

### **3.4.3 Effect of Fin Height and Fin Density**

Different fin heights and fin densities have been proven to give dissimilar thermal and hydrodynamic performance for other types of fin [21, 25, 99, 100]. Therefore, these two parameters will be tested to see if there is an optimum fin height and fin density. Correlations for the Nusselt number will be also produced. Indeed, a correlation linking the Nusselt number to the Reynolds number as a function of the fin height will be tentatively produced. Another correlation linking the Nusselt number, the fin density and the Reynolds number will be also tentatively produced.

## **3.5 Adhesion Strength**

Once the geometric variables such as base angle, fin density and fin height will be well understood, the viability of using these pin fins in an industrial application will be the focus of this work. The highest stress that is going to be sustained by the pin fins will be a shear stress since the flow will tend to shear the fins from their base during normal operation. For this reason, the fins adhesion will be tested in shear mode to determine if the adhesion and cohesion of the pin fins are sufficient to sustain the real stresses produced by the working fluid. Various types of cold spray pin fin will be experimented upon, in particular several heights and fin densities. For that part of the project, the pin fin arrays will also be sprayed on the bond coat manufactured by wire arc spray which is currently used at BEC with the purpose of knowing if there is a difference in adhesion/cohesion compared to the fins deposited directly on an aluminum substrate.

### 3.6 Fin Arrays with Streamwise Anisotropic Materials

Usually, for high temperature applications such as recuperators, the same material is used for the pin fins along the stream as it can be seen in Figure 3.4.

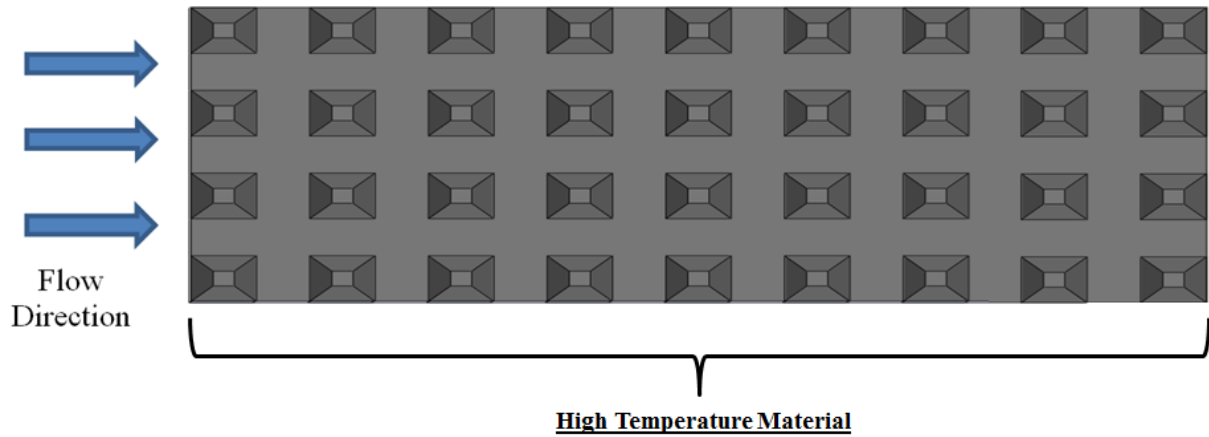


Figure 3.4: Schematic of typical pin fin material for high temperature application

The viability of varying the material along the stream length, as depicted in Figure 3.5, will be tested.

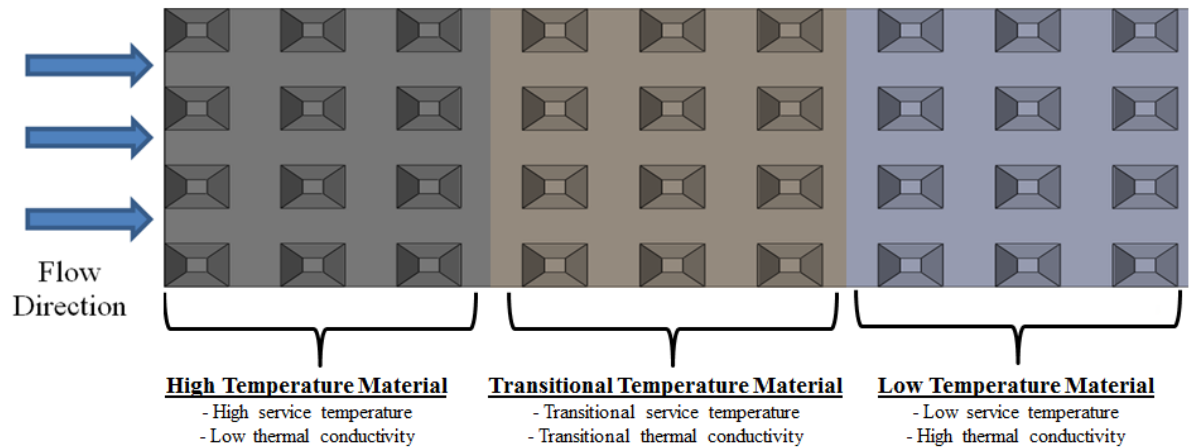
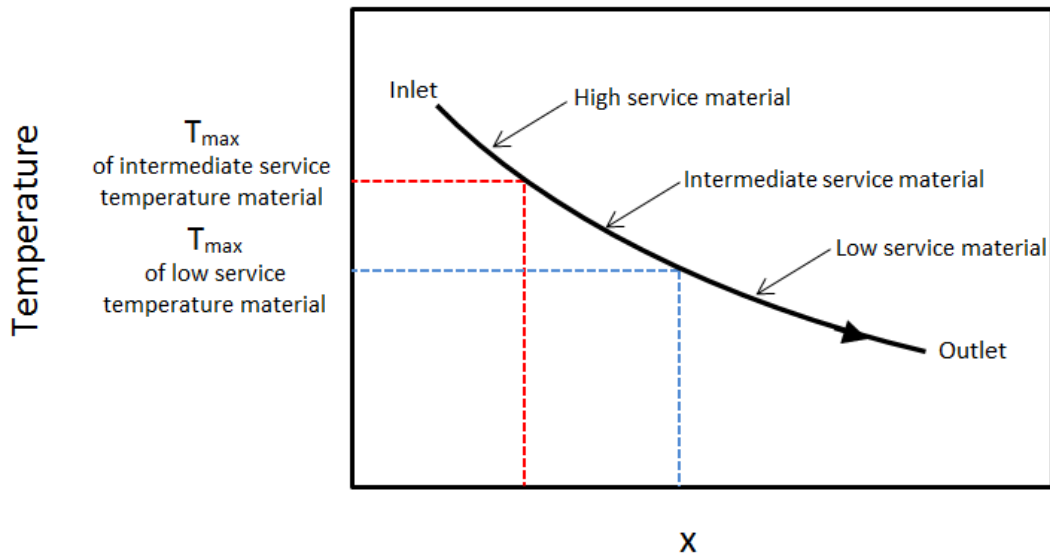


Figure 3.5: Schematic of the innovative streamwise anisotropic material approach

The novelty of this approach is that once the flow temperature is low enough, the material will be changed to one that has a lower efficient service temperature but a higher thermal

conductivity. The heat exchanger could be separated into sections. Indeed, a transfer of energy occurs between the fins and the fluid, resulting in a decrease of the local fluid temperature along the length of the heat exchanger, as depicted in a typical cooling curve of temperature as a function of distance (Figure 3.6).



**Figure 3.6: Typical cooling curve of temperature as function of distance**

The possibility of creating multi-material heat exchangers would allow the designer the freedom of using the material combination which not only maximizes the thermal performance, but also allows for the substitution of high cost, heavy alloys by lighter and cheaper materials in the low efficient service temperature section of the heat exchanger.

In order to prove the concept of streamwise anisotropic materials, the thermal and hydrodynamic performance of these fin arrays will be tested experimentally. A comparison between single material samples and a multi-material sample will be performed.

### **3.7 Numerical Simulation of the Cold Spray Impact Process**

After the streamwise anisotropic samples trials, the shear strength of each material will be quantified and characterized. The potential differences in shear strength will be tentatively explained by modeling the particle impact using finite element analysis since the bonding process cannot be described and visualized experimentally due the high deformation rate of the particles and substrate during the impact. Consequently, a numerical simulation of particles impacting on a substrate will be performed using the commercial software package Abaqus. Analysis of results obtained by the simulation will be investigated to have a better fundamental understanding of the cold spray impact process itself.

## **Chapter 4 - Experimental and Numerical Research Approach**

This chapter details the experimental research approach, the numerical model developed and the equipment used to be able to achieve the previously outlined research objectives. More specifically, this chapter will define the experimental apparatus used to deposit the fin arrays, the different analysis techniques required to characterize the fin arrays, the experimental setup used to determine the thermal and the hydrodynamic characteristics of the fin arrays produced by CGDS, the equipment required to evaluate their shear strengths, and finally the finite element analysis model used to simulate cold spray impacts.

### **4.1 Experimental Equipment to Manufacture Fin Arrays**

The present section gives details about the equipment used to produce fin arrays. For this project, there are access to two cold spray facilities, one at BEC and the other at the University of Ottawa. The first one was used to do preliminary and feasibility studies as well as to provide correlations to the industrial partner for their model. Once this is proven to be feasible and some correlations were produced, more in-depth studies such as the evaluation of the mechanical properties and the production of streamwise anisotropic samples were performed with the cold spray system at the University of Ottawa. For that reason, more details will be given for the University of Ottawa's cold spray system.

#### **4.1.1 Brayton Energy Canada Cold Spray Facility**

The system owned by BEC is a commercial Plasma Giken PCS-1000 (Plasma Giken Co. Ltd., Saitama, Japan), shown in Figure 4.1.



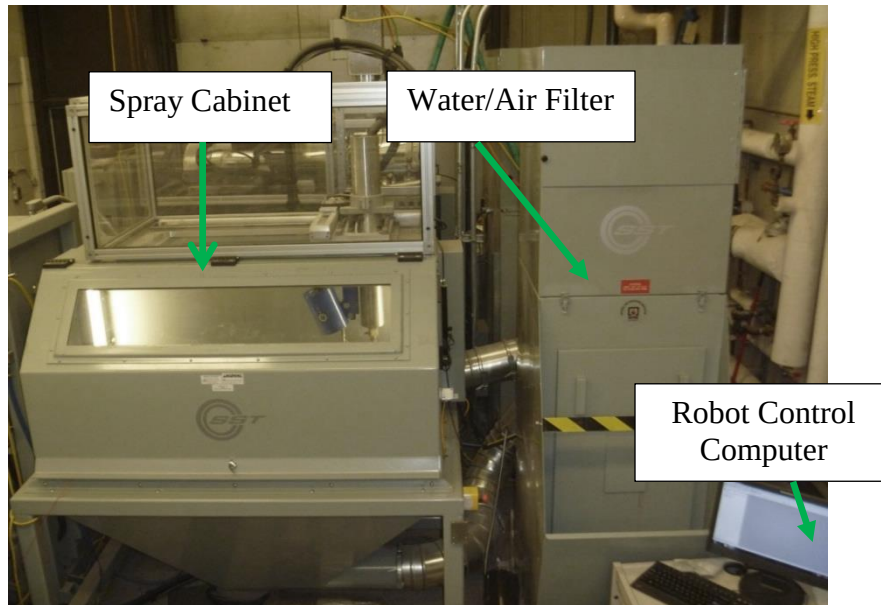
**Figure 4.1: Plasma Giken PCS-1000 located at BEC's facility**

The PCS-1000 is a high pressure system which means that the powder is injected axially before the throat of the nozzle. The spray nozzle that was used to do the feasibility study has a 3.0 mm throat diameter and a 6.5 mm exit diameter. This system has the capability to spray at high powder feed rates which makes it suitable for high volume commercial applications. It can also reach gas temperature of up to 1000 °C and a maximum gas pressure of 5 MPa. Therefore, since it has the capacity to reach high pressure and gas temperature, materials that have a higher critical velocity, such as stainless steel and Inconel, could be deposited more easily with this system while maintaining high deposition efficiencies. BEC uses nitrogen ( $N_2$ ) as propellant; therefore  $N_2$  was used throughout these studies.

#### **4.1.2 University of Ottawa Cold Spray Laboratory Facility**

The spray system that has been used at the University of Ottawa Cold Spray Laboratory is the SST EP cold spray system manufactured by CenterLine (Windsor) Limited (Windsor, Ontario, Canada), shown in Figure 4.2. The sample to be sprayed is placed into the spray cabinet, which is used to protect the users from the particles in suspension in the air. The non-deposited powder is collected in the water/air filtering system.

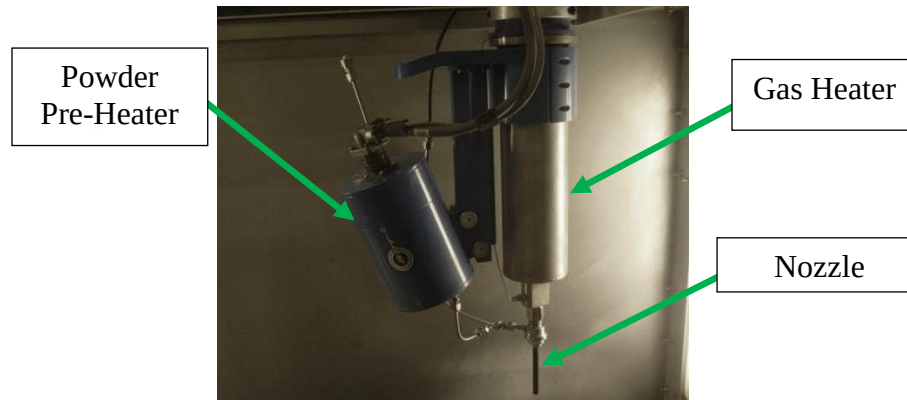




**Figure 4.2: Cold spray system SST series EP provided by Centerline**

This system has a lower maximum gas temperature and pressure, compared to the Plasma Giken system, with a maximum gas temperature of 550 °C while the gas pressure can reach up to 3.4 MPa. However, even if the system used by the industrial partner has an higher maximum gas temperature and pressure, it is justified to use the system at the University of Ottawa for the in-depth studies since if it is possible to deposit the fin arrays with the lower maximum parameters system, it will be feasible to deposit the same materials with a system with higher maximum parameters.

For the system at the University of Ottawa, the gas temperature is increased by means of a heater, shown in Figure 4.3, while the gas pressure is controlled by a valve. The nozzle used is a converging-diverging (DeLaval) nozzle. It has a throat diameter of 2.0 mm and an exit diameter of 6.4 mm.



**Figure 4.3: Assembly inside the spray cabinet**

With the system provided by CenterLine, it is possible to move the nozzle in planar X-Y direction with electrical stepped motors that are used to move linear screws. Movement in the height direction is not presently possible. The speed of these movements and the displacement distance are controlled by a software included with the system.

The supply of powder to the main flow stream is controlled by a commercially available powder feeder (Model: AT-1200HP, Thermach Inc., Appleton, Wisconsin, United States of America) shown in Figure 4.4.



**Figure 4.4: Thermach powder feeder**

The powder feeder consists of a pressurized canister and a perforated wheel equipped with a hammer. The hammer ensures constant feeding of powder by filling the holes in the wheel. It should be noted that the canister is pressurized with  $N_2$  which is also called the carrier gas. The velocity of the wheel as well as the size of the wheel holes can be varied to control the powder feed rate. Increasing the size of the holes and/or the angular velocity will increase the rate of powder supplied to the system. The canister is pressurized in order to create a differential pressure between the main flow and the powder feeder, which pushes the powder toward the nozzle. The powder could be pre-heated, if necessary, on its way to the main flow stream, as shown in Figure 4.3.

#### **4.1.3 Arc Spray Equipment**

Arc spray equipment was available at the industrial partner site. Samples that required a layer of bond coat were produced by the staff at BEC, using the heating rates, inlet pressure, wire feed rates, wire composition and traverse speed typical of those used in WMHE projects. The parameters used are given in the Paper #4 (section 5.4). The arc spray coatings were made using a Jet Force Arc Spray System 456HD, model 57456HD manufactured by Thermion Inc. (Silverdale, Washington, United States of America). Once the arc spray coatings were deposited, the fin arrays were produced on the arc spray layer by CGDS.

### **4.2 Material Characterization**

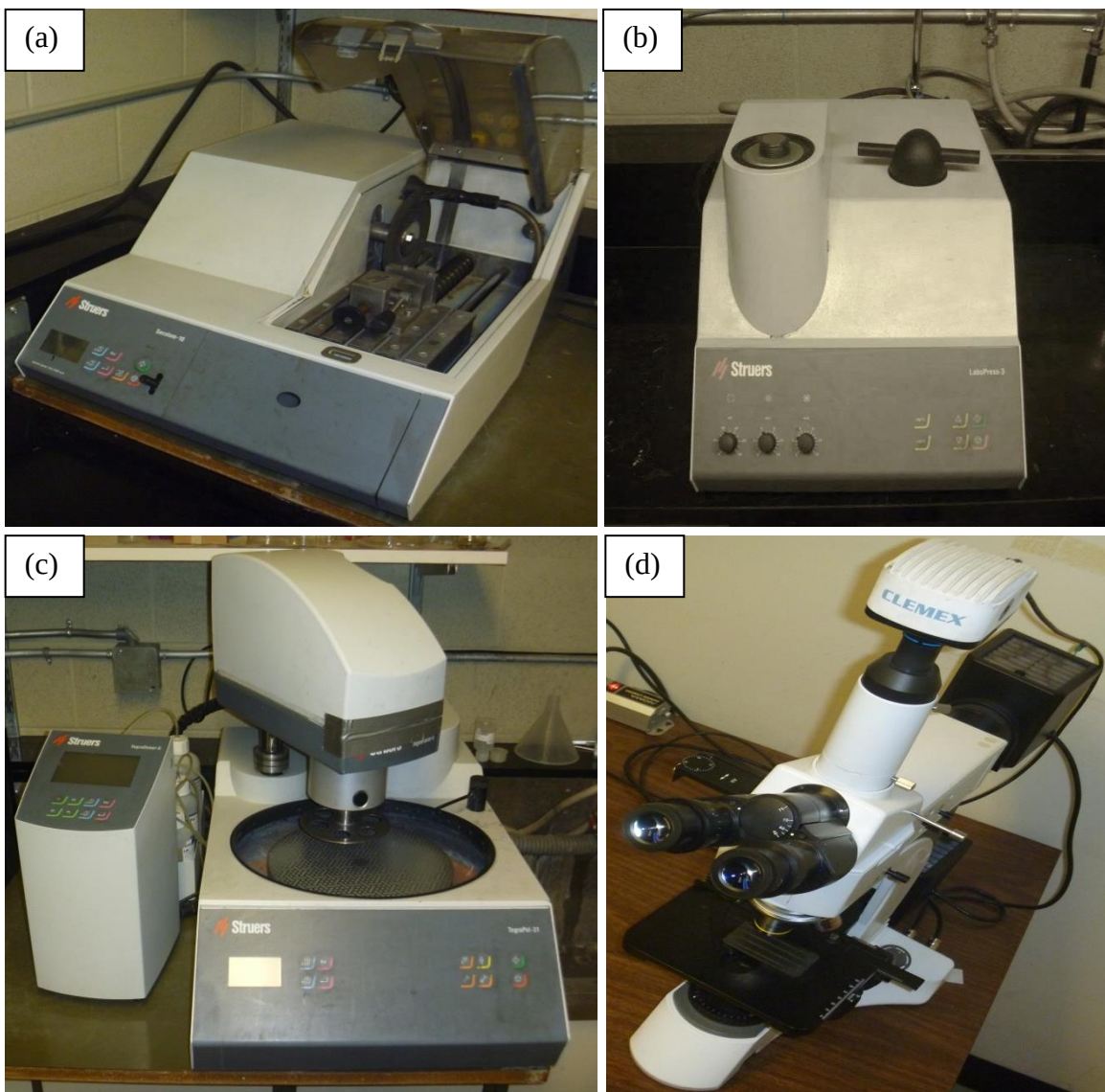
In order to achieve several of the objectives defined in Chapter 3, various material characterization techniques need to be employed. This section presents the various analysis methods used for the characterization of fin array samples for the feasibility study, for the investigation of the CGDS parameters, for the characterization of the feedstock powders and

to obtain the geometric data required for the thermal and hydrodynamic performance tests. This section presents also the techniques used to evaluate the fin and the substrate roughness as well as to determine the particle's velocity.

#### **4.2.1 Coating Characterization for the Feasibility Study and the Investigation of the CGDS Parameters**

In order to find the fin arrays properties manufactured through this research work, qualitative and quantitative methods were used. The feasibility of the use of CGDS to manufacture discontinuous pin fin array was demonstrated by noticing the formation of coating build-up that forms shapes resembling pin fins. Investigation of the CGDS parameters requires a more in-depth characterization of the coatings. This was accomplished by examining the coating cross-sections using optical microscopy. In order to be able to observe the cross-sections of the pin fin arrays, several steps need to be followed. Standard metallographic techniques were used to get as close as possible to the true coating microstructure. A true microstructure is theoretically obtained when the cross-sections is completely without artifacts (no deformation, no scratches, no pull-out, no introduction of foreign elements, etc.). Practically, a true microstructure can never be obtained, as there will be always several artifacts introduced by the preparation methods. However, standard metallographic techniques reduce these artifacts to a minimum. As the first step of the standard metallographic technique, the samples are sectioned using a Struers (Ballerup, Denmark) Secotom-10 cut-off saw (Figure 4.5 (a)) with the purpose of exposing the cross-section of the pin fins. Then, the cut section is mounted in a thermosetting epoxy resin (Struers LaboPress-3, Figure 4.5 (b)) in order to facilitate the following preparation techniques. Further preparation steps are necessary to obtain acceptable surface finish that can be observed under the optical microscope. The cross-section is ground

and polished to a 1  $\mu\text{m}$  surface roughness finish using the following equipment: Struers TegraPol, TegraForce-5 and TegraDoser-5 shown in Figure 4.5 (c). The optical microscope presented (Clemex Technologies Inc., Longueuil, Québec, Canada) in Figure 4.5 (d) was utilized to observe, qualitatively, if there is any deposition between the pin fins. In addition, the optical microscope was used, qualitatively, to see if there are some defects in the pin fin structures, such as cracks, interface defects, etc.



**Figure 4.5: Instruments used for standard metallographic techniques: (a) Sectioning (b) Mounting (c) Grinding and polishing (d) Optical microscope**

The pin fin cross-sections were also analyzed using the Clemex Vision Lite (Clemex Technologies Inc.) image analysis software in order to measure the porosity level quantitatively. The CDGS parameters were considered acceptable when there is a porosity less than 5% for any powders used, with no visible defects and no or slight deposition between the fins.

#### **4.2.2 Powders Characterization**

Various powders were used in this work in order to fulfill the objectives previously discussed in Chapter 3. Powders are characterized by the particle's morphology and the size distribution. Consequently, this section shows the analytical instruments that were employed to characterize the powders. First, a qualitative study was conducted using a Scanning Electron Microscopy (SEM) with the purpose of observing the average shape of the powders. Furthermore, the grain structure of the powders was also observed, when the powders were attacked by an etching reagent. The SEM used is the model EVO MA10 from ZEISS (Oberkochen, Germany), shown in Figure 4.6, which is available at the University of Ottawa.



**Figure 4.6: Scanning Electron Microscope utilized at the University of Ottawa**

The powder size distribution was quantitatively characterized using a Microtrac S3500 (Montgomeryville, Pennsylvania, United States of America) laser diffraction analysis shown in Figure 4.7. This instrument gives the average size distribution by using a laser diffraction analysis, which also determines the standard deviation from the average size.

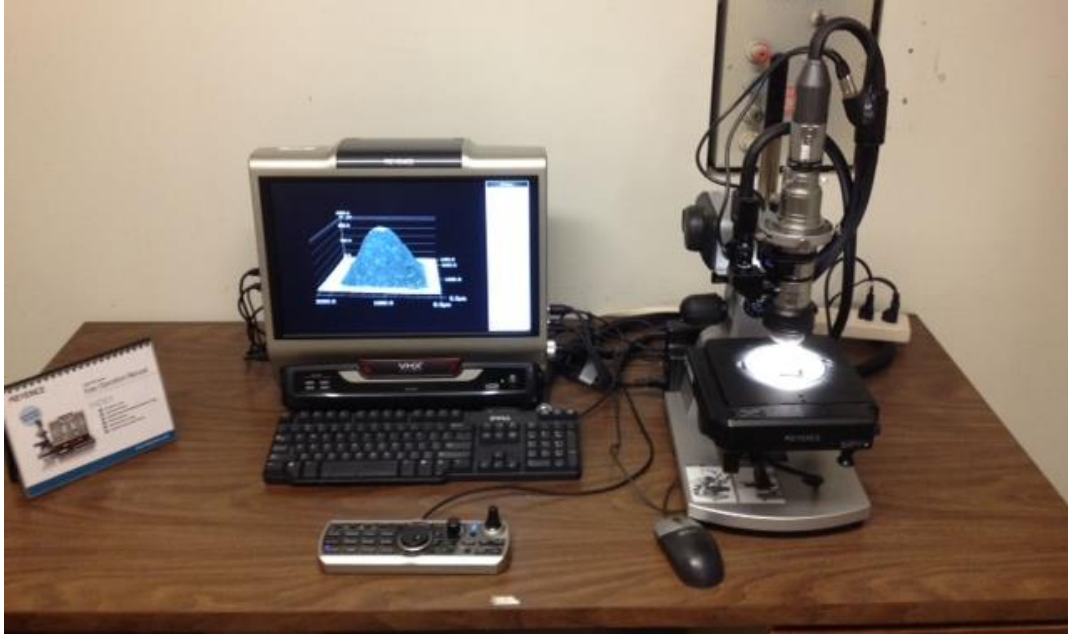


**Figure 4.7: Microtrac S3500 used to obtain the powder's size distribution**

### **4.2.3 Geometrical Characteristic**

The samples used in the heat transfer apparatus described in section 4.3 need to be geometrically characterized in order to know the total area of the pin fin arrays, the passages width and to determine if the samples are uniform. A non-destructive method is more appropriate in that case since the samples need to be intact to find the thermal and hydrodynamic performances. A depth of field microscope (VHX-2000, Keyence Corporation, Mississauga, Ontario, Canada), shown in Figure 4.8, was used. This microscope recomposes the features of the samples by taking images at several steps in the depth axis. With this instrument, the average height of the sample, the average base and the average passage width were measured. The average of each characteristic was calculated over 25 pin fins for each sample.



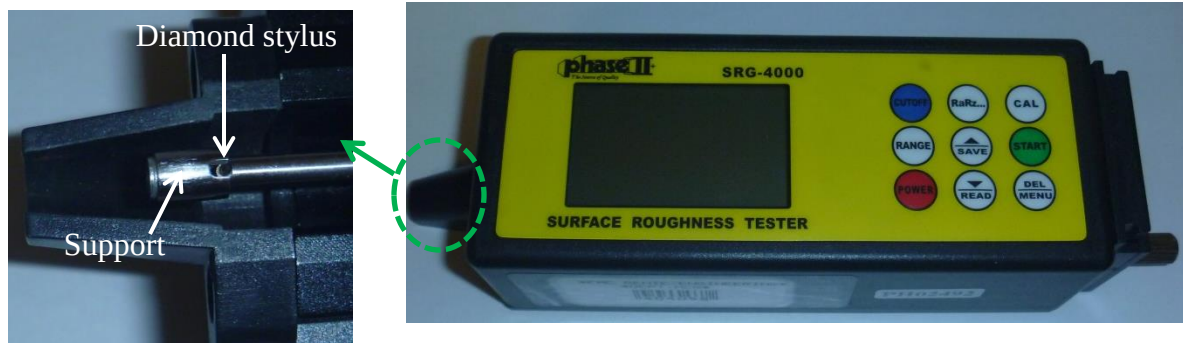


**Figure 4.8: Three-dimensional microscope Keyence VHX-2000 available at the University of Ottawa**

#### **4.2.4 Roughness**

The substrate and the bond coat roughness before spray were measured with the SRG-4000 Surface Roughness Tester contact profilometer (Phase II, Upper Saddle River, New Jersey, United States of America) shown in Figure 4.9. A diamond stylus (tip diameter of 5  $\mu\text{m}$ ) moves laterally for a specified distance while following the profile of the surface with which it is in contact. This surface profile could be quantified using several metrics with this profilometer such as the deviation from the arithmetic average, the maximum amplitude, the average of the maximum heights over the sampling length and the root mean square values. The arithmetic average of the absolute values of the profile height deviations from the mean line value was chosen as comparison parameter for the roughness value.



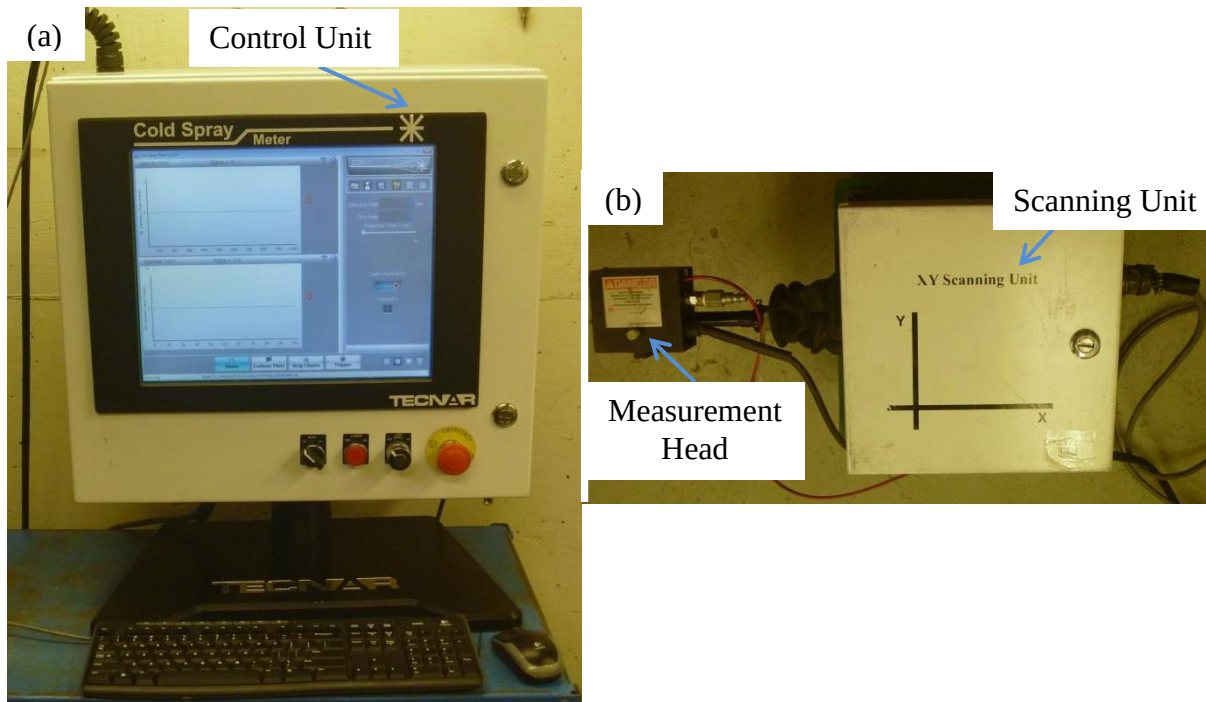


**Figure 4.9: SRG-4000 Surface Roughness Tester with a zoom view of the diamond stylus**

The profilometer mentioned previously cannot be used to determine the surface roughness after the spray since the diamond stylus support is larger than the spacing between the fins and since the fins are at an angle. In order to evaluate the roughness after the spray (when the fins are added to the substrate) for the heat transfer area, an image surface analysis method was preferred, using the three-dimensional microscope mentioned in section 4.2.3. The heat transfer area surface roughness was characterized in a two-step approach. The first step was to obtain the roughness profile of the heat transfer passage base (between the fins), using the conventional method for depth of field microscopy. In the second step, the roughness of the fins surface was obtained by tilting the microscope to the angle matching the base angle of the pyramid to obtain a perpendicular view of the fin surface. A two-dimensional correction algorithm was used to linearly correct the slope of the sample in a plane, and the roughness data was generated directly from the input image file. The total roughness was then determined to be equal to an area weighted average of the roughness of the entire flow passage.

### 4.2.5 Particle Velocity Measurement

The in-flight particle velocity was measured using a Cold Spray Meter (CSM) (Tecnar Automation Ltd., Saint-Bruno, Québec, Canada). The apparatus consists of a control unit, a planar XY scanning unit and a measurement head as shown in Figure 4.10.



**Figure 4.10: CSM apparatus: (a) The control unit (b) The scanning unit and the measurement head**

The measures were gathered in a free stream flow (without the substrate) at 15 mm from the nozzle exit. The velocity is estimated by measuring the time delay between two optical signals emitted by a particle when passing through a two slits mask with a known distance between the slits. The system has a selection algorithm which matches the intensity of the first optical signal to that of the second to ensure that the same particle is measured from one slit to the next. In this manner, a large amount of data points are collected and the system performs a statistical analysis to determine the average value and the standard deviation of the particle's flow velocity. An example of the output for aluminum particles used for the studies presented

in this research work is shown in Figure 4.11. The results of these measurements were useful to estimate the velocity input for the numerical model detailed later in this work.

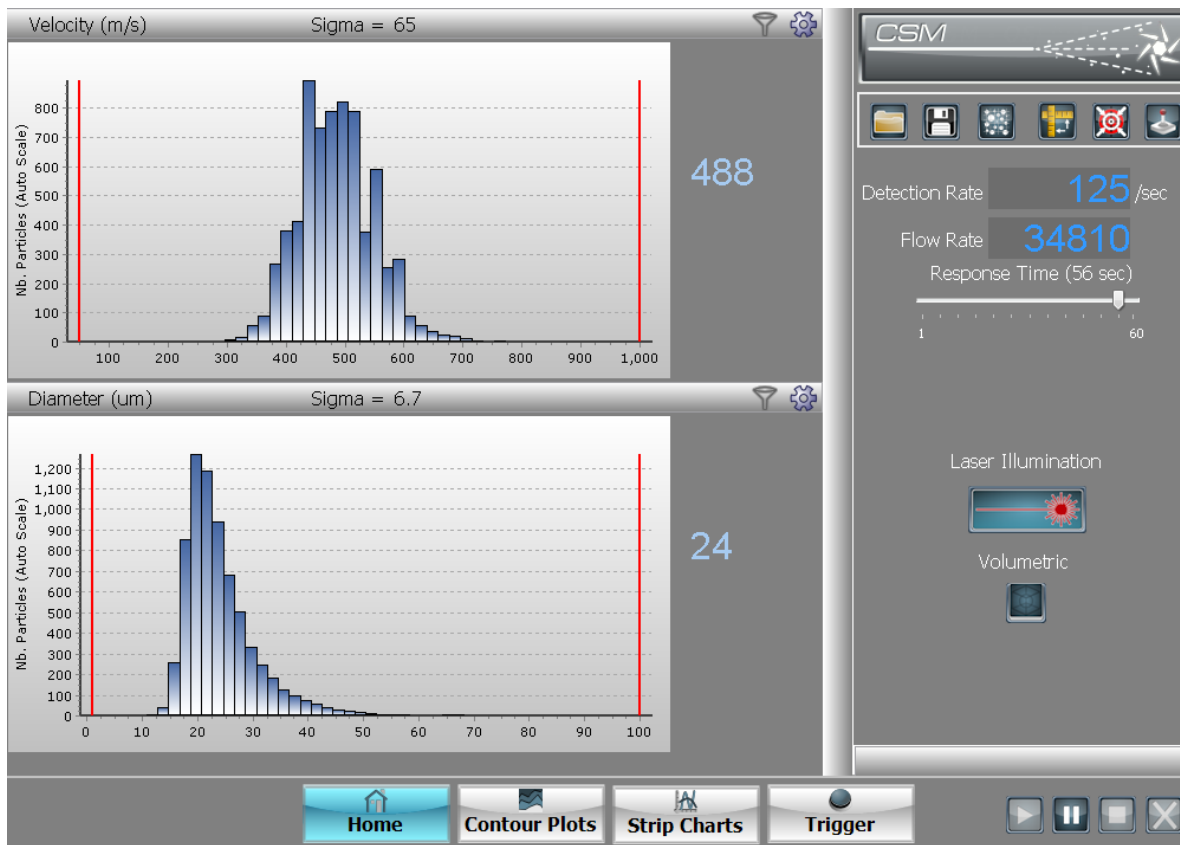
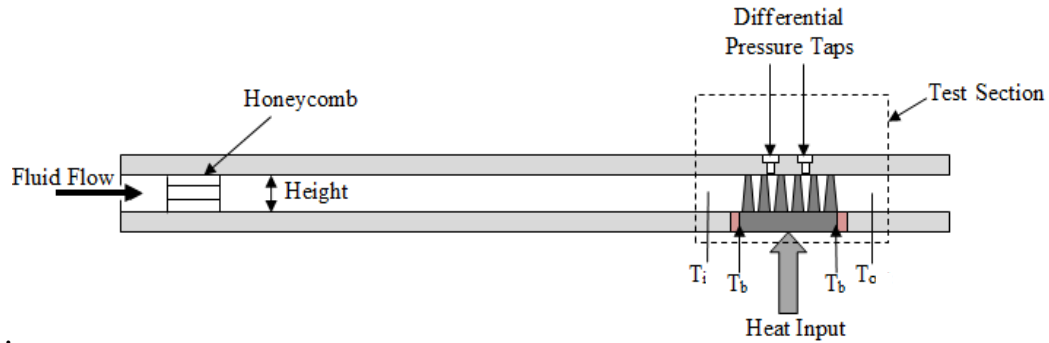


Figure 4.11: Output from the CSM for aluminum particles sprayed in this work

## 4.3 Experimental Equipment to Find Thermal and Hydrodynamic Performances

### 4.3.1 Overview

The thermal and hydrodynamic performances were investigated by an in-house apparatus developed at the University of Ottawa. A schematic of the apparatus is shown in Figure 4.12.



**Figure 4.12: Schematic of thermal and hydrodynamic test apparatus**

The apparatus is composed of a test section, a heating pad, an air supply and a data acquisition system. The air is supplied by a compressed air line equipped with a filter and dehumidifier to obtain dry room temperature air, free of oil contaminations. The air inlet flow rate is controlled by a needle valve and is measured with a flow meter. This air is smoothly expanded by a diffuser into a rectangular channel equipped with a honeycomb flow straightener. This rectangular channel is 51 mm wide by 457 mm long, with the top and bottom plates made of 19 mm thick acrylic. These plates are spaced apart using rectangular rubber gaskets, which are compressed to ensure there is no clearance between the fins and the shroud surface. These gaskets also ensure that the joint is leak-proof. A length of unobstructed flow after the honeycomb flow straightener ensures that the fluid is fully developed hydrodynamically before reaching the test section for the given hydraulic diameter and Reynolds number [14]. The test sample is subjected to a constant heat flux provided by a strip heater. A conduction block is used to ensure that heat is evenly distributed onto the substrate's surface. Finally, the flow is released to the environment after the test section.

It should be noted that all measurements were performed at steady-state and were repeated until statistical significance was achieved. Heat transfer data samples were taken at 2 Hz over a 30 second interval. Five sets of data samples were recorded for each flow rate tested.

Measurements were taken at flow rates varying between 10 and 70 SLPM, which correspond to BEC's application range.

### 4.3.2 Test Section

The most important part of the apparatus shown in Figure 4.12 is the test section since it is the zone from which the data are gathered, and will be presented in more detail. A set of differential pressure measurement taps are located 12.7 mm from the center of the test section. Two thermocouples are located 6.0 mm upstream and downstream of the test section at mid-height to record the fluid's local average temperature. Four thermocouples are attached to the fin array's base, with two on the side facing the incoming flow, and two more on the side facing the exiting flow. On each of these faces, one thermocouple is located in the middle of the width of the base, while the other thermocouple is located 6.0 mm from the wall. All the thermocouples used are T-type, butt-bonded, 28 gauge thermocouples.

All thermocouples were used to obtain useful raw data that are summarized in Table 4.1 while the pressure taps give a differential pressure over an inch of the fin array.

**Table 4.1: Data obtained with the thermocouples**

| <b>Thermocouple placement</b> | <b>Data obtained</b>       |
|-------------------------------|----------------------------|
| 6.0 mm upstream               | Inlet air temperature      |
| 6.0 mm downstream             | Outlet air temperature     |
| Fin base facing flow          | Base fin front temperature |
| Fin base exiting flow         | Base fin rear temperature  |

### 4.3.3 Analysis Method

When the raw data shown in section 4.3.2 are obtained, the following method is used in order to get processed data. Hence, to analyze the raw data obtained, flow conditions are expressed using the Reynolds number based on the flow hydraulic diameter, which is given by:

$$Re_{Dh} = \frac{\rho_{fl} u_{max} D_h}{\mu} \quad (4.1)$$

where  $\rho_{fl}$  is the fluid's density,  $u_{max}$  is the maximal fluid velocity in the fin array,  $D_h$  is the hydraulic diameter of the fin array flow passage and  $\mu$  is the fluid's dynamic viscosity. The variable  $D_h$  is given by:

$$D_h = \frac{4 \cdot A_{flow}}{P_{flow}} \quad (4.2)$$

where  $A_{flow}$  and  $P_{flow}$  are the flow channel's face area and perimeter, respectively.

The heat transfer performance is shown using the thermal conductance that can be calculated using:

$$UA = \frac{1}{R_{eq}} = h \cdot A_t \cdot \eta_o \quad (4.3)$$

where  $R_{eq}$  is the thermal circuit's equivalent resistance,  $h$  is the convective heat transfer coefficient,  $\eta_o$  is the overall surface efficiency which is an indicator of the performance of the extended surface and  $A_t$  is the total heat transfer area of the exposed fin and base surface. More details are given in section 2.1.2.3 on the calculation of  $\eta_o$  and  $A_t$  for pin fin applications.

The fin array's convective heat transfer coefficient and Nusselt number are calculated using the following relations:

$$h = \frac{q}{\Delta T_{lm} A_t \eta_o} \quad (4.4)$$

$$Nu_{Dh} = \frac{h \cdot D_h}{k_{fl}} \quad (4.5)$$

In these equations,  $k_{fl}$  is the thermal conductivity of the fluid,  $q$  is the heating rate and  $\Delta T_{lm}$  is the log-mean temperature difference between the fluid and the fin array. The log-mean temperature difference is calculated with:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (4.6)$$

where  $\Delta T_1$  and  $\Delta T_2$  are the temperature difference between the fin surface and the flow at the inlet and outlet, respectively. The total heat input rate to the system is calculated using:

$$q = \dot{M} \cdot (T_o - T_i) \cdot c_{p_{fl}} \quad (4.7)$$

where  $\dot{M}$  is the fluid mass flow rate,  $(T_o - T_i)$  is the fluid temperature difference between the entrance and the exit of the test section,  $c_{p_{fl}}$  is the fluid's thermal capacitance.

To determine which fin array transfers heat more efficiently, an efficiency indicator must be selected. Based on the literature review performed, the proposed method in Sahiti's work [40] is selected. This method is based on plotting the thermal and hydrodynamic data as a curve of thermal conductance as a function of the required pumping power for different flow conditions. This yields curves which can be directly compared one to another, with the most efficient fin array having a higher thermal conductance at a given pumping power. The pumping power for a given pressure loss was calculated according to Sahiti's definition [36]:

$$e = \frac{\dot{V} \cdot \Delta P_f}{\eta_{fan}} \quad (4.8)$$

In the previous relation,  $e$  is the pumping power,  $\dot{V}$  is the volumetric flow rate of the fluid,  $\Delta P_f$  is the pressure drop due to fins while  $\eta_{fan}$  is the fan efficiency which is selected to be 0.8 [36]. The pumping power is divided either by the fin array base (subscript  $b$ ) to yield the pumping power per unit area or by the volume (subscript  $v$ ) of the fin array to obtain the pumping power per unit volume.

Also, another important number in the fin array's field is the friction factor which is a number that predict the frictional energy loss based on the velocity of the flow and the resistance losses. To obtain a friction factor for the fin array, the pressure losses through the fin array must be converted with the following equation:

$$f = \frac{2 \cdot \Delta P_f}{N_r \cdot \rho_{fl} \cdot u_{max}^2} \quad (4.9)$$

where  $N_r$  is the number of rows.

Several graphs will be produced from this analysis in order to be able to analyze and to compare the pin fin arrays produced by CGDS. The graphs that will result from this experimental and analysis work are explained in Table 4.2.

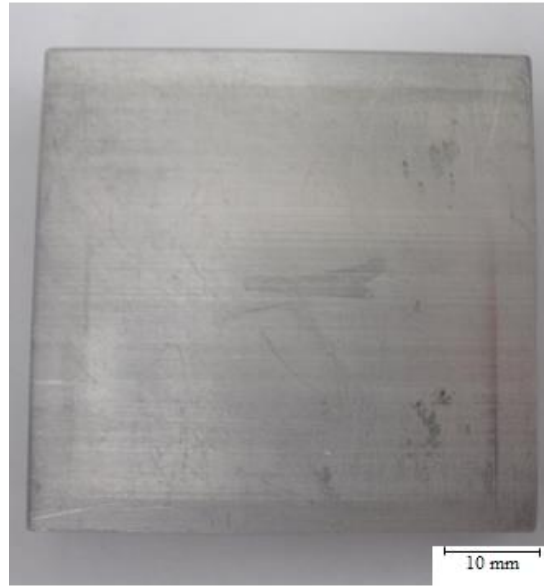


**Table 4.2: Graphs to be produced and their purpose**

| <b>Graph</b>    |                | <b>Purpose</b>                                                                                                                                                                                                                                     |
|-----------------|----------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| <u>Ordinate</u> | <u>Abcissa</u> |                                                                                                                                                                                                                                                    |
| $UA$            | $Re_{Dh}$      | Higher $UA$ at a given Reynolds number on that graph will give which one is better in terms of heat transfer compare the others. The difference in heat transfer could result from the increase in $\eta_o$ and/or $h$ and/or $A_t$ in this graph. |
| $h$             | $Re_{Dh}$      | This graph gives an indication, if there is a difference in the flow characteristics since $h$ is separated from $A_t$ in this graph.                                                                                                              |
| $Nu_{Dh}$       | $Re_{Dh}$      | Dimensionless Nusselt number versus Reynolds number will be plotted in order to be able to find correlation between these two numbers for the parameters studied.                                                                                  |
| $\Delta P_f$    | $Re_{Dh}$      | This graph will compare the pressure drop per inch of fin array at given Reynolds number.                                                                                                                                                          |
| $f$             | $Re_{Dh}$      | Dimensionless friction factor versus Reynolds number in order to find correlation between these two dimensionless numbers to be able to compare with the other fin array shapes in the literature.                                                 |
| $UA_v$          | $e_v$          | This graph will give which fin array is more efficient than the others per unit volume which is very important for compact heat exchanger application.                                                                                             |
| $UA_b$          | $e_b$          | This graph will give which fin array is more efficient than the others per unit footprint area which is important when there is no limitation on the height of the sample.                                                                         |

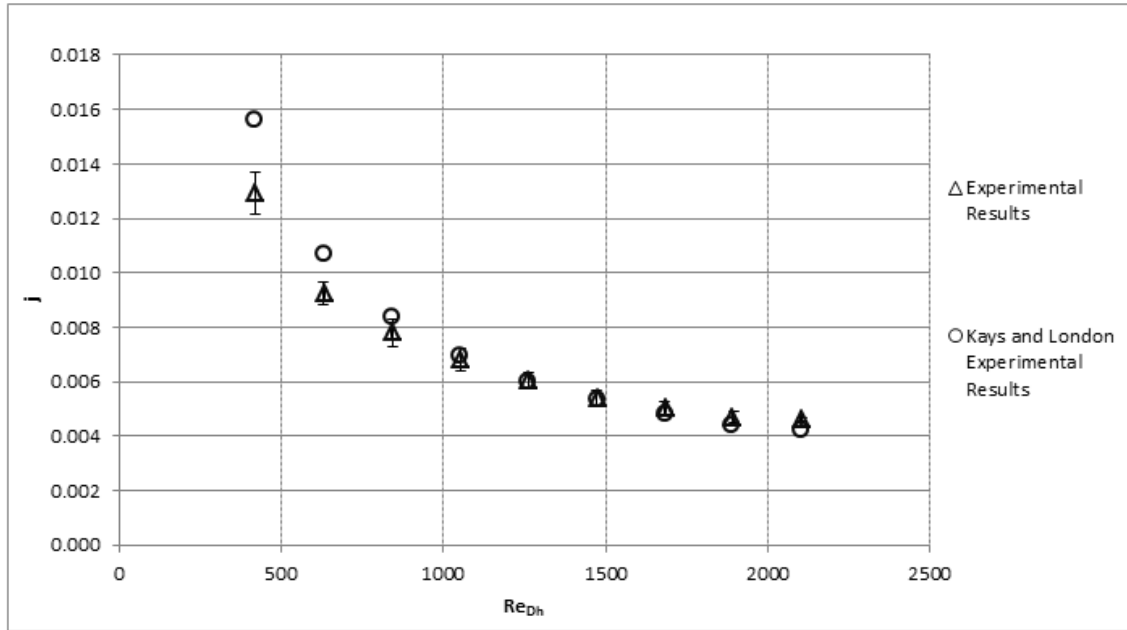
#### 4.3.4 Accuracy and Precision

The accuracy and the precision of this in-house apparatus were tested. The accuracy was verified against correlations produced by Kays and London [15]. A flat plate (a sample without the fin array on the top), as shown in Figure 4.13, was placed in the apparatus and the heat transfer data was found and compared with the Kays and London data [15] as well as the pressure drop associated with a flat plate in a rectangular channel.



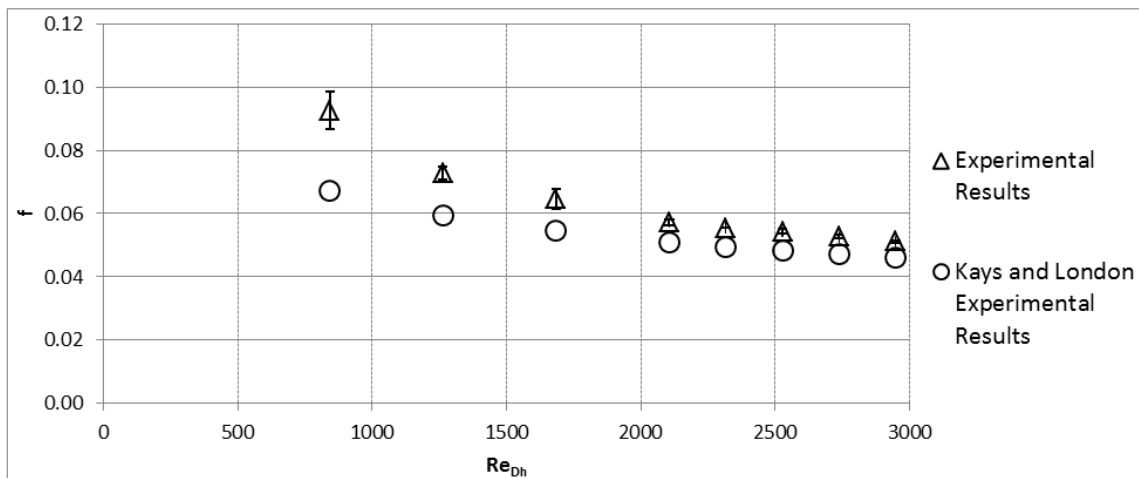
**Figure 4.13: Flat plate used to confirm the accuracy of the test apparatus**

Figure 4.14 shows the graph produced in order to verify the accuracy of the in-house apparatus for the heat transfer. The results obtained using the test fixture shows a high similarity with published results found in literature, with higher deviation at low flow rates. The deviation at low flow rates can be explained by the fact that the instrument uncertainty is higher at low flow rates due to the full scale measurement uncertainty. It should be noted that the channel height was set to 2.0 mm for these tests.



**Figure 4.14: Colburn factor vs. Reynolds number based on the hydraulic diameter**

The accuracy on the pressure drop is shown in Figure 4.15 as friction factor vs. Reynolds number. The error is higher at low flow rates since the pressure drop for a flat plate at these low flow rates are close to the lower limit of the apparatus. However, for straight cut fin arrays and pyramidal fin arrays, this is not an issue since the pressure drop encountered is significantly higher compared to a flat plate.



**Figure 4.15: Friction factor vs. Reynolds number based on the hydraulic diameter**

The precision of the apparatus was confirmed by repeating the experiment at least five times for each data point and the deviation from these values was calculated using the root-square method [101]. Deviation of the measurements from the mean value less than 5% was obtained. Therefore, the apparatus is considered to be precise as this type of uncertainty is usually acceptable in this field [6].

## 4.4 Mechanical Properties Characterization

As mentioned previously the shear strength evaluation is an important parameter to investigate in order to know if the pin fin arrays are viable to be used in industrial applications. For this reason, a shear test should be selected. The adhesion/cohesion strength of the pin fin array was quantitatively characterized using the European test method EN15340-2007: Determination of shear load resistance of thermally sprayed coatings [102]. This standard consists of applying an increasing shear load until the coating breaks. A schematic of the standard test method is shown in Figure 4.16.

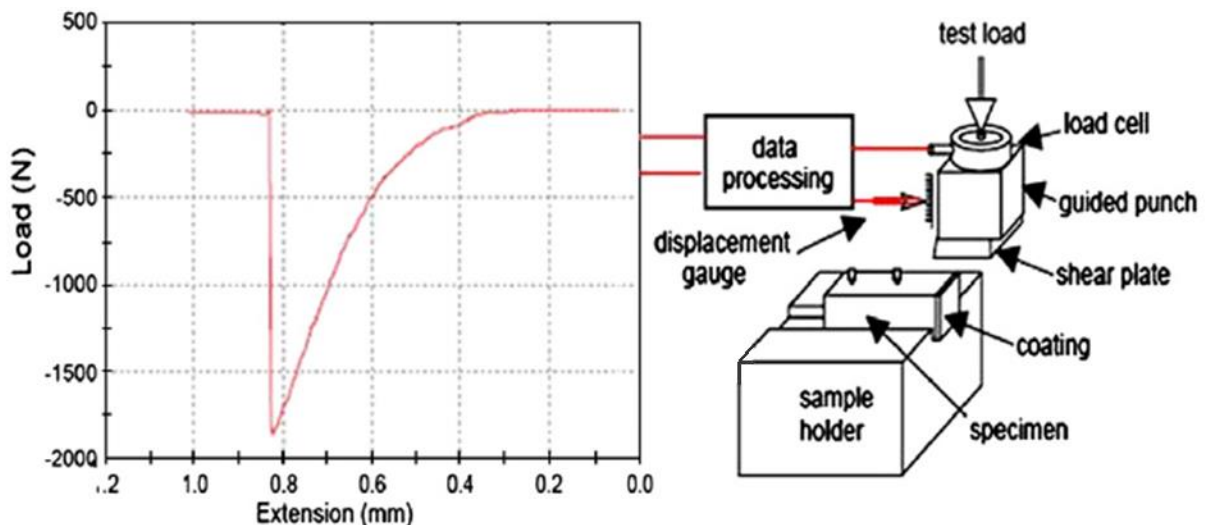


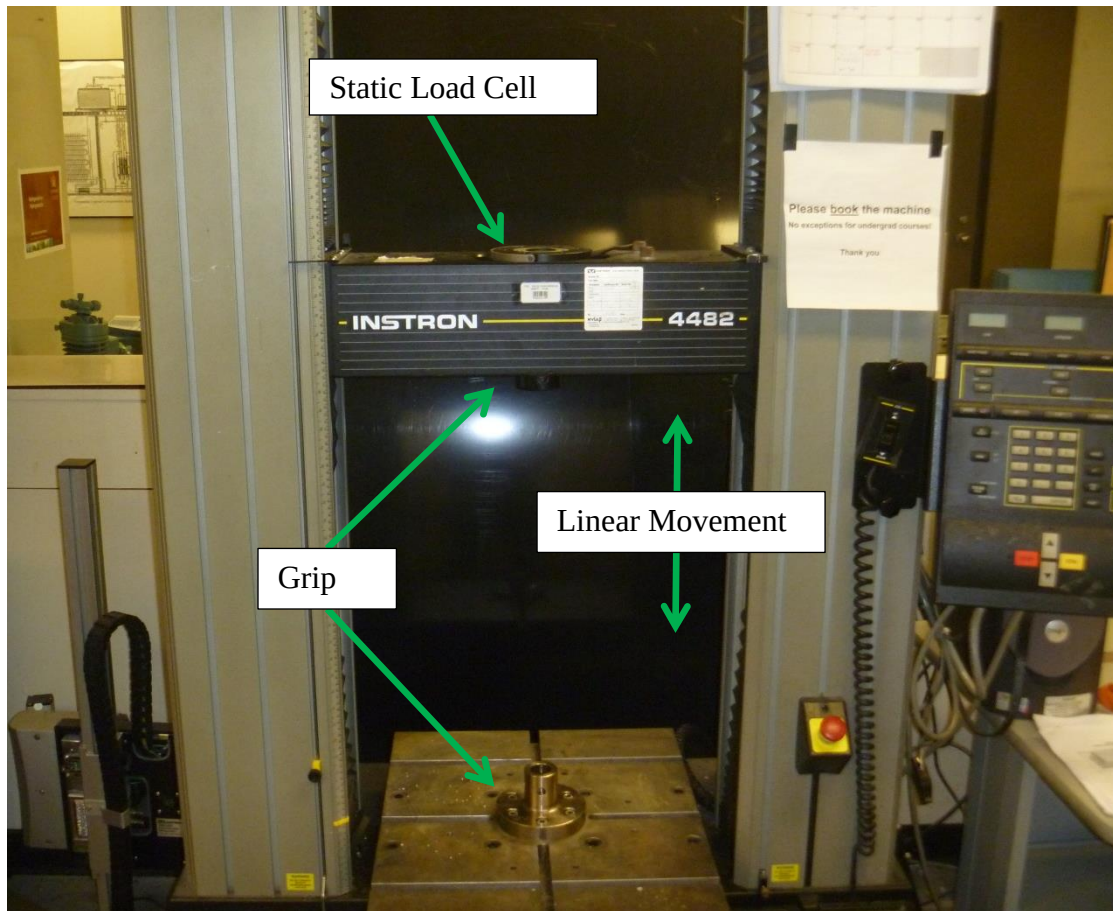
Figure 4.16: Schematic of the apparatus used for the EN15340 with a typical resulting graph [103]

The failure of the coating could be into one of the following three modes:

1. Adhesion > Cohesion : The coating delaminates under the shear stress and its base stays on the substrate
2. Adhesion = Cohesion: The coating delaminates while its base tear off from the substrate
3. Adhesion < Cohesion: The base of the coating tears off from the substrate while the coating itself stays intact

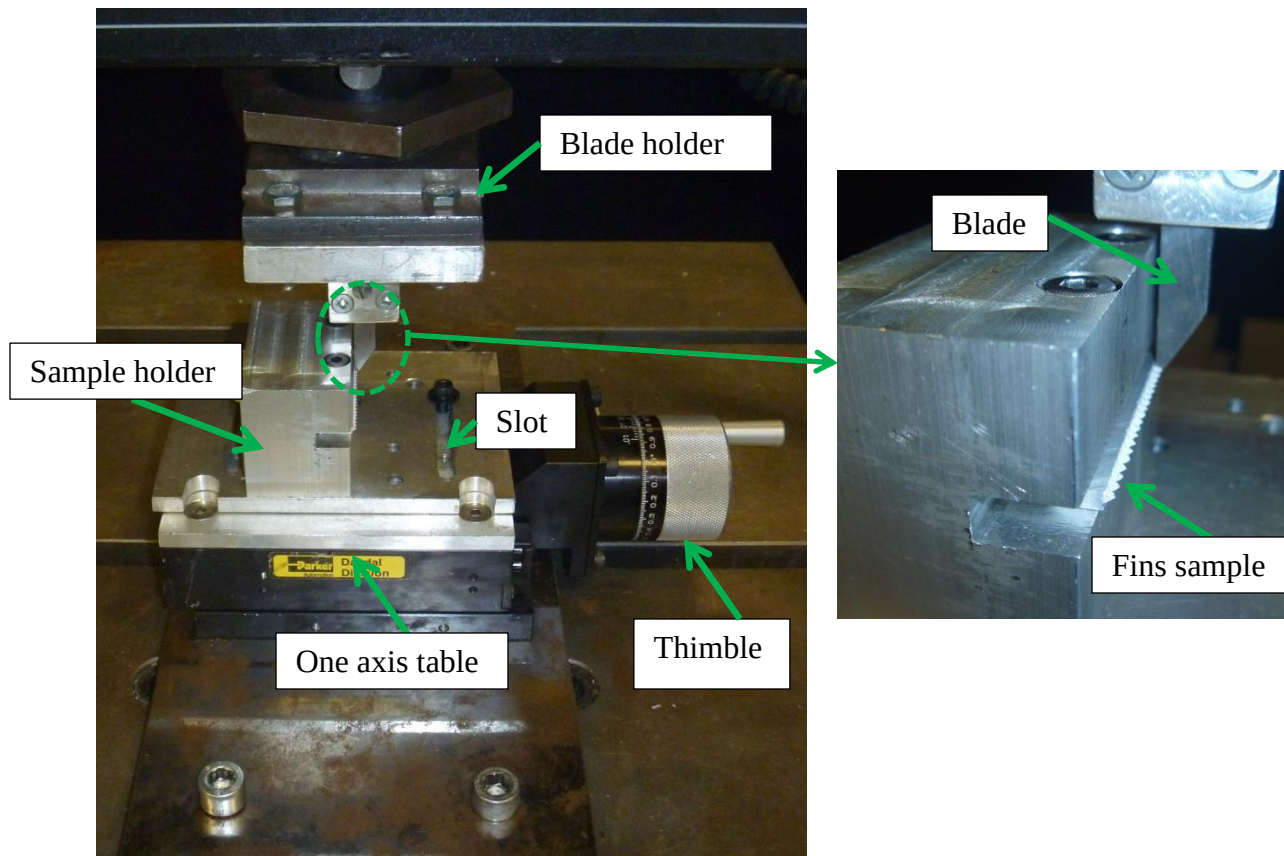
It should be noted that the adhesion of a coating is defined as the adherence of the coating on the substrate while the cohesion is defined as the adherence of the coating between each sprayed layer. The data were reported with the shear strength and the failure mode observed for each trial.

An in-house test apparatus that respects the standard was designed and used. The force is applied by a commercially available Instron, (High Wycombe, Buckinghamshire, United Kingdom), model 4482 equipped with a static load cell of 100 kN as shown Figure 4.17.



**Figure 4.17: Instron 4482 at University of Ottawa**

The in-house test apparatus is attached to the Instron in order to transmit the load to the coating as depicted in Figure 4.18. A blade holder is attached to the upper grip of the Instron while the lower grip is removed and replaced by a moving table. The linear movement of the blade holder was set to  $50 \mu\text{m/s}$ , as specified in the standard [102]. Consequently, when the machine touches the coating, it increases the load in order to keep that linear speed which, eventually breaks the pin fins. During the test, the maximum load recorded was taken as the shear load resistance. The shear strength is determined by dividing that shear load resistance by the pin fins' sheared area. The latter is obtained with the three-dimensional optical microscopy shown in section 4.2.3.



**Figure 4.18: In-house shear test apparatus**

The shear distance was set, with the use of the thimble, to 50  $\mu\text{m}$  from the coating/substrate interface. The reported shear strength values are the average of 15 tests of 3 pin fins for each sample tested. Shear strengths of pin fins with different heights and fin densities were evaluated. This study also focused on exploring a method that could increase the shear strength, such as by grit-blasting the substrate before spraying to potentially strengthen the mechanical bonding at the substrate/coating interface. Another trial was performed on the fins deposited on the bond coat, since for BEC's application, the pin fins will be sprayed on a bond coat deposited by arc spray, and it is important to determine how the shear strength changes between samples directly sprayed on the bulk aluminum 6061-T6 and other fin array samples deposited on a bond coat layer. Finally, the pin fins were deposited on a bond coat

produced by cold spray with the purpose to see if better mechanical properties are obtained by producing the bond coat directly by a layer of CGDS instead of producing it with arc spray.

It should be noted that the samples were manufactured with the optimized parameters, which yield fin arrays with clean passages and low levels of porosity. The fracture surfaces resulting from these shear tests were observed under the SEM (section 4.2.2) in order to understand physically what factors contribute to change the shear strength of the pin fin array.

## **4.5 Streamwise Anisotropy Study**

Certain pin fin arrays were sprayed using only one material, while others were sprayed with different materials along the length of the heat exchanger. These samples were tested and compared with the in-house heat transfer apparatus described in section 4.3. Consequently, these tests were to tentatively prove that using a streamwise anisotropic fin array is more thermally efficient than the traditional single material pin fin arrays. According to the graph presented in the literature review section 2.1.5, three materials were sprayed for the proof of concept which are: stainless steel 304 (high efficient service temperature), commercially pure nickel (intermediate efficient service temperature) and aluminum (low efficient service temperature).

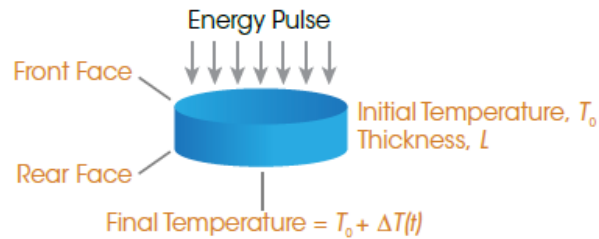
A laser flash diffusivity testing apparatus, shown in Figure 4.19, (DLF-1300, TA Instruments, New Castle, Delaware, United States of America) was used at room temperature to determine the thermal conductivity of the cold sprayed materials.





**Figure 4.19: DLF-1300 at TA Instruments laboratory [104]**

This apparatus is used to find the thermal diffusivity and the specific heat capacity on samples with a diameter of 12.7 mm. A laser source heats the sample while the two faces are monitored in terms of temperature (Figure 4.20).



**Figure 4.20: Schematic of the laser flash diffusivity process [104]**

The thermal diffusivity is measured by the following procedure; the time response to reach half of the maximal temperature increase ( $t_{1/2}$ ) is found using a thermogram similar to Figure 4.21 and the thermal diffusivity is determined by [104]:

$$\alpha_m = 0.1388 \frac{L^2}{t_{1/2}} \quad (4.10)$$

where  $\alpha_m$  is the thermal diffusivity of the material,  $L$  is the sample thickness and  $t_{1/2}$  is half of the maximal temperature increase.

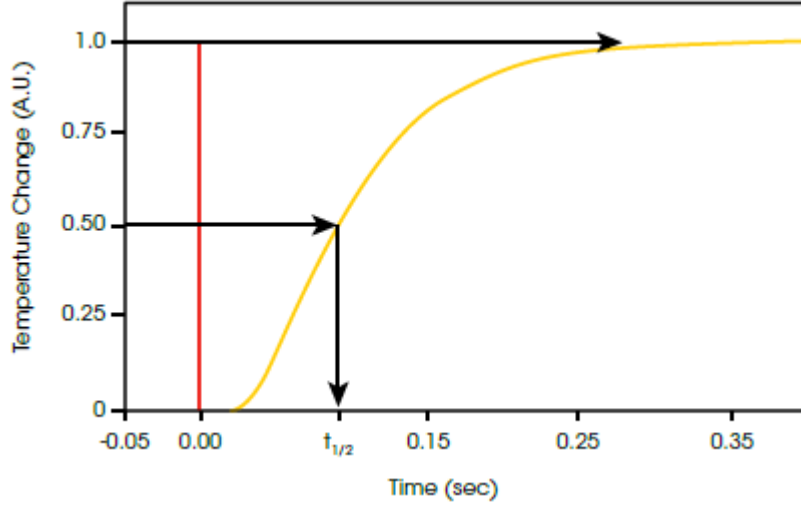


Figure 4.21: Example of a thermogram produced by laser flash [104]

The specific heat capacity is also determined using the laser flash analysis [105]. The relationship to determine the specific heat capacity is given by:

$$c_{p_m} = \frac{aAp}{M\Delta T_{max}} \quad (4.11)$$

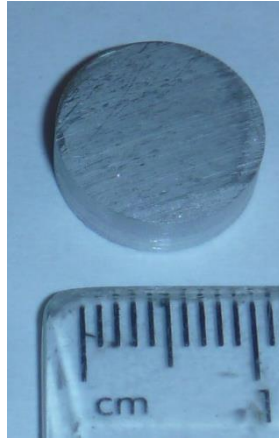
where  $c_{p_m}$  is its specific heat capacity,  $a$  is its absorptivity,  $A$  is its surface area,  $p$  is the energy density of the laser pulse,  $M$  is its mass and  $\Delta T_{max}$  is the maximum temperature rise.

Finally, the thermal conductivity is related to the thermal diffusivity, the specific heat capacity and the density by the following equation:

$$\alpha_m = \frac{k_m}{\rho_m \cdot c_{p_m}} \quad (4.12)$$

where  $k_m$  is its thermal conductivity and  $\rho_m$  is its density. The samples were weighed and the volume was measured by water volume displacement to obtain the coating density.

Cold sprayed samples of each material were produced and analyzed at the TA Instruments laboratory located in New Castle, Delaware. The aluminum sample sent to TA Instruments is shown in Figure 4.22.



**Figure 4.22: Aluminum coupon produced for analysis by DLF-1300**

It is important to find the thermal conductivity of the different cold sprayed materials since it differs from the bulk material [106]. The observed value is usually much lower than that of the bulk material due to the amount of dislocations created by the cold spray process, which shortens the mean free path for electron movement, but also due to other contributing factors such as the quality of the inter-particle bonding and the presence of surface oxides in the powder, which become imbedded in the coating [106–109].

Also, the streamwise anisotropic samples were submitted to shear tests in order to see if the shear strength values are still superior than those required for the application of this technology and if they change depending on the feedstock materials.

## **4.6 Numerical Simulation**

The objective of this section is to present the model details and modeling tools used in Abaqus/Explicit to estimate the particle and substrate deformation during a cold spray impact.

#### 4.6.1 Shock Wave and Elasticity

For high energy impacts, such as those encountered by bullets in the study of ballistics, the hydrodynamic pressure is calculated with the Mie-Grüneisen linear Equation Of State (EOS) instead of the linear elastic part of stress-strain curve (Young's modulus). Due to the impact, shock waves are propagated through the material, creating an abrupt change of pressure, density, material speed and consequently of the strain.

The Mie-Grüneisen equation of state is used to determine the pressure/strain in a shock compressed solid. An equation of state assumes the pressure ( $P_m$ ) is function of the density ( $\rho_m$ ) and the internal energy per unit mass ( $e_m$ ), as seen below:

$$P_m = f(\rho_m, e_m) \quad (4.13)$$

Whereas a linear equation of state is in the following form:

$$P_m = f(\rho_m) + g(\rho_m)e_m \quad (4.14)$$

The Mie-Grüneisen equation consists of a special form of the Grüneisen model that describes the effect of the change of volume on the vibrational properties of crystal lattices. In other words, how the change in density will affect the internal energy of the material. The Grüneisen model is given by:

$$\Gamma = V \left( \frac{dP}{dE} \right)_V \quad (4.15)$$

where  $\Gamma$  is the Grüneisen parameter,  $V$  is the volume,  $P$  is the pressure and  $E$  is the internal energy. Assuming that the Grüneisen parameter is independent of  $P$  and  $E$ , it can be rewritten as follows, when integrated:

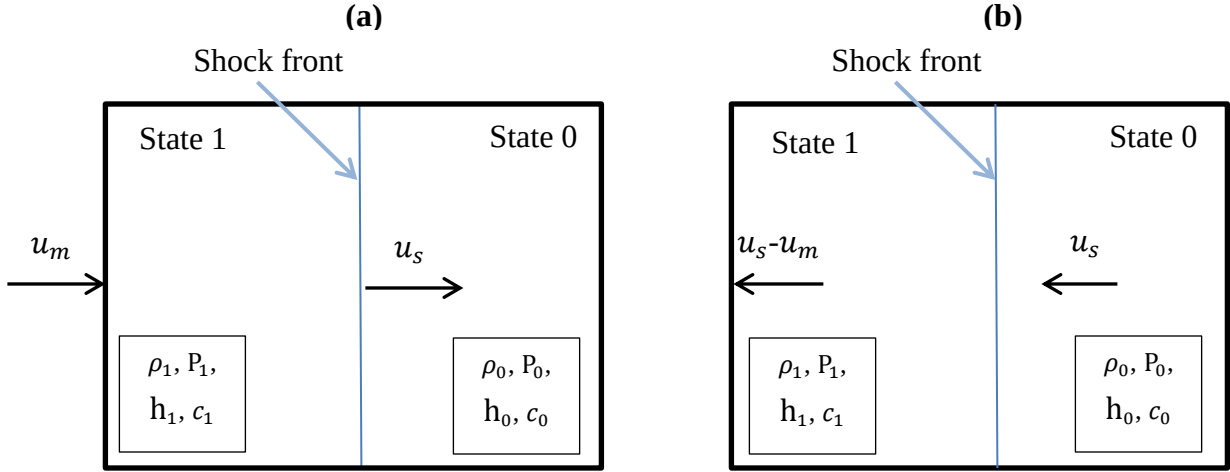
$$P - P_0 = \frac{\Gamma}{V} (E - E_0) \quad (4.16)$$

where  $P_0$  and  $E_0$  are respectively the pressure and internal energy at a reference state. The reference is taken from a model that expressed the pressure dependence on the density and internal energy, which does not take into account how the change in volume affects the vibration of the lattice (cold curve). In the case of the Mie-Grüneisen model, the reference (cold curve) is estimated by the Hugoniot equations.

$$P_m - P_H = \frac{\Gamma}{V} (E_m - E_H) \quad (4.17)$$

where  $P_H$  and  $E_H$  are respectively the pressure and internal energy of the Hugoniot equations.

The Hugoniot equations are developed by respecting the three following conservation equations for a shock compressed material: mass, momentum and energy. When a shock wave propagates through the material, the shock is moving at certain speed ( $u_s$ ) while the material is compressed due to the pressure behind the shock at another speed ( $u_m$ ), as schematized in Figure 4.23 (a). However, the Hugoniot equations are developed using a moving reference frame (the shock front), therefore the velocities, from the point of view of the shock wave, can be expressed as in Figure 4.23 (b). The material behind the shock wave is moving at  $u_s - u_m$  while the material in front of the shock wave is moving at  $u_s$  from the point of view of the shock wave, which is considered immobile using this point of view.



**Figure 4.23: Shock wave propagation through the material with: (a) Static frame (b) Moving frame**

Therefore, assuming steady state and the area affected by the shock wave is the same before and after this shock wave, the conservation equations could be expressed as:

Mass conservation: 
$$\frac{\dot{M}}{A} = \rho_0 u_s = \rho_1 (u_s - u_m) \quad (4.18)$$

Momentum conservation: 
$$P_1 - P_0 = \frac{\dot{M}}{A} (u_m) \quad (4.19)$$

Energy conservation  
(1<sup>st</sup> law of thermodynamics): 
$$0 = \dot{M} (h_1 - h_0 + \frac{1}{2} (u_s - u_m)^2 - \frac{1}{2} (u_s)^2) \quad (4.20)$$

where  $\dot{M}$  is the mass flow rate,  $A$  is the area and  $h$  is the specific enthalpy. Also, it was determined experimentally that the relationship between the speed of the shock wave and the material is linear and is given by [110]:

$$u_s = c_0 + s \cdot u_m \quad (4.21)$$

where  $c_0$  is the speed of sound at an uncompressed state while  $s$  is the Hugoniot slope.

By using the four equations above, it is possible to express the Hugoniot energy and the Hugoniot pressure needed in the Mie-Grüneisen equation.

From the mass, momentum and energy, the Hugoniot energy is:

$$e_H = \frac{P_H \eta}{2\rho_0} \quad (4.22)$$

where the compressive strain  $\eta$  is given by  $\eta = 1 - \rho_0/\rho_1$  and  $P_H$  is the Hugoniot pressure. It should be noted that the demonstration of the Hugoniot energy is presented in Appendix A.

From the mass, momentum and experimental relationship, the Hugoniot pressure is:

$$P_H = \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} \quad (4.23)$$

The demonstration of the Hugoniot pressure is shown in Appendix B.

Using these two equations in the Mie-Grüneisen equation (4.17) and the relation from the Grüneisen model ( $\Gamma = \frac{\Gamma_0 \rho_0}{\rho_1}$ ), the Mie-Grüneisen equation is expressed in the linear equation of state form [110]:

$$P_m = \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} \left( 1 - \frac{\Gamma_0 \eta}{2} \right) + \Gamma_0 \rho_0 e_m \quad (4.24)$$

The following equation is the equation implemented in the model, where  $\rho_0$ ,  $c_0$ ,  $s$  and  $\Gamma_0$  are material parameters. The demonstration of the Mie-Grüneisen equation expressed in the linear equation of state form is presented in Appendix C.

The equation above only gives the hydrodynamic stress (which causes volume changes). The deviatoric stress (which induces shape changes) is given by the following equation [110]:

$$d = 2G_e \epsilon_d \quad (4.25)$$

where  $d$  is the deviatoric stress,  $G_e$  is the elastic shear modulus while  $\epsilon_d$  is the elastic deviatoric strain.

#### 4.6.2 Plasticity

The plastic deformation of the particles and the substrate were modeled using the Preston-Tonks-Wallace (PTW) model, as to date; it is the most accurate model to predict cold spray deformation, as mentioned in section 2.2.2.3 of the literature review. According to the definition given by Preston et al. [82], the flow stress ( $\sigma$ ) is defined as:

$$\sigma = 2 \left[ \hat{\tau}_s + \alpha \ln \left[ 1 - \phi \exp \left( -\delta - \frac{\theta \epsilon_p}{\phi \phi} \right) \right] \right] G_p \quad (4.26)$$

$$\phi = \frac{s_0 - \hat{\tau}_y}{n}, \quad \delta = \frac{\hat{\tau}_s - \hat{\tau}_y}{\alpha}, \quad \phi = \exp(\delta) - 1 \quad (4.27)$$

where  $\hat{\tau}_s$  is the normalized work hardening saturation stress,  $\hat{\tau}_y$  is the normalized yield stress,  $\theta$  is strain hardening rate,  $\epsilon_p$  is the plastic strain,  $s_0$  is the saturation stress at 0 K and  $n$  is the strain hardening constant. The plastic shear modulus ( $G_p$ ) is function of temperature and it is defined by this equation from the Mechanical Threshold Stress (MTS) model [111]:

$$G_p(T) = G_0 - \frac{B}{\exp\left(\frac{J}{T}\right) - 1} \quad (4.28)$$

where  $G_0$  is the shear modulus at 0 K,  $B$  is a material constant,  $J$  is a temperature material constant and  $T$  is the node temperature.

The equations to determine the work hardening saturation stress ( $\hat{\tau}_s$ ) and yield stress ( $\hat{\tau}_y$ ) are given below [85]:

$$\hat{\tau}_s = \max \left\{ s_0 - (s_0 - s_\infty) \operatorname{erf} \left[ \kappa \hat{T} \ln \left( \frac{\gamma_\xi^\xi}{\dot{\epsilon}_p} \right) \right], s_0 \left( \frac{\dot{\epsilon}_p}{\gamma_\xi^\xi} \right)^\beta \right\} \quad (4.29)$$

$$\hat{\tau}_y = \max \left\{ y_0 - (y_0 - y_\infty) \operatorname{erf} \left[ \kappa \hat{T} \ln \left( \frac{\gamma_\xi^\xi}{\dot{\epsilon}_p} \right) \right], \min \left\{ y_1 \left( \frac{\dot{\epsilon}_p}{\gamma_\xi^\xi} \right)^{y_2}, s_0 \left( \frac{\dot{\epsilon}_p}{\gamma_\xi^\xi} \right)^\beta \right\} \right\} \quad (4.30)$$



where  $\hat{T} = T/T_m$ ,  $T_m$  is the melting temperature,  $s_\infty$  is the saturation stress near the melting temperature,  $\kappa$  is the temperature dependence constant,  $\gamma$  is the strain rate dependence constant,  $\dot{\epsilon}_p$  is the plastic strain rate,  $\beta$  is the high strain rate exponent,  $y_0$  is the yield stress constant at 0 K,  $y_\infty$  is the yield stress constant near melting temperature,  $y_1$  is the medium strain rate constant and  $y_2$  is the medium strain rate exponent. The two definitions in the maximum function are used to model both low and high strain rate regimes, respectively. For the lower strain rates (where stresses are governed by thermal activation), the first definition of the maximum function is used whereas the second definition is used for the high strain rate regime (which is governed by dislocation drag mechanisms).

The material parameter ( $\dot{\xi}$ ) in equations 4.29 and 4.30 is defined as [85]:

$$\dot{\xi} = \frac{1}{2} \left( \frac{4\pi\rho_m}{3M_a} \right)^{1/3} \left( \frac{G_p(T)}{\rho_m} \right)^{1/2} \quad (4.31)$$

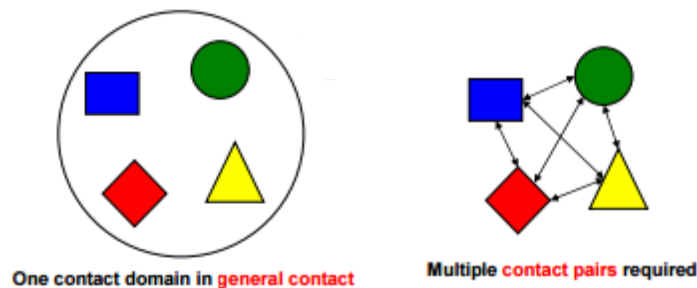
where  $M_a$  is the atomic mass.

It should be noted that the PTW model and the MTS shear behavior are not included in Abaqus. Consequently, these models were incorporated in the finite element analysis by a user subroutine (VUHARD).

### 4.6.3 Layout of the Model

A three-dimensional Finite Element Analysis (FEA) model using Abaqus/Explicit was developed and used to simulate the impact behavior of cold sprayed particles on a substrate. In order to reduce the computational time, the three-dimensional problems were simplified into half symmetric models. Hexahedral 8-node reduced integration thermally coupled elements (C3D8RT) were employed to construct the mesh for both the substrate and the

particles. A reduced integration means there is less integration point by element which yields to lower computational time. It is suggested to use a reduced integration when plasticity is involved since this type of integration produced a less stiff element [112]. A thermally coupled element means that the model will solve simultaneously the stress and heat transfer equations (coupled thermal-stress analysis). The mesh size near the contact area was inputted as 1/50 of the particle diameter [113]. The severe distortion of the mesh, which occurs for particles and substrates during high velocity impacts, was mitigated by setting the distortion control parameter to 0.4. The distortion control parameter prevents solid elements from inverting or distorting excessively. The number 0.4 means that the constraints activate when an element undergoes 60% of nominal strain (distortion length ratio) [114]. Constraints are enforced using a penalty approach [110]. The penalty method is an algorithm for constrained optimization problem. The contact interaction at the particle/substrate and particle/particle interfaces was represented by a general contact condition with a coefficient of friction of 0.5. A general contact allows all regions (bodies) of the model to be in contact whereas with contact pairs condition the surfaces in contact need to be specified as schematized in Figure 4.24 [115]. Again, the constraints are enforced by using a penalty method [115].



**Figure 4.24: Different types of contact in Abaqus [115]**

The initial particles velocity was assumed to be the average value measured by CSM (section 4.2.5) while the initial substrate and particle temperature were assumed to be 400 K [116].

The shape of the particle used in the model was determined by an approximation of the particle images taken by SEM (section 4.2.2).

## Chapter 5 - Results

This chapter consists of six peer reviewed papers published in international journals relevant to the work. Four manuscripts have been published in the Journal of Thermal Spray Technology while the two other have been published in the International Journal of Heat and Mass Transfer. As it is possible to see below, these papers are intended to fulfill the objectives of this thesis presented in Chapter 3.

### 5.1 Paper #1: Proof of Concept

**This paper was reproduced according to the copyright agreement signed with ASM International.**

The paper that follows, entitled “Net Shape Fins for Compact Heat Exchanger Produced by Cold Spray” was the first published paper on this novel type of fins manufactured using the cold spray process. Therefore, this paper was intended to show the proof of concept for the production of this new type of fins, as well as to demonstrate the accuracy and the precision of the in-house heat transfer and pressure drop apparatus developed at the University of Ottawa. Near-net shaped pyramidal fin arrays were successfully produced with clean passages and low porosity. An in-house apparatus was demonstrated to have good accuracy and precision by reproducing an experiment performed by Kays and London [15]. This paper addresses the objectives of the feasibility study of using the CGDS as a process to manufacture discontinuous fin arrays, the investigation of the cold spray parameters and the comparison of the new fin types with the continuous fins presently used at BEC with thermal and hydrodynamic tests.

Some preliminary works are not included in this paper such as the type of masks tested in order to find a successful candidate that meets the requirements given in section 3.2. The steel

wire mesh was chosen as the best candidate for our application. Steel machined mask and polymer mask were also tested. However, these types of mask have presented several drawbacks. The steel machined masks are costly to produce, but it was possible to produce fin arrays with them. For the polymer masks, they were not able to withstand the flow from the cold spray process used to deposit the particles. The polymer masks either melt, tear or the particles pierce them and consequently, no fin array was made using this type of masks.



# Net Shape Fins for Compact Heat Exchanger Produced by Cold Spray

Yannick Cormier, Philippe Dupuis, Bertrand Jodoin, and Antoine Corbeil

(Submitted April 29, 2013; in revised form June 25, 2013)

This work explores the manufacturability of pyramidal fin arrays produced using the cold spray process. Near-net shaped pyramidal fin arrays of various sizes and fin densities were manufactured using masks made of commercially available steel wire mesh. The feedstock powders used to produce the fins are characterized using scanning electron microscopy. Obstruction of the masks was investigated. The standoff distances between the substrate, mesh, and nozzle were empirically determined. Fin array characterization was performed using digital microscopy. The fin arrays' heat transfer performance was assessed experimentally for a range of Reynolds number relevant to the application sought. The fins produced using the cold spray process outperform traditional straight (rectangular) fins at the same fin density and it is hypothesized that this is due to increased fluid mixing and turbulence.

**Keywords** cold spray, compact heat exchanger, heat transfer, net shape forming, pyramidal fin array

## 1. Introduction

Distributed power generation (DPG) offers a novel approach to reduce power losses, as electricity is carried over shorter distances to the end users, thus decreasing power line losses. It also presents the advantage of protecting the grid against widespread electrical failure, as generators individually serve a smaller population (Ref 1, 2). Micro-turbines (MT) are instrumental in the development of a high efficiency DPG system. MTs are small scale, single stage, and single shaft gas turbine generators. They provide the combined advantages of low emissions, high reliability, and good efficiency in a compact package (Ref 3). The major drawback is the associated wasted heat. Increasing the viability of DPG is directly linked with the development of higher efficiency MTs that generate between 25 and 500 kW. The simplest MT designs often have efficiencies below 20%, a lower value than diesel, and gasoline engines which typically operate between 30 and 35% thermal efficiencies over the same power production range (Ref 4). However, MTs equipped with a recuperator preheating the inlet gas using

the heat recovered from the hot exhaust gases have the potential to theoretically attain 50% thermal efficiency (Ref 4). This waste heat could also be used for plant processes requiring heating or hot water production. In these cases, efficiencies in the order of 80% are attainable by such systems, referred to as combined heat-power (CHP) systems (Ref 3).

Recuperators play a vital role to a successful MT design, and often represent 20–25% of the cost and 20–35% of the volume of a MT package (Ref 4). Typically, unit cost and volume are closely tied together, as material cost is one of the main contributors to the total cost. Increasing recuperator efficiency by 5–10% can improve the overall thermal efficiency of the system by the same amount (Ref 4). Currently, recuperators operate at efficiencies ranging from 75 to 85%. These recuperators usually use shell and tube or packed bed geometries, however they experience problems such as leakage, high pressure losses, axial conduction losses, and high manufacturing cost (Ref 5, 6).

A design of compact heat exchangers, referred to as a wire mesh heat exchanger (WMHE), was developed as a solution for common challenges faced by high efficiency compact heat exchanger. WMHE are based on the concept of using a metallic wire mesh as a conductive material inside a cellular metal sandwich structure, first investigated in the late 1980s (Ref 7, 8). The first configuration developed using this concept was the folded wire mesh geometry, shown in Fig. 1. This geometry is produced by successively folding wire mesh textiles and brazing thin metallic sheets to these folds to create a flow barrier. The disadvantage of this technique is that the contact area per fold between the mesh the brazed plates is small, combined with the maximum number of times the wire mesh can be folded per unit length, thus limiting the heat that can be transferred (Ref 9). This limitation, in addition to the large cost associated with the manufacturing technique, calls for an improved production method which can reduce both cost and volume while maintaining high heat transfer properties.

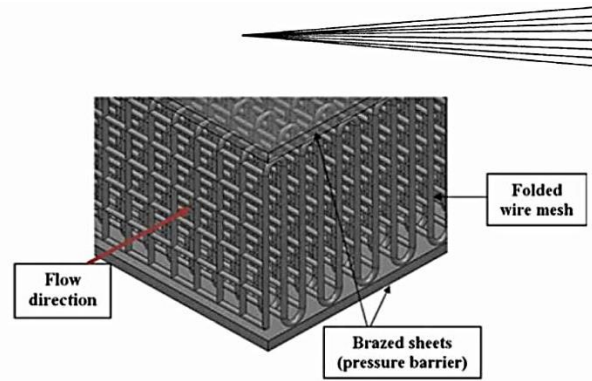
This article is from a presentation at the 2013 International Thermal Spray Conference, held May 13–15, 2013, in Busan, South Korea, and has been expanded from the original presentation.

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| Nomenclature    |                                                                           |
|-----------------|---------------------------------------------------------------------------|
| $\Delta T_1$    | Inlet temperature difference (K)                                          |
| $\Delta T_2$    | Outlet temperature difference (K)                                         |
| $\Delta T_{lm}$ | Log-mean temperature difference (K)                                       |
| $\eta_f$        | Individual fin efficiency                                                 |
| $\eta_o$        | Overall fin efficiency                                                    |
| $\theta$        | Spray angle ( $^\circ$ )                                                  |
| $\mu$           | Dynamic viscosity (Pa s)                                                  |
| $\rho$          | Fluid density ( $\text{kg/m}^3$ )                                         |
| $A_f$           | Fin heat transfer area ( $\text{m}^2$ )                                   |
| $A_{flow}$      | Net flow area ( $\text{m}^2$ )                                            |
| $A_{tot}$       | Total heat transfer area ( $\text{m}^2$ )                                 |
| $A_u$           | Unfinned heat transfer area ( $\text{m}^2$ )                              |
| $B$             | Base fin length (m)                                                       |
| $C_p$           | Fluid specific heat capacity ( $\text{kJ}/(\text{kg K})$ )                |
| $D$             | Base diameter (m)                                                         |
| $d_h$           | Hydraulic diameter (m)                                                    |
| $FD$            | Fin density (fin/m)                                                       |
| $H$             | Channel height (m)                                                        |
| $h$             | Convective heat transfer coefficient ( $\text{W}/(\text{m}^2 \text{K})$ ) |
| $I_1$           | Bessel function of order one                                              |
| $I_2$           | Bessel function of order two                                              |
| $j$             | Colburn factor                                                            |
| $k_f$           | Fluid thermal conductivity ( $\text{W}/(\text{m K})$ )                    |
| $k_m$           | Fin material thermal conductivity ( $\text{W}/(\text{m K})$ )             |
| $L$             | Sample length (m)                                                         |
| $\dot{m}$       | Mass flow rate ( $\text{kg/s}$ )                                          |
| $m$             | Fin heat transfer parameter ( $\text{m}^{-1}$ )                           |
| $Nu_D$          | Nusselt number based on hydraulic diameter                                |
| $N_f$           | Number of fins                                                            |
| $P_{flow}$      | Flow perimeter (m)                                                        |
| $Pr$            | Prandtl Number                                                            |
| $q$             | Heat flux ( $\text{W}/\text{m}^2$ )                                       |
| $Re_D$          | Reynolds number based on hydraulic diameter                               |
| $R_{eq}$        | Equivalent thermal resistance ( $\text{K/W}$ )                            |
| $S$             | Space between fin edges (m)                                               |
| $T_{in}$        | Inlet fluid temperature (K)                                               |
| $T_{out}$       | Outlet fluid temperature (K)                                              |
| $UA$            | Thermal conductance ( $\text{W/K}$ )                                      |
| $V$             | Fluid velocity (m/s)                                                      |
| $W$             | Channel width (m)                                                         |

To this end, a new method for producing WMHE has recently been developed. The innovation of this approach resides in the fact that the wire mesh textile is stacked and cut, then sealed with a thin metallic layer, as shown in Fig. 2(a) and (b) (flow barriers on the longitudinal side are not represented for clarity). These features allow more contact area per unit length between the heat transfer medium and the pressure barrier (brazed plates). Nevertheless, this design/method for producing WMHE does offer some areas for improvement. Currently, the cost associated with brazing a thin metal sheet to the wire mesh is high and the operation is labor intensive. To replace this process, research has been performed where a thick and dense metallic coating is produced using thermal or kinetic spray processes. Once this coating has been ap-



**Fig. 1** Wire mesh heat exchanger using folded wire mesh geometry as heat transfer medium

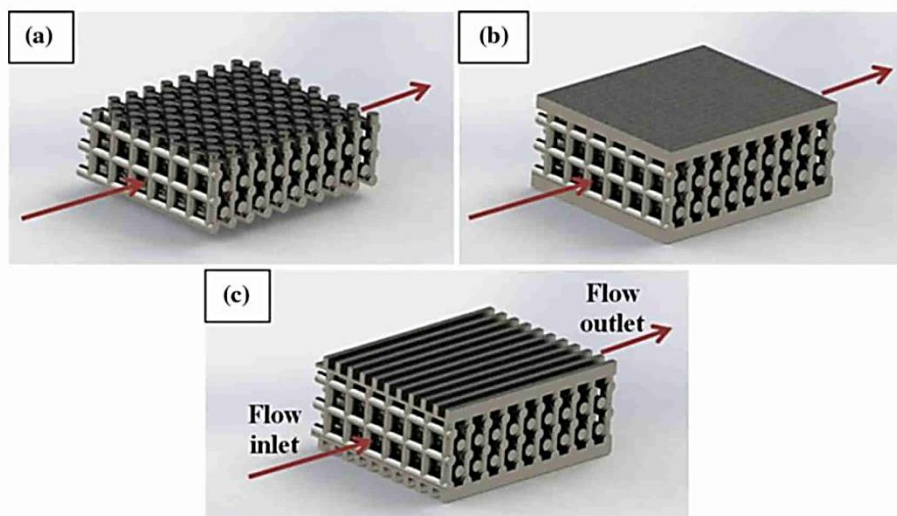
plied, fins are machined on the outer surface to produce simple rectangular (straight cut) fin geometries (Fig. 2c). While this operation saves costs, it is time consuming and is currently restricted by machining tolerances.

Other interesting and promising types of heat exchangers were developed using a similar principle but with different types of internal heat transfer medium, such as metal foam heat exchangers encased in a metallic skin deposited using wire-arc or plasma spray, as investigated by Salimi Jazi et al. (Ref 10, 11). In this case, thermal spray processes are used to seal commercially available nickel foam bricks with Inconel 625 to create fully enclosed heat exchangers. No heat transfer enhancing features are used on the outer surface of the heat exchangers.

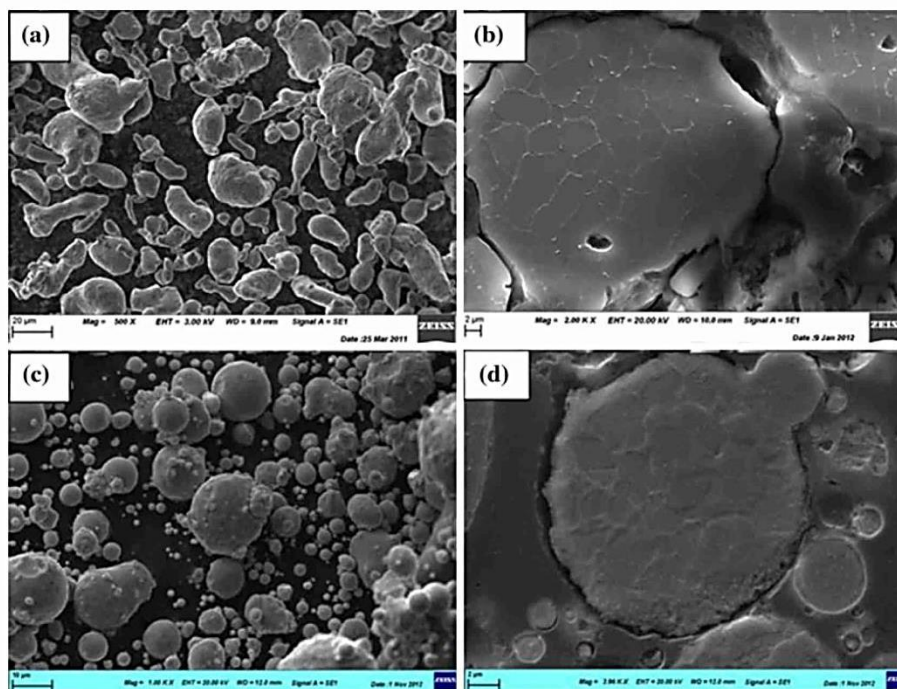
Thermal spray deposition processes operate on the principle of heating and accelerating a carrier gas to project particles onto a substrate in order to form a dense coating. Combustion and plasma spray processes such as high velocity oxy fuel (HVOF) and plasma spray use a high temperature gas stream to melt the projected particles, where adhesion is sustained by the particles solidifying on the substrate surface and onto each other. Kinetic spray processes such as cold spray, rely on momentum transferred from an inert gas, usually helium or nitrogen, to relatively cold particles (usually less than half the melting temperature) which are projected onto a substrate's surface. The gas is accelerated to supersonic speeds using a convergent-divergent (DeLaval) nozzle, where its kinetic energy is imparted to the particles from fluid drag effects (Ref 12, 13). Upon colliding with the substrate, this energy is used to plastically deform these particles and to induce adhesion.

In this paper, the viability of using cold spray to manufacture near-net shape fin arrays was investigated as a potential solution to the problems encountered by fin machining. Pyramidal fin arrays were produced using cold spray which required no post-processing. The aim of this study is to demonstrate the manufacturability of such fins and also to start comparing the heat transfer properties of pyramidal and rectangular fin arrays by studying specifically the 24 fins/in. case.

Two fin materials were selected for their widely different thermal and mechanical properties: aluminum and stainless steel. These materials have a high resistance to



**Fig. 2** Successive steps to manufacture wire mesh heat exchanger: (a) stacked wire mesh, (b) brazed thin metallic sheets, (c) machined external fins



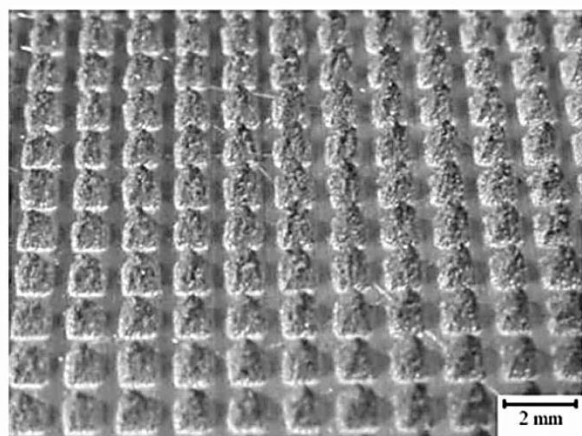
**Fig. 3** SEM images of (a) As-received pure aluminum feedstock powder; (b) etched cross-section of pure aluminum particle; (c) As-received 304L stainless steel feedstock powder; (d) etched cross-section of 304L stainless steel particle

corrosion and are known to be easily deposited with cold spray. Stainless steel's operating temperature is higher than aluminum, but its thermal conductivity is lower. To this end, applications that have high process temperatures would require the former material, while lower process temperature applications could benefit from the increase in performance the latter material could offer.

## 2. Experimental Coating System Description

The cold spray system used for this work is a commercially available Plasma Giken PCS-1000 (Ref 14). The spray nozzle has a 3.0 mm throat diameter and a 6.5 mm exit diameter.





**Fig. 4** Fin array overview of aluminum sample Al-2 produced using cold spray and masking technique

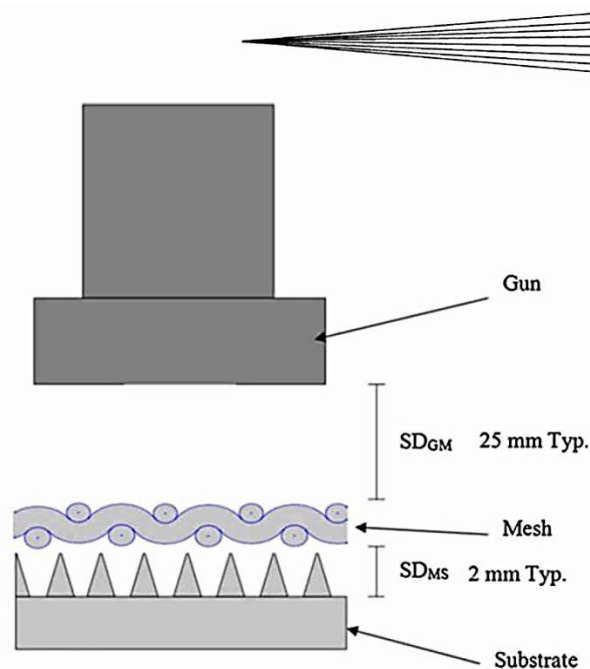
The first feedstock powder used was aluminum (99.8% Al), provided by Centerline (Windsor) Ltd. (Windsor, Ontario, Canada), and distributed commercially under the name SST-A5001. The powder size range is characterized by a  $D_{10}$  value of 13.29  $\mu\text{m}$ , a  $D_{50}$  value of 25.35  $\mu\text{m}$ , and a  $D_{90}$  value of 49.58  $\mu\text{m}$ , as measured by a Microtrac S3500 laser diffraction analysis apparatus provided by Microtrac (Montgomeryville, Pennsylvania, USA). Figure 3(a) presents an image of the powder taken by a scanning electron microscopy (SEM) EVO-MA 10 provided by Zeiss (Oberkochen, Germany). It presents an elongated, non-spherical geometry. The grain structure is shown in Fig. 3(b), as etched with Keller's reagent. The grains have an equiaxial shape in the center of the particle, while the exterior is composed mostly of elongated grains.

The second material used was commercially available stainless steel grade 304L, provided by Sandvik Osprey Ltd (Sandviken, Sweden). The powder size range is characterized by a  $D_{10}$  value of 4.55  $\mu\text{m}$ , a  $D_{50}$  value of 14.14  $\mu\text{m}$ , and a  $D_{90}$  value of 31.30  $\mu\text{m}$ , as assessed by cumulative volume based laser diffraction analysis. Figure 3(c) presents an SEM image of this powder. It presents a spherical geometry. The grain structure is shown in Fig. 3(d), as etched with Methanolic Aqua Regia reagent. The particles observed have an equiaxial grain geometry.

### 3. Experimental Procedure

#### 3.1 Masking Technique

To create pyramidal fin arrays (Fig. 4), commercially available plain woven, steel wire mesh (McMaster-Carr, Aurora, OH, USA) was used to mask the substrate and generate the required fin geometry. This process is illustrated in Fig. 5, where  $SD_{MS}$  and  $SD_{GM}$  designate the standoff distances from the mesh to the substrate and from the mesh to the nozzle, respectively. This creates a pyramidal fin array that closely mimics the reversed geometry of the chosen wire mesh, with peaks aligning with the



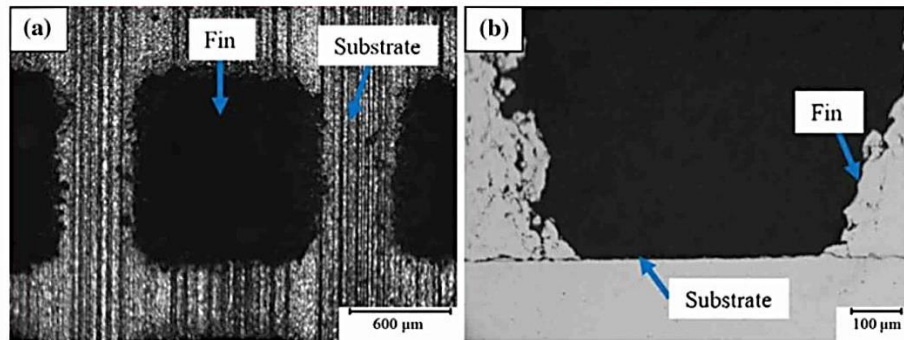
**Fig. 5** Schematic of the masking technique used to create pyramidal fin arrays

holes of the screened area and little to no deposition underneath the wires (Fig. 6). Other thermal spray processes were not considered for this production method as they would clog the mask.

Fins were cold sprayed on aluminum substrates (Al6061 T6) with standard dimensions of 5.1 cm by 5.1 cm. The cold spray trials detailed in this work were done on extruded aluminum substrates instead of thermal sprayed coatings on the WMHE to demonstrate viability. Further work will be performed to determine the parameters required to produce the same features on specific thermal spray sealed heat exchangers. The substrates were cleaned and degreased using acetone after being cut to size. No further coupon preparation, such as grit-blasting or heat treatments, was performed. The spray parameters used to obtain these samples using nitrogen as the carrier gas are given in Table 1 for both feedstock powders. The number of passes was varied according to the target height of the fin arrays. No post-treatment of the sprayed samples was performed.

Standard metallographic procedures were used to obtain cross-sections of the sprayed samples. Control specimens were sprayed without masking to obtain a standard for comparison and to establish a porosity level baseline. All the image analysis was performed using Clemex Vision Lite (Clemex Technologies Inc., Longueuil, QC, Canada) image analysis software. Porosity levels are reported with a mean value of 10 measurements for each data set. Spray angles are determined from the measured geometries of the sprayed samples.

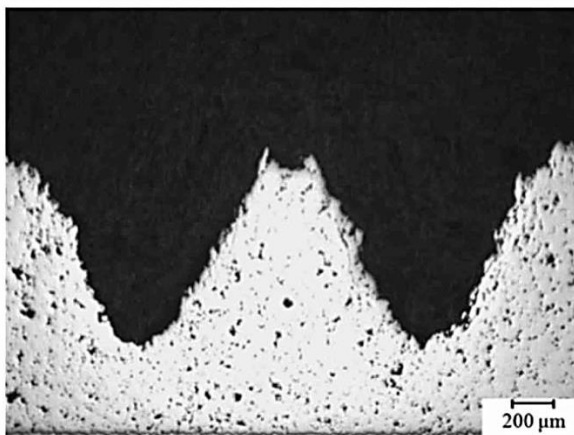
Several trials were performed to determine the effect of the standoff distance between the wire mesh and the gun, as well as the distance between the mask and the substrate. It was determined that starting with a very small distance between the mesh and the substrate (less than



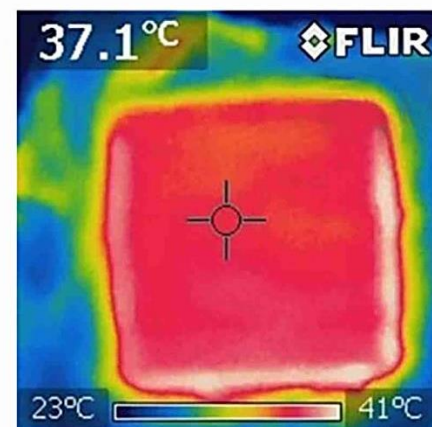
**Fig. 6** (a) Top view of typical fin deposition (aluminum sample); (b) cross-section of typical fin deposition illustrating the clean area between the fins (aluminum sample)

**Table 1** Spray parameters

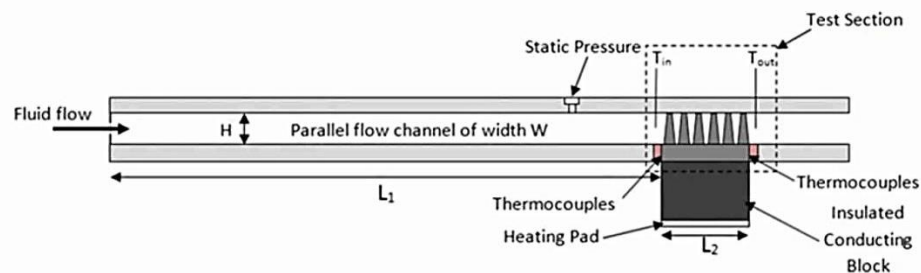
| Powder type  | Nozzle inlet pressure, MPa | Nozzle inlet gas temperature, °C | SD <sub>GM</sub> , mm | SD <sub>MS</sub> , mm | Nozzle traverse speed, mm/s |
|--------------|----------------------------|----------------------------------|-----------------------|-----------------------|-----------------------------|
| Al SST-A5001 | 3                          | 300                              | 25.4                  | 2.0                   | 350                         |
| SS 304L      | 3                          | 900                              | 25.4                  | 2.0                   | 300                         |



**Fig. 7** Deposition between fins when mesh-substrate distance is larger than 3 mm for typical aluminum sample

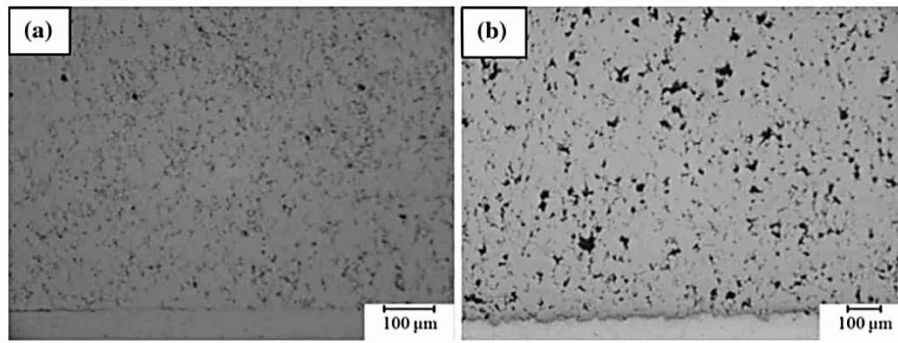


**Fig. 9** Uniform heat flux on the fins (without flow) resulting in a uniform surface temperature

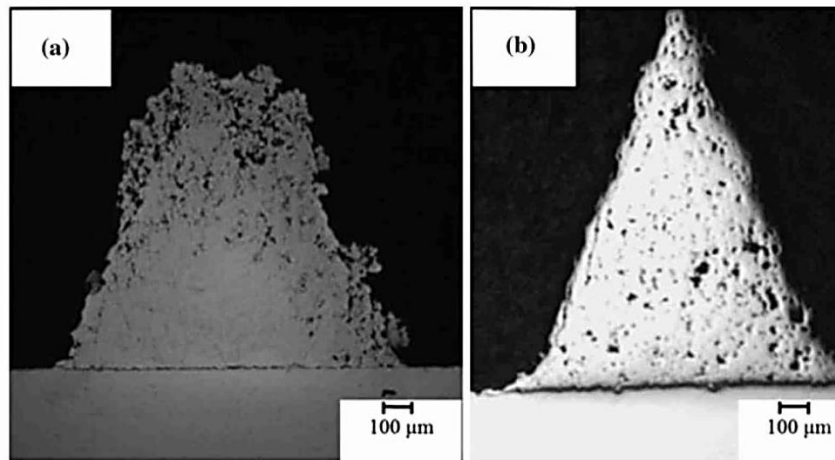


**Fig. 8** Schematic of the heat transfer test fixture





**Fig. 10** Cross-section of coatings control samples used to characterize the porosity level (a) Al SST-A5001; (b) SS 304L



**Fig. 11** Cross-section of a fin of samples showing different porosity distributions (a) Al-1; (b) SS-2

**Table 2** Pyramidal fin geometry of tested samples

| Sample number | Fin density, Fins/in. | Wire diameter, in. | Fin height, mm | Hydraulic diameter, mm | Total heat transfer area, mm <sup>2</sup> | Spray angle, ° |
|---------------|-----------------------|--------------------|----------------|------------------------|-------------------------------------------|----------------|
| Al-1          | 12                    | 0.035              | 1.8            | 1.57                   | 4335                                      | 19.3           |
| Al-2          | 16                    | 0.028              | 2.2            | 1.49                   | 5820                                      | 11.3           |
| Al-3          | 16                    | 0.018              | 1.4            | 1.13                   | 4767                                      | 22.0           |
| Al-4          | 24                    | 0.014              | 1.4            | 0.92                   | 6117                                      | 14.1           |
| SS-1          | 12                    | 0.035              | 2.0            | 1.67                   | 4671                                      | 17.1           |
| SS-2          | 16                    | 0.028              | 1.5            | 1.27                   | 4599                                      | 16.3           |
| SS-3          | 30                    | 0.016              | 1.0            | 0.76                   | 5129                                      | 12.4           |

**Table 3** Samples tested for comparison

| Sample number | Fin density, Fins/in. | Fin width, in. | Fin spacing, in. | Height, mm | Hydraulic diameter, mm | Total heat transfer area, mm <sup>2</sup> |
|---------------|-----------------------|----------------|------------------|------------|------------------------|-------------------------------------------|
| US            | N/A                   | N/A            | N/A              | 2.0        | 3.85                   | 2581                                      |
| SC-B          | 24                    | 0.015          | 0.025            | 1.7        | 0.92                   | 9807                                      |
| SC-C          | 24                    | 0.015          | 0.025            | 1.4        | 0.87                   | 8344                                      |

0.75 mm) created problems where the mask would become embedded in the coatings produced. Increasing the distance to more than 3.0 mm has a detrimental effect on spray angles and spray edge cleanliness. At this

standoff distance, some deposition underneath the wires is noted and edges do not appear as sharp, as shown in Fig. 7. The optimal distance found is from 1.0 to 2.5 mm between the wire mesh mask and the substrate. A distance



of 2.5 mm was used for the samples tested in this work. The optimal standoff distance between the mesh and the cold spray nozzle was found to be 25 mm.

### 3.2 Heat Transfer Test Apparatus

Performance of the various fin arrays produced was assessed using an in-house heat transfer and pressure drop test apparatus, adapted from the literature (Ref 5). A schematic diagram of the apparatus is shown in Fig. 8.

The main components of this apparatus are the test section, heating pad, air supply and data acquisition system. The air is supplied by a compressed air line, properly filtered and dehumidified to obtain room temperature, dry air, free from oil contamination. The air inlet flow rate is controlled by a flow meter and a needle valve. This air is forced into a rectangular channel with the test section located near the exit. This rectangular channel's top and bottom plates (76.2 mm wide by 457.2 mm long by 19.05 mm thick) are made of acrylic, and they are spaced apart using a rectangular rubber gasket, which is compressed to obtain the required hydraulic diameter. These gaskets also ensure that the joint is leak-proof. A length of 320 mm of unobstructed flow ensures that the fluid is fully developed hydrodynamically when it reaches the test section for the given hydraulic diameter and Reynolds number (Ref 15).

The test sample is subjected to a constant heat flux provided by a strip heater. A conducting block is used to ensure that heat is evenly distributed onto the substrate's surface, resulting in a constant surface temperature under free convection conditions, as shown in Fig. 9.

A stagnation pressure tap is located 25.4 mm upstream from the test section. Two thermocouples are located 6 mm before and after the test section at mid-height to record the fluid local average temperature. Four thermocouples are attached to the fin array's base, with two on the side facing the incoming flow, and two on the side facing the exiting flow. On each face, one thermocouple is located on the centerline, while the other thermocouple is located 6 mm from the wall. All the thermocouples used are T-type, butt-bonded, thin gage (28 gage) thermocouples.

Measurements were performed at steady-state conditions and repeated until statistical significance was achieved. Data samples were taken over a 30 s interval, at 2 Hz. Five sets of data samples were recorded for every flow condition. Measurements were taken at flow rates corresponding to Reynolds numbers varying from 200 to 2000, as this is the typical application range sought for these exchangers. Deviation of the measurements from the mean value was less than 5%.

## 4. Results

### 4.1 Fin Array Characterization

Low porosity levels are observed for the bulk control coatings produced (without masking), with measurements indicating less than 5% porosity levels, as shown in Fig. 10. Fins produced with the aluminum powder have a different porosity distribution than the fins produced with

stainless steel powder. Figure 11(a) shows that the porosity in the aluminum fins varies from a fully dense core to a porous exterior layer. Stainless steel shows a relatively uniform porosity level throughout the coating, as seen in Fig. 11(b). Some of the voids are relatively large (20  $\mu\text{m}$ ) which is comparable to the voids observed in the outer layer of the aluminum fins. This difference in porosity distribution is attributed to the fact that aluminum and stainless steel have different bonding characteristics. The primary factor is that the aluminum core is compacted by the impacts of the subsequent particles on the newly formed coating, while the porous exterior layer does not benefit from this phenomenon. This phenomenon is much more pronounced for the aluminum feedstock because its surface oxide layer is easily broken, compared to stainless steel. Furthermore, the aluminum particles are larger and their non-spherical morphology tends to increase the fluid drag, which both contribute to increase the particle momentum and oxide breaking power. The second factor is that aluminum is sprayed at parameters on the lower edge of the deposition window to reduce nozzle clogging problems, while stainless steel is sprayed with a high enough gas temperature and pressure to overcome its higher critical velocity and allow a better initial adhesion without compaction (Ref 16, 17).

Several geometries of pyramidal fins were successfully deposited on aluminum substrates, from 12 to 30 fins/in. Table 2 presents the different pyramidal fin geometries of the samples produced while Table 3 shows the other samples tested for reference or comparison purposes. Sample labeled US is an unfinned surface while samples labeled SC are vertical rectangular (straight cut) fin samples, with the suffix B and C designating that the fins were machined from bulk Al 6061 T6 or from a pure aluminum cold sprayed coating, respectively.

Scalability does not seem to be an issue, as much larger samples (101.6 mm by 152.4 mm) were also successfully produced with no differences noted when compared to the smaller samples, as shown in Fig. 12.

Deposition of material is noted on the mesh, as shown in Fig. 13. The build-up has a pyramidal prism shape along the wire length, and once a critical height has been attained, deposition on the mesh slowly stops. This critical height is a function of the sprayed material, spray parameters and fin density. At this critical height, the impact angle is so small that the normal component of the particle velocity is not sufficient to create adhesion. The wire mesh sizes tested up to 24 wires/in. do not clog, even after several minutes of deposition. At 30 wires/in. with a wire diameter of 0.016" (0.41 mm), clogging of some holes is noted with the aluminum feedstock. The stainless steel feedstock powder does not show this behavior. This is explained by looking at the particle size distribution for these two materials, noting that the average aluminum particle is 80% larger than the average stainless steel particle and that the aluminum adheres very easily when sprayed, making it much more likely to clog. The stainless steel, on the other hand, is much harder to deposit and has reduced tendency to block the mesh holes. Mesh densities higher than 30 wires/in. were not tested.



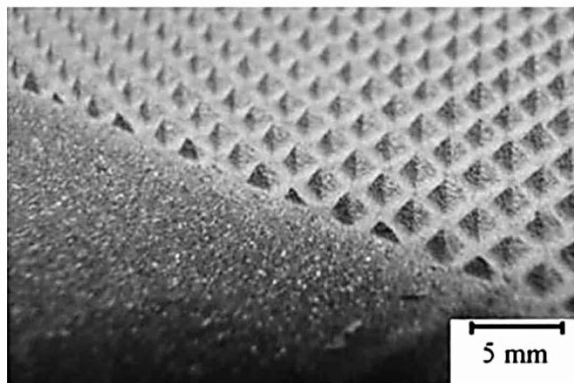


Fig. 12 Typical stainless steel pyramidal fin arrays

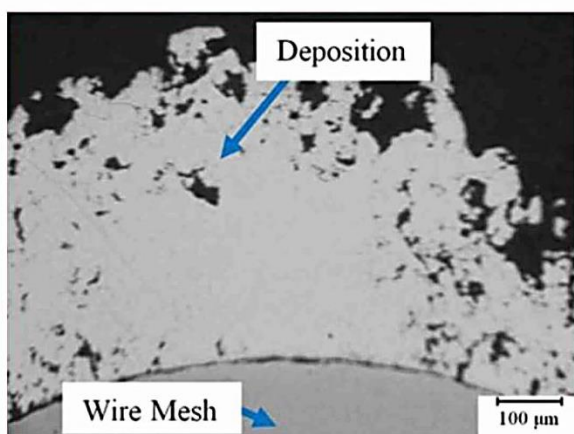


Fig. 13 Deposition on the masking mesh of aluminum sample Al-1

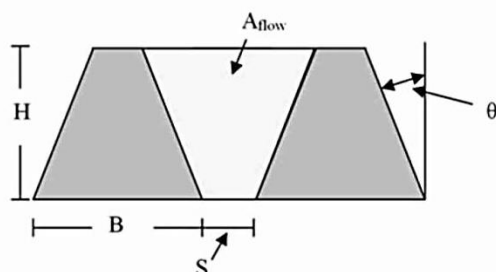


Fig. 14 Schematic of the fin array unit with geometric variables identification

## 4.2 Heat Transfer Performance

**4.2.1 Data Analysis Method.** To analyze the raw data obtained and compare it in suitable manner, non-dimensional parameters must be used. For this purpose, flow conditions are expressed using the Reynolds number. This study provides non-dimensional numbers with a characteristic length based on the hydraulic diameter for the

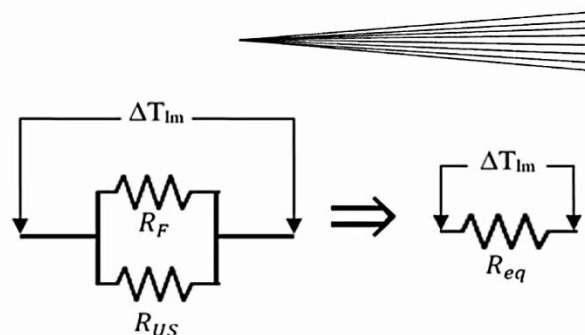


Fig. 15 Fin array thermal circuit diagram

comparison of different fin arrays. Performance is assessed using the thermal conductance of the fin array, which is equal to the amount of heat transferred per unit temperature difference between the fin array and the fluid, measured in Watts per degree Kelvin. Figure 14 illustrates the geometric variable used to perform the analysis.

The Reynolds number based on the hydraulic diameter is given by:

$$Re_D = \frac{\rho V d_h}{\mu} = \frac{\dot{m} \cdot d_h}{W \cdot FD \cdot A_{flow} \cdot \mu} \quad (\text{Eq 1})$$

where  $\dot{m}$  is the mass flow rate,  $d_h$  is the hydraulic diameter of the fin array,  $W$  is the sample width,  $FD$  is the fin density,  $\mu$  is the fluid's dynamic viscosity, and  $A_{flow}$  is the total face area the fluid can flow into. The variables  $d_h$  and  $A_{flow}$  are given by:

$$d_h = \frac{4 \cdot A_{flow}}{P_{flow}} = \frac{2 \cdot A_{flow}}{\frac{H}{\cos(\theta)} + S + H \cdot \tan(\theta)} \quad (\text{Eq 2})$$

$$A_{flow} = (S + H \cdot \tan(\theta)) \cdot H \quad (\text{Eq 3})$$

where  $P_{flow}$  is the flow channel's perimeter,  $\theta$  is the fin angle,  $S$  is fin spacing from edge to edge, and  $H$  is the fin height.

Figure 15 shows a thermal circuit of the fin array, with the reduction of the thermal resistances of the fins and the unfinned surface designated respectively by  $R_F$  and  $R_{us}$  to an equivalent resistance  $R_{eq}$ .

The thermal conductance parameter  $UA$  can be calculated as:

$$UA = \frac{1}{R_{eq}} = h \cdot A_{tot} \cdot \eta_o \quad (\text{Eq 4})$$

where  $h$  is the convective heat transfer coefficient,  $\eta_o$  is the overall surface efficiency, and  $A_{tot}$  is the total heat transfer area, given by:

$$A_{tot} = A_f + A_u = \left( 2N_f B \sqrt{H^2 + \left( \frac{B}{2} \right)^2} \right) + (WL - N_f B^2) \quad (\text{Eq 5})$$

In the previous equation,  $N_f$  is the number of fins in the array,  $B$  is the fin's length at its base,  $L$  is the sample's length,  $A_f$  denotes the total fin heat transfer area, while  $A_u$  represents the unfinned base's exposed area. The overall efficiency is calculated using the following equation:

$$\eta_o = 1 - \frac{A_f}{A_{tot}}(1 - \eta_f) \quad (\text{Eq 6})$$

where  $\eta_f$  is the individual fin efficiency, defined as the heat transferred by the fin over the maximum possible heat that can be transferred, which is numerically equal to:

$$\eta_f = \frac{2 I_2(2mH)}{mH I_1(2mH)} \quad (\text{Eq 7})$$

with:

$$m = \sqrt{\frac{4h}{k_m D}} \quad (\text{Eq 8})$$

In the previous equations adapted from Incropera's work (Ref 15),  $I_1$  and  $I_2$  denote the first and second order Bessel functions, respectively, while  $k_m$  is the thermal conductivity of the fin's metal.

The individual fin efficiency equation for triangular pin fins are used for the sprayed pyramidal pin fins in this work since it is the shape with the most similar geometry. The only difference between the two is the shape of the base, since the equation is for a circular base while the manufactured fins have a square base. Therefore,  $D$  in the previous equation is replaced by the fin length at the base ( $B$ ).

The fin array's convective heat transfer coefficient and Nusselt number are calculated with the following relations:

$$h = \frac{q}{\Delta T_{lm} A_{tot} \eta_o} \quad (\text{Eq 9})$$

$$Nu_D = \frac{h \cdot d_h}{k_f} \quad (\text{Eq 10})$$

In these equations,  $q$  is the heating rate,  $\Delta T_{lm}$  is the log-mean temperature difference between the fluid and the fin array, and  $k_f$  is the fluid's thermal conductivity. The log-mean temperature difference is calculated with:

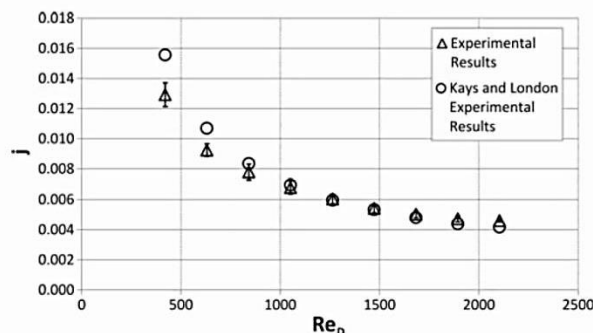
$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (\text{Eq 11})$$

where  $\Delta T_1$  and  $\Delta T_2$  are the temperature difference between the fin surface and the flow at the inlet and outlet, respectively. The total heat input rate to the system is calculated using:

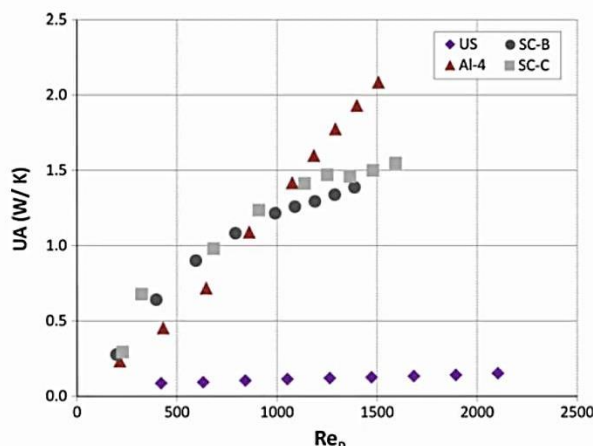
$$q = \dot{m} \cdot (T_{out} - T_{in}) \cdot C_p \quad (\text{Eq 12})$$

where  $(T_{out} - T_{in})$  is the fluid temperature difference between the entry and the exit of the test section,  $C_p$  is the fluid's thermal capacitance.

An uncertainty analysis was performed using the method described in Coleman and Steele (Ref 18). The uncertainties calculated with the root-square-method for the thermal conductance, the mean heat transfer coefficient, and Reynolds number were estimated to be less than 5% for all three measurement types.



**Fig. 16** Measurement accuracy comparison for an unfinned surface



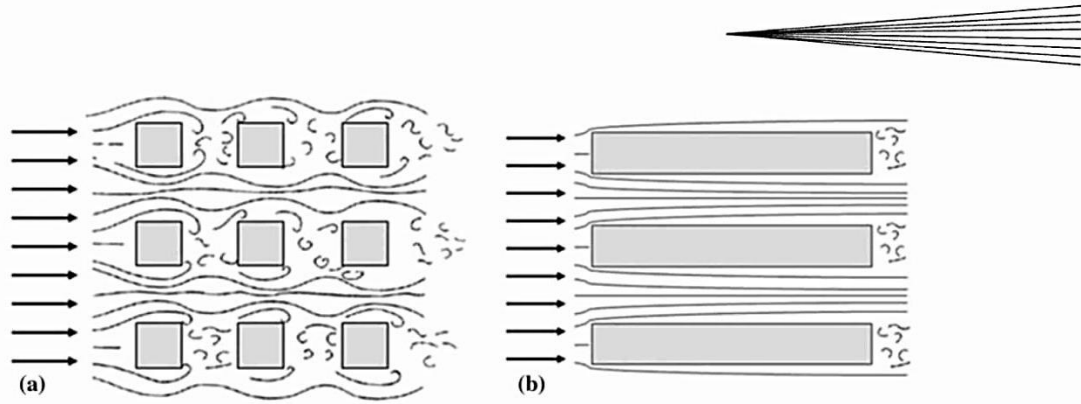
**Fig. 17** Thermal conductance comparison between Al-4 (triangle), straight cut bulk (circle), and straight cut coating (square) while the unfinned surface is used as reference (diamond)

Tests were performed to determine the apparatus' accuracy. Experiments were conducted with an empty channel, heated on one side and insulated on the other, and then compared to existing curves for the same conditions produced by Kays and London (Ref 19). These curves use the Colburn factor as the performance indicator and their authors do not specify all the testing conditions required to transform this data back into raw data. The comparison must then be made using this parameter, which stems from an analogy between the transfer of heat and momentum, and which can be computed with:

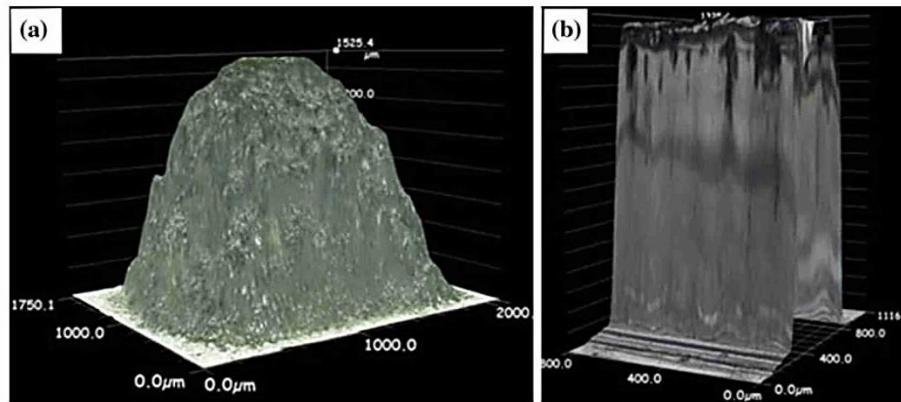
$$j = \frac{Nu_D}{Re_D \cdot Pr^{1/3}} \quad (\text{Eq 13})$$

In the previous relation,  $Pr$  is the Prandtl number of the fluid. For these tests, the channel height was set to 2.0 mm, which yielded a hydraulic diameter of 3.85 mm. Tests were performed at flow rates of 0.0002 to 0.001 kg/s in 0.0001 kg/s increments, which correspond to Reynolds numbers of 421-2104. Figure 16 shows the results obtained using the test fixture, which show a high similarity with published results found in the literature, with the maximal deviation mostly at very low flow rates. The error can be

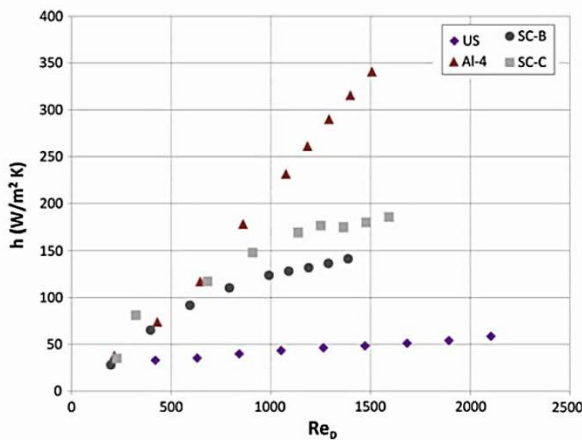




**Fig. 18** Illustration of flow pattern for (a) pyramidal fin arrays; (b) straight cut fin arrays. Intensive mixing/turbulent zones are more frequent in pyramidal fin arrays



**Fig. 19** Three dimensional images (from digital microscopy) of surface at  $\times 300$  magnification for (a) pyramidal fin arrays; (b) straight cut fin arrays



**Fig. 20** Convective coefficient comparison between Al-4 (triangle), straight cut bulk (circle), and straight cut coating (square) while the unfinned surface is used as reference (diamond)

explained noting that the instrument uncertainty is highest at low flow rates due to the full scale measurement uncertainty, with the theoretical error at 15%. These tests act as a basis for the validation of the results obtained for

the pyramidal fin arrays, where no published literature has been found to compare with.

**4.2.2 Performance Comparison with Traditional Fin Arrays.** The thermal conductance as a function of the Reynolds number for various fin array samples is shown in Fig. 17. The circle and square symbols represent data obtained from rectangular fin samples made from bulk aluminum and an aluminum coating, respectively. The triangle symbols represent a pyramidal fin sample made with the technique detailed previously, at the same fin density and hydraulic diameter than the straight cut samples. Finally, data obtained from an unfinned surface is represented by a diamond, and is shown for comparison purpose.

Very little performance difference can be seen between the rectangular fins cut from bulk Al 6061 and those cut from cold sprayed pure aluminum (Fig. 17). This is to be expected as the nominal compositions of these materials are very similar, the porosity level in the cold spray coating is less than 1% while the fin density, the width and the thickness of the samples are also identical. It is noted that the error bars on the graph were so small that they were barely visible and thus were not included. The increase in thermal conductance over the range of Reynolds number studied is due to the increase in the



convective heat transfer coefficient. This is to be expected as flow detachment along the length of the fin creates turbulences which increase heat transfer. Increasing the Reynolds number also increases the detached flow length, creating more area with a higher heat transfer coefficient, leading to better thermal performance.

The performance of straight cut fins was compared with sprayed pyramidal fin array (Fig. 17). The pyramidal fin array with a fin density of 24 fin/in has a performance comparable to the straight cut fins with the same density when comparing thermal conductance up to  $1200 Re_D$ . Beyond  $Re_D$  of 1200, pyramidal type fin thermal conductance values surpass those of straight cut fins with the same fin density by up to 30% at  $Re_D$  of 1500. This effect is believed to be due to a large increase in the convective heat transfer coefficient from fluid mixing attributable to two mechanisms. The first factor is the turbulent wake behind the fins. This turbulence is due to the flow detachment from the fin surface, which does not occur as prominently for continuous fins, as illustrated in Fig. 18. The second contributing effect is the fin roughness. Machined fins have a very low surface roughness due to the high quality surface finish inherent to the machining technique, while sprayed fins have a high degree of roughness (Fig. 19a, b). These three dimensional pictures were obtained using a depth of field microscope VHX-2000 provided by Keyence (Mississauga, ON, Canada), and show that the machined fins have a lower roughness than the instrument's resolution at  $300\times$  magnification ( $2\text{ }\mu\text{m}$ ). The roughness levels found in pyramidal fins are believed to induce mixing of the fluid and further promote convective heat transfer. This is confirmed by Fig. 20, where the convective heat transfer coefficient of pyramidal fins is almost 100% greater than the convective coefficient of traditional fin arrays at  $Re_D$  of 1500. This increase in convective heat transfer coefficient is slightly offset by the increased heat transfer area offered by continuous straight cut fins. A 36% difference in total heat transfer area is measured between straight cut fins and pyramidal fins at the same fin density and height ( $8344\text{ mm}^2$  versus  $6117\text{ mm}^2$ , respectively), as shown in Tables 2 and 3. Overall, the contribution of the increased convective heat transfer coefficient is sufficient to still provide an increased thermal conductance for pyramidal type fins.

## 5. Summary and Conclusion

Near-net shape pyramidal fin arrays of various sizes were deposited using commercially available wire mesh as a masking medium for the cold spray process. Fin characterization was performed using optical microscopy, while SEM micrographs were used to characterize the aluminum and stainless steel feedstock powders. The effect of standoff distances between the spray gun, mask and substrate was investigated, with an optimal range identified. Clogging of the wire mesh mask was found to occur at 30 wires/in. with a wire diameter of  $0.016''$

( $0.41\text{ mm}$ ) for the aluminum feedstock, while no problem was encountered with the stainless steel feedstock powder. Fin array performance was determined using a steady-stated forced air convection system subjected to a constant heat flux boundary condition on one side for a 24 fin/in. pyramidal sample and was compared to an equivalent 24 fin/in. rectangular fin sample. Further heat transfer experiments will be performed to determine the thermal performance of the other pyramidal fin array samples presented in this work and will be detailed in a later communication as the purpose of the current paper was to mostly demonstrate the process feasibility and potential. The new type of fins produced outperforms traditional straight cut fins at the same fin density and hydraulic diameter due to fluid mixing increasing the convective heat transfer coefficient. Significant opportunities exist to further improve this technique in every aspect, such as optimization of spray parameters and the study of the effect of other material types or anisotropy of the fin density, for example. A more in-depth characterization of the fluid mechanics properties of the pyramidal fin array will be performed in the near future.

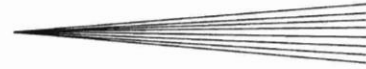
## Acknowledgments

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## References

1. A.B. Lovins, *Small is Profitable: The Hidden Economic Benefits of Making Electrical Resources the Right Size*, Rocky Mountain Institute, 2002
2. S.M. Kaplan and F. Sissine, *Smart Grid: Modernizing Electric Power Transmission and Distribution; Energy Independence, Storage and Security; Energy Independence and Security Act and Resiliency; Integra*, TheCapitol.Net Inc., 2009
3. A. Corbeil, "Study of Small Hydraulic Diameter Media for Improved Heat Exchanger," M.A.Sc. Thesis, University of Ottawa, 2009
4. D. Bohn, *Micro Gas Turbine and Fuel Cell: A Hybrid Energy Conversion System with High Potential*, NATO Research and Technology Organisation, 2005
5. J. Tian, T. Kim, T.J. Lu, H.P. Hodson, D.T. Queheillalt, D.J. Syceck, and H.N.G. Wadley, The Effects of Topology Upon Fluid-Flow and Heat-Transfer Within Cellular Copper Structures, *Int. J. Heat Mass Transf.*, 2004, **47**(14-16), p 3171-3186
6. Z. Anxionnaz, M. Cabassud, C. Gourdon, and P. Tochon, Heat Exchanger/Reactors (HEX Reactors): Concepts, Technologies: State-of-the-Art, *Chem. Eng. Process.*, 2008, **47**(12), p 2029-2050
7. R.L. Shaner, Heat Exchanger Having Metal Wire Screens, and Method of Making Stack of Screens Therefor, US4840228, Year of Priority (Issued): 1985, 1989
8. J. Chisholm, Method of Making a Crimped Wire Mesh Heat Exchanger/Sink, US4843693, Year of Priority (Issued): 1988, 1989
9. J. Assad, A. Corbeil, P. Richer, and B. Jodoin, Novel Stacked Wire Mesh Compact Heat Exchangers Produced Using Cold Spray, *J. Therm. Spray Technol.*, 2011, **20**(6), p 1192-1200
10. H.R. Salimi Jazi, J. Mostaghimi, S. Chandra, L. Pershin, and T. Coyle, Spray-Formed, Metal-Foam Heat Exchangers for High





- Temperature Applications, *J. Thermal Sci. Eng. Appl.*, 2009, **1**(3), p 1-7
11. F. Azarmi, J. Saaedi, T.W. Coyle, and J. Mostaghimi, Microstructure Characterization of Alloy 625 Deposited on Nickel Foam Using Air Plasma Spraying, *Adv. Eng. Mater.*, 2008, **10**(5), p 459-465
12. A.P. Alkhimov, A.N. Papyrin, V.F. Dosarev, N.I. Nesterovich, and M.M. Shuspanov, Gas-Dynamic Spraying Method for Applying a Coating, US5302414, Year of Priority (Issued): 1992, 1994
13. A.O. Tokarev, Structure of Aluminum Powder Coatings Prepared by Cold Gas-Dynamic Spraying, *Met. Sci. Heat Treat.*, 1996, **38**(3-4), p 136-139
14. Plasma Giken, <http://www.plasma.co.jp/en/>, Consulted November 2012
15. F.P. Incropera, D.P. DeWitt, T.L. Bergman, and A.S. Lavine, *Fundamentals of Heat and Mass Transfer*, 6th ed., Wiley, New York, 2006
16. T. Schmidt, F. Gärtner, H. Assadi, and H. Kreye, Development of a Generalised Parameter Window for Cold Spray Deposition, *Acta Mater.*, 2006, **54**, p 729-742
17. H. Assadi, T. Schmidt, H. Richter, J.-O. Kliemann, K. Binder, F. Gärtner, T. Klassen, and H. Kreye, On Parameter Selection in Cold Spraying, *J. Therm. Spray Technol.*, 2011, **20**(6), p 1161-1176
18. H.W. Coleman and W.G. Steele, *Experimentation and Uncertainty Analysis for Engineers*, Wiley, New York, 1999
19. W.M. Kays and A.L. London, *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York, 1984

## 5.2 Paper #2: Base Angle Effect

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The paper presented in the following pages is about the investigation of the first geometric parameter; the base angle. It was proven that spraying at higher height and grinding down to a specified height has little effect on the performance compared to an as-sprayed fin array of similar height. This result is important since following this results all fin arrays will be sprayed higher and ground down in order to reduce the bypass flow that could occur due the non-uniformity of the fin height during the cold spray process. Also, a new method to compare the fin arrays, adapted from the literature [40], was introduced in this paper: the heat transfer surface efficiency by unit volume. This method is based on plotting the thermal and hydrodynamic data as a curve of thermal conductance per unit of fin array volume as a function of the required pumping power per unit volume for different flow conditions. This yields curves which can be directly compared one to another, with the most efficient fin array having a higher thermal conductance at a given pumping power. This paper addresses the objective of the effect of base angle presented in section 3.4.2.



## Performance evaluation of near-net pyramidal shaped fin arrays manufactured by cold spray



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### ABSTRACT

Near-net pyramidal shaped fin arrays have been produced using the Cold Gas Dynamic Spraying (CGDS) process. Some fin arrays have been modified to trapezoid prism geometry by grinding the top of the pyramidal fins to study the effect of varying the base angle, at a constant fin height. All fin arrays have been tested for thermal and hydrodynamic performance. Little variation in thermal conductance between ground and as-sprayed fins is observed for the same fin heights, while a slightly more significant variation in pressure loss through the fin array is found. A comparison of these performances was performed with plain rectangular fins. The new fin geometry outperforms the traditional rectangular fins when comparing the thermal conductance per unit pumping power for a given heat exchanger volume over the range of Reynolds numbers studied.

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### 1. Introduction

Significant efforts have been made in the last decades to decrease the world's dependency to fossil fuels. One of the fronts which has shown major improvement is gas turbine efficiency. To this end, components such as recuperators have been developed to recover heat that is usually trapped and wasted in the exhaust gases of combustion processes. Relatively new heat exchanger designs have shown promises, with the potential to increase the thermal efficiency of these components from 75–85% to 90–95% [1]. Emerging recuperator designs have a common structure comprised of an internal heat transfer media with a flow barrier which prevents the mixing of the hot and cold flows and another stage of heat transfer enhancing features encased in an external shell, as illustrated in Fig. 1. Several geometries have been proposed for the internal heat transfer media, such as metal foams, lattice frames, packed beds, and wire mesh.

Wire mesh heat exchangers (WMHE) were first investigated in the late 1980s, with the folded wire mesh geometry [3,4]. This geometry is obtained by folding wire screen on itself and brazing the resulting structure to flow barriers (plates), as shown in

Fig. 2. This type of heat exchangers has the potential for a higher performance per unit volume and features low axial (along the flow direction) conduction losses compared to traditional heat exchangers. However, this configuration has its drawbacks. First, there is a maximum amount of times a wire mesh sheet can be folded on itself per linear inch, which depends on the mesh material and density [5]. Second, the small contact area per fold between the brazed sheet and the wire mesh sheet, as can be observed in Fig. 2 [5], creates a significant thermal resistance at this interface. The third disadvantage stems from the brazing process itself. This manufacturing technique is costly, mainly due to the use of a vacuum furnace, which is required to complete the joining process.

A new method for producing WMHE has recently been developed which mitigates these problems. Instead of folding the wire mesh onto itself, sheets are stacked and then cut perpendicular to the stacking direction, yielding thin wafers of wire mesh textiles. These wafers are then sealed using a dense metallic coating produced using thermal spray processes [5]. The applied coating thickness allows machining plain rectangular (straight cut) fins on its outer surface (Fig. 3). This design reduces the costs associated with producing WMHE by removing the costly brazing operation, but the external fin manufacturing is restricted by machining constraints. Furthermore, it is currently not economically viable to machine other types of fins, such as wavy offset or pin fin arrays.

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## Nomenclature

|                     |                                                    |             |                                                             |
|---------------------|----------------------------------------------------|-------------|-------------------------------------------------------------|
| $\Delta P_{outlet}$ | outlet differential pressure (Pa)                  | $H$         | fin height (m)                                              |
| $\Delta P_{fin}$    | fin differential pressure (Pa)                     | $h$         | convective heat transfer coefficient (W/(m <sup>2</sup> K)) |
| $\Delta P_{inlet}$  | inlet differential pressure (Pa)                   | $l_1$       | Bessel function of order one                                |
| $\Delta P_{tot}$    | total differential pressure (Pa)                   | $l_2$       | Bessel function of order two                                |
| $\Delta T_1$        | inlet temperature difference (K)                   | $k_m$       | fin material thermal conductivity (W/(m K))                 |
| $\Delta T_2$        | outlet temperature difference (K)                  | $L$         | sample length (m)                                           |
| $\Delta T_{lm}$     | log mean temperature difference (K)                | $\dot{m}$   | mass flow rate (kg/s)                                       |
| $\eta$              | fan efficiency                                     | $m$         | fin heat transfer parameter (m <sup>-1</sup> )              |
| $\eta_f$            | individual fin efficiency                          | $N_f$       | number of fins                                              |
| $\eta_o$            | overall fin efficiency                             | $P_{flow}$  | flow perimeter (m)                                          |
| $\theta$            | pyramid angle (°)                                  | $q$         | heat flux (W/m <sup>2</sup> )                               |
| $\mu$               | dynamic viscosity (Pa s)                           | $Re_{Dh}$   | Reynolds number based on hydraulic diameter                 |
| $\rho$              | fluid density (kg/m <sup>3</sup> )                 | $R_{eq}$    | equivalent thermal resistance (K/W)                         |
| $A_f$               | fin heat transfer area (m <sup>2</sup> )           | $S$         | space between fin edges (m)                                 |
| $A_{flow}$          | net flow area (m <sup>2</sup> )                    | $T$         | top fin length (m)                                          |
| $A_{tot}$           | total heat transfer area (m <sup>2</sup> )         | $T_{in}$    | inlet fluid temperature (K)                                 |
| $A_u$               | unfinned heat transfer area (m <sup>2</sup> )      | $T_{out}$   | outlet fluid temperature (K)                                |
| $B$                 | base fin length (m)                                | $U_{max}$   | maximum fluid velocity (m/s)                                |
| $C_p$               | fluid specific heat capacity (kJ/(kg K))           | $UA$        | thermal conductance (W/K)                                   |
| $D$                 | base diameter (m)                                  | $UA_v$      | thermal conductance per unit volume (kW/(m <sup>3</sup> K)) |
| $D_{mean}$          | mean base fin length (m)                           | $V$         | volume (m <sup>3</sup> )                                    |
| $d_h$               | hydraulic diameter (m)                             | $\dot{V}_f$ | volumetric flow rate (m <sup>3</sup> /s)                    |
| $e_v$               | pumping power per unit volume (kW/m <sup>3</sup> ) | $W$         | channel width (m)                                           |
| $FD$                | fin density (fin/m)                                |             |                                                             |

Metal foam heat exchangers enclosed in a metallic shell deposited using wire-arc or plasma spray have been investigated by Salimi Jazi et al. [6,7]. In this case, wire arc or plasma spray processes are used to seal commercially available nickel foam bricks with Inconel 625 to create fully encased heat exchangers. However, no heat transfer enhancing features, such as fins, are used on the outer surface of the heat exchangers.

Rectangular fin arrays provide an increase in heat transfer performance with respect to an unfinned surface due to the additional heat transfer area. Increases of the convective heat transfer coefficient due to this type of extended surface are usually neglected and the pressure loss through the array is low due to their geometry [8,9]. If the design specifications allow for a higher head loss, several other fin geometries can yield superior thermal performance for a given surface area. Wavy offset fins

increase the amount of area available for heat transfer while also increasing the amount of fluid mixing. This creates a fin array with a lower thermal resistance at the expense of a slightly higher head loss [8,9]. These fins are typically produced using sheet metal strips bent over themselves to form a wavy shape, with each strip slightly offset from the previous to induce mixing. Using pin fins instead will further increase the convective heat transfer coefficient by creating turbulent wakes behind each fin, which subsequently promotes fluid mixing [10]. This yields fin arrays which have better thermal performance at the expense of a higher pressure drop. Sahiti et al. [11–13] have demonstrated that pin fins offer the most effective way of increasing the heat transfer rate within a given heat exchanger volume, when compared to other types of fins.

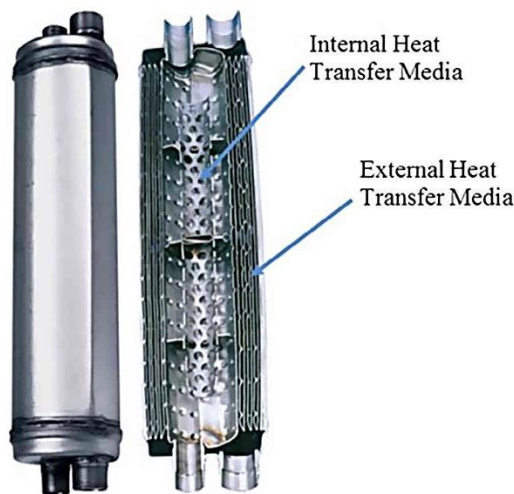


Fig. 1. Typical heat exchanger between two heat transfer media (adapted from [2]).

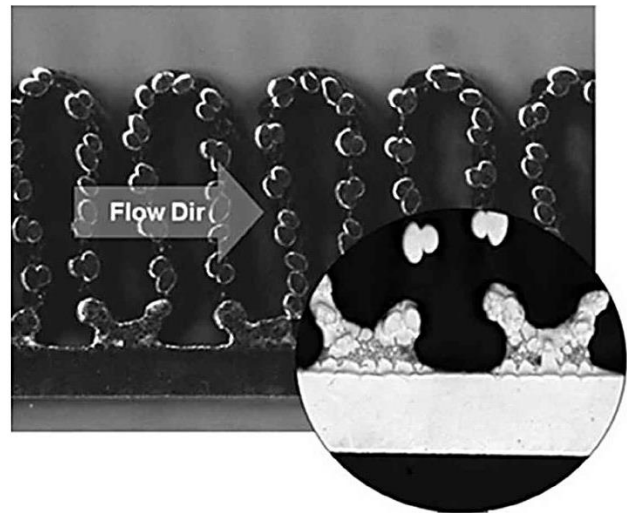


Fig. 2. Folded wire mesh with a zoomed section of the brazed plate contact surface.



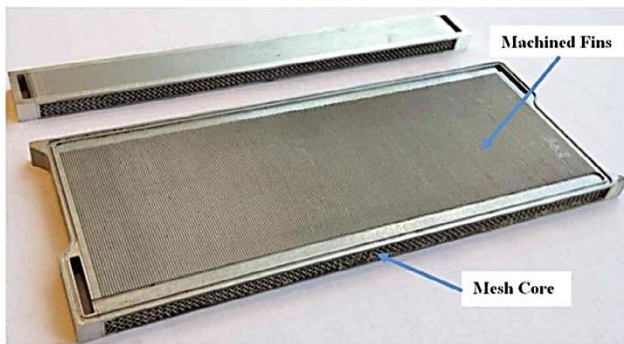


Fig. 3. Wire mesh heat exchanger with machined external fins on a mesh core.

Various types of pin fin array geometries were extensively studied. Staggered cylindrical pin fins were first investigated by Brigham and Van Fossen [14,15] and revealed that long pins fins (ratio of fin height to fin diameter is larger than 4) tend to transfer more heat than short pin fins. Work presented by Metzger et al. [16] detailed the effects of the array's orientation on the heat transfer and pressure loss through these arrays. Their results indicated that circular pins fins in an intermediate configuration between staggered and inline can reduce the pressure loss while increasing the amount of heat transferred. They also found that the surface convective heat transfer coefficient for pin fins was approximately 100% larger than that of the end wall. Since these landmark works, some attention has been given to pin fins with square cross-sections. You and Chang [17] studied the heat transfer and pressure loss properties of inline square pin fins confined in a rectangular channel using numerical modeling. Subsequently, Jeng and Tzeng [18] experimentally confirmed these numerical simulations of fluid flow through square based pin fins and determined that their performance can exceed that of circular pin fins.

The manufacturability of aluminum and stainless steel square base pyramidal pin fin arrays produced using Cold Gas Dynamic Spraying (CGDS or simply Cold Spray) was demonstrated by Cormier et al. [19]. Fin arrays were manufactured with fin densities varying between 12 and 30 fins per inch at fin heights of 1.0–2.2 mm. The thermal performance of the aluminum samples was determined and compared to a rectangular fin sample at 24 fins per inch, with the new type of fin having a higher thermal conductance than the traditional type of fin array. Much work is still left to be done to optimize the performance of this new type of fin geometry.

In this paper, the thermal and hydrodynamic performance of aluminum square base pyramidal and trapezoidal pin fin arrays produced with CGDS was investigated. The convective heat transfer coefficient and thermal conductance parameters were evaluated using a forced flow convective heat transfer apparatus at low Reynolds numbers (less than 3500). This fixture was also used to determine the pressure loss through the fin array, as well as the required pumping power. Investigations were limited to one configuration of uniformly distributed pin fins with a fin density of 12 fins per inch. The effect of varying the fin's base angle was characterized. Additionally, the performance of this new type of fin array was compared to traditional rectangular fin arrays with similar characteristic dimensions.

## 2. Experimental production technique description

### 2.1. CGDS process and masking technique

Thermal spray deposition processes, such as Plasma spraying and High Velocity Oxy Fuel (HVOF) spraying operate on the principle of using the thermal energy of a carrier gas to melt or partially melt solid powder particles and using the gas' kinetic energy to accelerate these particles onto a substrate in order to form a dense coating upon impact, deformation and solidification [20,21]. Kinetic spray processes such as Cold Gas Dynamic Spraying (CGDS), however, solely rely on the transfer of momentum from an inert gas, usually helium or nitrogen, to relatively cold particles (usually less than half the material's melting temperature) which are projected onto a substrate's surface [22]. A schematic of a CGDS apparatus is shown in Fig. 4. The high pressure driving gas is heated and is subsequently accelerated to supersonic speeds using a converging–diverging (DeLaval) nozzle, where its kinetic energy is imparted to the particles by drag effects [22,23]. Upon colliding with the substrate, the particle's kinetic energy is converted to plastically deform the particles and induce adhesion. This deformation is sustained by the adiabatic shear instabilities created during the impact, which causes the material to viscously flow outwards from the center of the particle [24]. Bonding to the substrate is attributed to localized high pressure waves breaking oxide layers and promoting intimate contact between the particle and the substrate [25].

Fig. 5 illustrates the technique used to create square based pyramidal fin arrays using commercially available plain woven, steel wire mesh (McMaster-Carr, Aurora, OH, USA) to mask the substrate. This configuration creates a pyramidal fin array (Fig. 6) which closely mimics the reversed geometry of the chosen wire mesh, with peaks aligning with the holes of the screened area

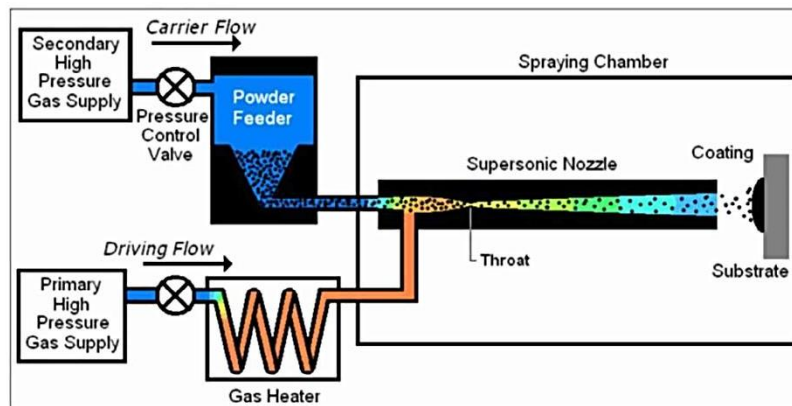


Fig. 4. Schematic of CGDS apparatus.



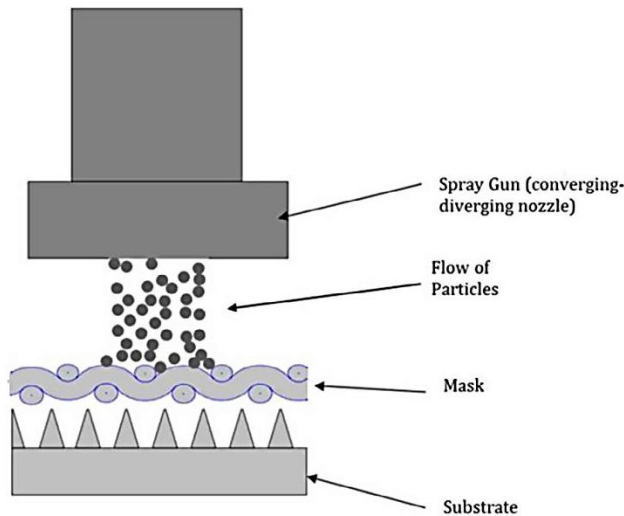


Fig. 5. Schematic of the spray process with masking technique.

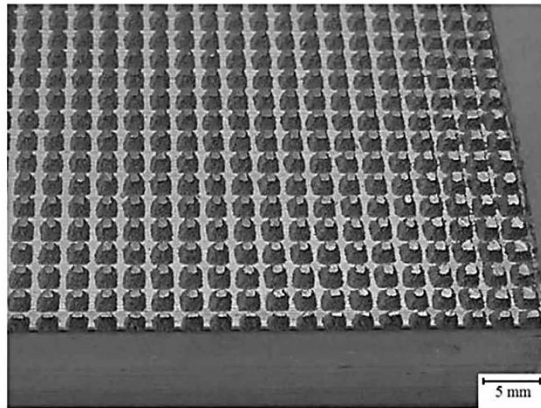


Fig. 6. Overview of typical fin array.

and little or no deposition underneath the wires [19]. The main factor that determines if adhesion occurs with the CGDS process is the particle's normal impact velocity with respect to the substrate. Thus, since the CGDS flow near the mask is affected by viscous effects (due to the presence of the mask walls and a shock wave present near the mask), particles present in the CGDS flow in the vicinity of the mask wires are slowed down. As such, the particles tend to build up a coating preferentially underneath the center of the mask holes, on the substrate. This creates a zone near the center of the holes where buildup is rapid when compared to the sides, leading to the creation of an angle between the top and the base, forming a pyramidal shape. When this angle becomes sharp enough, particles do not have a sufficient normal impact velocity and adhesion stops as particles simply bounce off. The working principles of the CGDS process make it an ideal candidate for the proposed masking technique used to produce pyramidal fins. Using other thermal spray techniques would cause issues where the mesh would either melt or be clogged by the molten particles in the high temperature gas stream. Manufacturing parameters were chosen to ensure that the dimensions of the pyramid base would follow those of the wire mesh mask as closely as possible.

The CGDS system used for this work is a commercially available Plasma Giken PCS-1000 [26]. The spray nozzle has a 3.0 mm throat

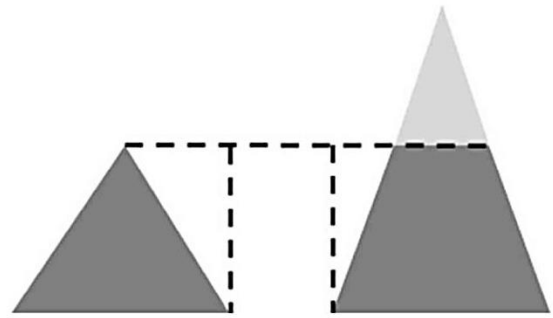


Fig. 7. Pyramidal vs trapezoidal geometries, showing the difference in base angle.

diameter and a 6.5 mm exit diameter. The feedstock powder material used is commercially pure aluminum (99.8% Al) from Centerline (Windsor) Ltd. (Windsor, Ontario, Canada). It has an elongated shape with a mean particle size ( $D_{50}$ ) of 25.35 microns. Other details about powder morphology and size distribution are given elsewhere [19].

## 2.2. Test sample preparation

Fins were cold sprayed on aluminum substrates (Al6061 T6) with standard dimensions of 5.1 cm by 5.1 cm. The cold spray trials detailed in this work were done on extruded aluminum substrates instead of thermal sprayed coatings on the WMHE to isolate the effects of the fin parameters on the heat transfer and hydrodynamic performance of the test samples. The substrates were cleaned and degreased using acetone after being cut to size. No further surface preparation steps, such as grit-blasting or heat treatment, was performed. To determine the effect of changing the fin base angle for a constant height, samples were cold sprayed to a height larger than the target height and then ground down to a uniform height of 1.0 mm using a conventional sanding belt with 320 grit sandpaper and water as coolant. This process yielded fins with a trapezoidal prism shape, as illustrated in Fig. 7. No other type of post-treatment of the sprayed samples was performed.

## 3. Testing procedure

### 3.1. Performance test apparatus

Performance of the various fin arrays produced was assessed using an in-house heat transfer and pressure drop test apparatus, adapted from the literature [19,27]. A schematic of the apparatus and its various components is shown in Fig. 8.

The main components of this apparatus are the test section, the heating pad, the air supply and the data acquisition system. The air is supplied by a compressed air line, properly filtered and dehumidified to obtain room temperature, dry air, free from oil contamination. The air inlet flow rate is controlled by a needle valve and is measured with a flow meter. This air is smoothly expanded into a horizontal rectangular channel with a diffuser and a honeycomb flow straightener. This rectangular channel's top and bottom plates (76.2 mm wide by 457.2 mm long by 19.05 mm thick) are made of acrylic, and they are spaced apart using a rectangular rubber gasket, which is compressed to ensure there is no clearance between the fins and the shroud surface. These gaskets also ensure that the joint is leak-proof. A length of 457 mm of unobstructed flow ensures that the fluid is fully developed hydrodynamically before reaching the test section for the given hydraulic diameter and Reynolds number [8].



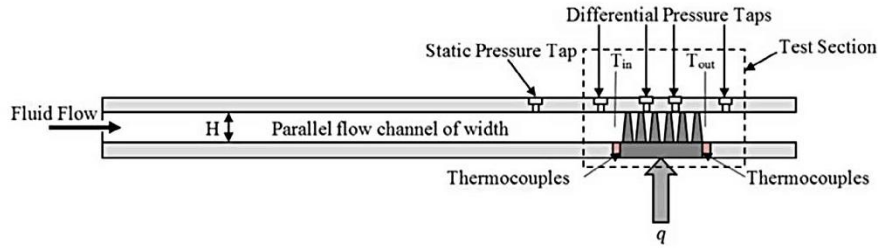


Fig. 8. Schematic of the heat transfer test fixture.

The test sample is subjected to a constant heat flux provided by a strip heater. A conducting block is used to ensure that heat is evenly distributed onto the substrate's surface, resulting in a constant surface temperature under free convection conditions.

A stagnation pressure measurement tap is located 25.4 mm upstream from the test section, while two sets differential pressure measurement taps are located 31.8 and 12.7 mm upstream (high pressure side) and downstream (low pressure side) from the middle of the test section. Two thermocouples are located 6 mm before and after the test section at mid-height to record the fluid local average temperature. Four thermocouples are attached to the fin array's base, with two on the side facing the incoming flow, and two on the side facing the exiting flow. On each face, one thermocouple is located on the centerline, while the other thermocouple is located 6 mm from the wall. All the thermocouples used are T-type, butt-bonded, thin gauge (28 gauge) thermocouples. Radiative heat transfer losses are neglected since they would account for less than 0.5% of the total heat input for similar test surfaces [28,29].

### 3.2. Heat transfer testing procedure

Measurements were performed at steady-state conditions and repeated until statistical significance was achieved. Data samples were taken over a 30 s interval, at 2 Hz. Five sets of data samples were recorded for every flow condition. Measurements were taken at flow rates corresponding to Reynolds numbers varying from 200 to 2000, as this is the typical application range sought for these exchangers. Deviation of the measurements from the mean value was less than 5%. The accuracy of the test fixture was validated with the experimental results of Kays' and London's work [9]. The error bars are not shown in the results section as they are smaller than the markers. A detailed description of the apparatus' accuracy and instrument uncertainties is given in Cormier et al. [19].

### 3.3. Pressure loss testing procedure

Measurements were performed at steady-state conditions and repeated until statistical significance was achieved. Five sets of data samples were recorded for every flow condition. Measurements were taken at flow rates corresponding to Reynolds numbers varying from 200 to 3500, to identify the trends and to obtain sufficient data for predictive modeling. Deviation of the measurements from the mean value was less than 5%. For hydrodynamic performance tests, one set of pressure taps is used to obtain the fin array friction losses, which include skin friction and form drag, while another set is used to determine the total pressure drop of the fin array, which includes entrance and exit effects as well as fin array friction losses. This allows the measurement of the fin array friction loss independently of the flow constriction and expansion pressure losses. Measurements were limited to Reynolds numbers of approximately 2500 for the total pressure loss due to instrumentation limits.

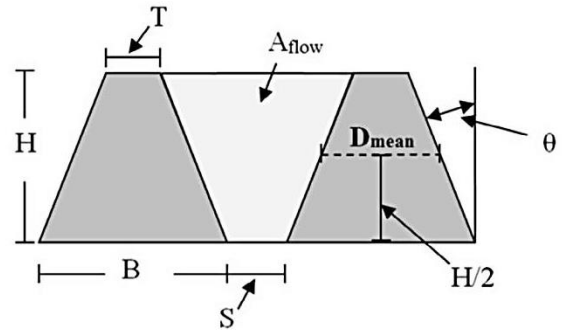


Fig. 9. Schematic of the fin array unit with geometric variables identification.

### 3.4. Data analysis method

To analyze the raw data obtained, flow conditions are expressed using the Reynolds number. Since the test samples have the same base dimensions, performance evaluations are done by comparing convective heat transfer coefficients and the thermal conductances. Fig. 9 illustrates the geometric variables used to perform the analysis.

The Reynolds number based on the hydraulic diameter is calculated with:

$$Re_{Dh} = \frac{\rho U_{max} d_h}{\mu} = \frac{\dot{m} \cdot d_h}{W \cdot FD \cdot A_{flow} \cdot \mu} \quad (1)$$

where  $\dot{m}$  is the mass flow rate,  $d_h$  is the hydraulic diameter of the fin array flow passage,  $W$  is the fin array sample width,  $FD$  is the fin density,  $\mu$  is the fluid's dynamic viscosity and  $A_{flow}$  is the face area the fluid can flow into between two fins. The variables  $d_h$  and  $A_{flow}$  are given by:

$$d_h = \frac{4 \cdot A_{flow}}{P_{flow}} = \frac{2 \cdot A_{flow}}{\frac{H}{\cos(\theta)} + S + H \cdot \tan(\theta)} \quad (2)$$

$$A_{flow} = (S + H \cdot \tan(\theta)) \cdot H \quad (3)$$

where  $\theta$  is the fin angle,  $S$  is fin spacing from edge to edge at the base,  $H$  is the fin height and  $P_{flow}$  is the flow channel's perimeter.

The fin array's convective heat transfer coefficient is calculated from the raw data with:

$$h = \frac{q}{\Delta T_{lm} A_{tot} \eta_o} \quad (10)$$

In these equations,  $q$  is the heating rate,  $\Delta T_{lm}$  is the log-mean temperature difference between the fluid and the fin array and  $A_{tot}$  is the total heat transfer area. The total heat input rate to the system is calculated using:

$$q = \dot{m} \cdot (T_{out} - T_{in}) \cdot Cp \quad (12)$$



where  $(T_{out} - T_{in})$  is the fluid temperature difference between the entry and the exit of the test section,  $C_p$  is the fluid's thermal capacitance. The log-mean temperature difference is calculated with:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (11)$$

where  $\Delta T_1$  and  $\Delta T_2$  are the temperature difference between the fin surface and the flow at the inlet and outlet respectively. The total heat transfer area is given by:

$$A_{tot} = A_f + A_u = N_f \left( \frac{B^2}{\tan(\theta) \cos(\theta)} - \frac{T^2}{2 \sin(\theta)} \right) + (WL - N_f B^2) \quad (5)$$

In the previous equation,  $N_f$  is the number of fins in the array,  $B$  is the fin's length at its base,  $L$  is the test sample's length,  $A_f$  designates the total fin heat transfer area,  $A_u$  denotes the unfinned base's exposed area while  $T$  is the fin's top length, given by:

$$T = B - 2H \tan(\theta) \quad (6)$$

The overall efficiency is calculated using the following equation:

$$\eta_o = 1 - \frac{A_f}{A_{tot}} (1 - \eta_f) \quad (7)$$

where  $\eta_f$  is the individual fin efficiency, defined as the heat transferred by the fin over the maximum possible heat that can be transferred, which is numerically equal to:

$$\eta_f = \frac{2}{mH} \frac{I_2(2mH)}{I_1(2mH)} \quad (8)$$

with:

$$m = \sqrt{\frac{4h}{k_m D}} \quad (9)$$

In the previous equations adapted from Incropera [8],  $I_1$  and  $I_2$  denote the first and second order Bessel functions, respectively, while  $k_m$  is the thermal conductivity of the fin's metal. The individual fin efficiency equation for triangular pin fins are used for the geometries in this work since it has the most similar geometry. The only difference between the two is the shape of the base, as the equation was developed for circular base cones while the manufactured fins present a square base. Therefore, the diameter of the fin ( $D$ ) in the previous equation is replaced by the fin length at the base ( $B$ ).

The thermal conductance is computed with:

$$UA = \frac{1}{R_{eq}} = h \cdot A_{tot} \cdot \eta_o \quad (4)$$

where  $\eta_o$  is the overall surface efficiency.

The total pressure loss measured over the fin array can be divided as:

$$\Delta P_{tot} = \Delta P_{inlet} + \Delta P_{fin} + \Delta P_{outlet} \quad (13)$$

where  $\Delta P_{tot}$  is the total pressure loss through the fin array,  $\Delta P_{inlet}$  is the pressure loss due to flow contraction at the inlet of the array,  $\Delta P_{fin}$  is the pressure loss due to skin friction and flow separation through the fin array and  $\Delta P_{outlet}$  is the pressure gained from the expansion of the flow at the outlet of the fin array.

To obtain the efficiency curves produced in Section 4.3, the required fan power input per unit volume was calculated using:

$$e_v = \frac{\dot{V}_f \cdot \Delta P_{fin}}{\eta \cdot V} \quad (14)$$

In the previous relation, adapted from Sahiti's work [12],  $e_v$  is the pumping power per unit volume,  $\dot{V}_f$  is the volumetric flow rate of the fluid,  $\eta$  is the fan efficiency while  $V$  is the volume of the heat

**Table 1**

1.0 mm Pyramidal fin dimensions (in microns) of tested samples for the spray angle study.

| Sample | B       |                 | S       |                 | H       |                 |
|--------|---------|-----------------|---------|-----------------|---------|-----------------|
|        | Average | SD <sup>a</sup> | Average | SD <sup>a</sup> | Average | SD <sup>a</sup> |
| 1.0    | 1159    | 51              | 932     | 39              | 1006    | 69              |
| M1.3   | 1394    | 81              | 821     | 55              | 901     | 41              |
| M1.5   | 1586    | 67              | 490     | 67              | 1058    | 55              |
| M1.8   | 1429    | 67              | 780     | 53              | 953     | 40              |
| M2.0   | 1559    | 79              | 548     | 55              | 1075    | 43              |
| M2.4   | 1626    | 70              | 583     | 79              | 954     | 45              |

<sup>a</sup> SD: standard deviation.

**Table 2**

Important geometric variables of pyramidal and trapezoidal samples.

| Sample | Hydraulic diameter mm | Total heat transfer area mm <sup>2</sup> | $D_{mean}$ mm | $H/D_{mean}$ ratio | Base angle ° |
|--------|-----------------------|------------------------------------------|---------------|--------------------|--------------|
| 1.0    | 1.12                  | 3372                                     | 0.61          | 1.63               | 58.5         |
| M1.3   | 1.02                  | 3637                                     | 0.81          | 1.24               | 60.3         |
| M1.5   | 0.96                  | 3894                                     | 1.07          | 0.93               | 61.8         |
| M1.8   | 0.95                  | 3907                                     | 1.13          | 0.89               | 66.6         |
| M2.0   | 0.94                  | 4071                                     | 1.17          | 0.86               | 68.7         |
| M2.4   | 0.93                  | 4162                                     | 1.22          | 0.82               | 72.6         |

**Table 3**

Samples tested for comparison.

| Sample number | Fin density Fins per in | Fin width mm | Fin spacing mm | Height mm | Hydraulic diameter mm | Total heat transfer area mm <sup>2</sup> |
|---------------|-------------------------|--------------|----------------|-----------|-----------------------|------------------------------------------|
| US            | N/A                     | N/A          | N/A            | 2.0       | 3.85                  | 2581                                     |
| SC            | 12                      | 1.23         | 0.89           | 1.0       | 0.94                  | 3478                                     |

exchanger. A fan efficiency of 0.8 was arbitrarily chosen as a reasonable value to perform these calculations.

### 3.5. Test sample geometric data

Table 1 presents a list of the 1.0 mm height pin fin arrays produced and tested, with the average base ( $B$ ), spacing ( $S$ ) and height ( $H$ ) dimensions and their associated standard deviations measured for 25 fins per sample. Fin array samples were measured using a depth of field microscope VHX-2000 which was acquired from Keyence (Mississauga, Ontario, Canada) which allows building three dimensional images of the given samples. The number next to the prefix "M" indicates the original height of the ground sample. For all pyramidal fin samples, the fin density was set at 12 fins per inch. Table 2 shows the important geometric variables for pyramidal and trapezoidal samples calculated using the data from Table 1. Table 3 shows a list of comparison/reference samples. The sample labeled SC (straight cut) has rectangular fins machined from bulk Al 6061 while the sample labeled US is an unfinned Al 6061 reference sample. Pictures of these two types of samples are shown in Fig. 10. Sample SC was used to compare with the pin fin samples, having comparable base length and fin spacing with a 90° base angle.

## 4. Results

### 4.1. Heat transfer performance

Fig. 11 shows the thermal conductance of the different fin arrays tested as a function of the Reynolds number. The results reveal



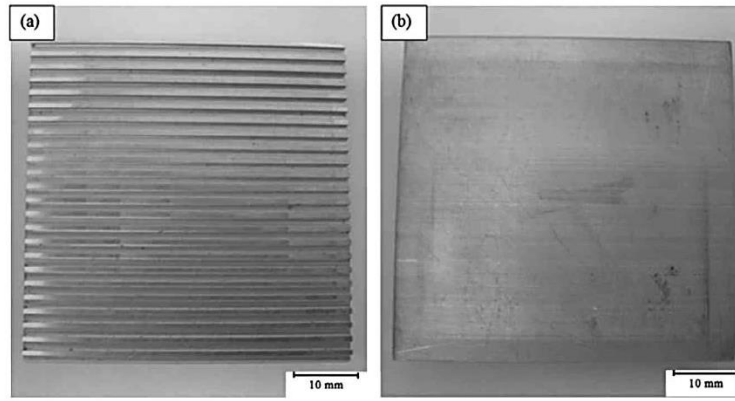


Fig. 10. Comparison samples (a) straight cut (SC) machined from bulk Al 6061 (b) unfinned surface (US) of bulk Al 6061.

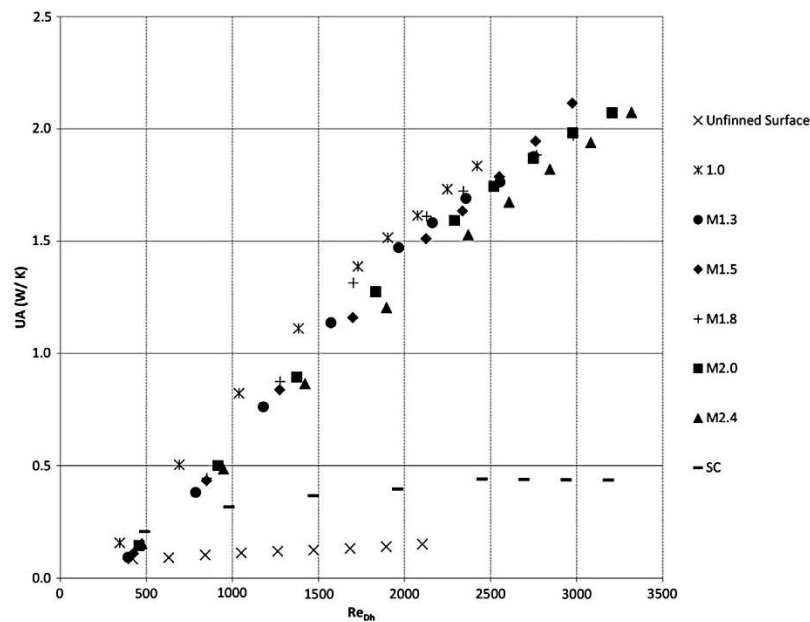
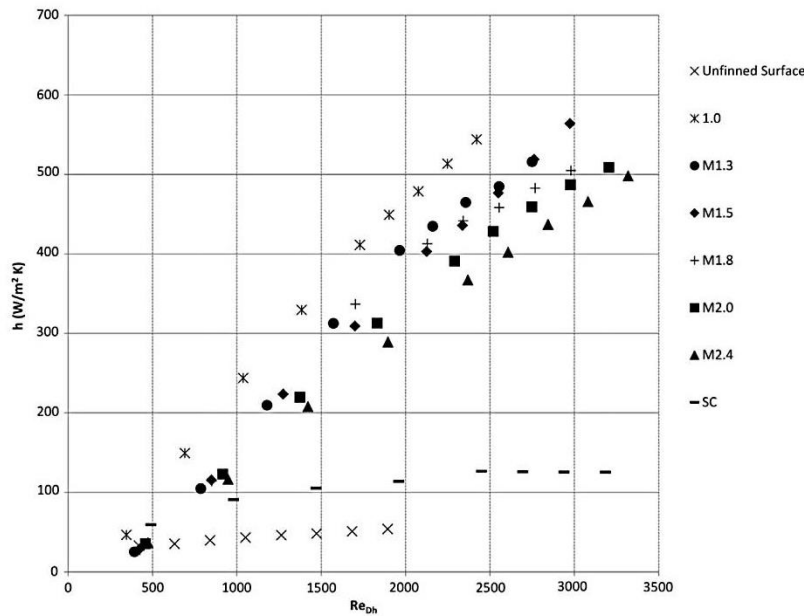


Fig. 11. Thermal conductance comparison between 1.0 (star), ground from 1.3 (circle), ground from 1.5 (diamond), ground from 1.8 (cross), ground from 2.0 (square), ground from 2.4 (triangle) and rectangular fin array (line) while the unfinned surface is used as reference (X).

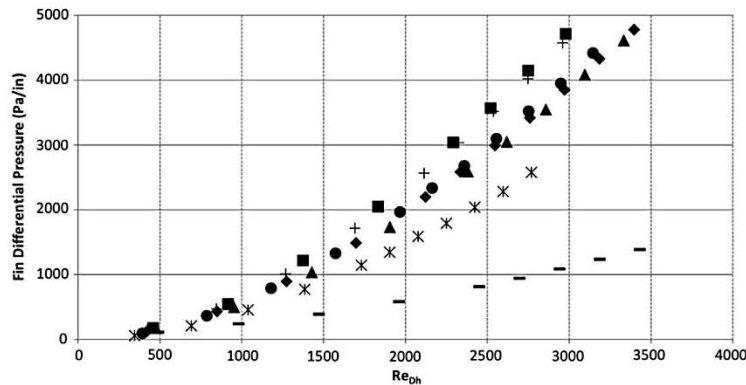
little variations between the pyramidal/trapezoidal fin samples at all Reynolds number tested. It is observed that all the results could be grouped around a line of thermal conductance which increases linearly with Reynolds number up to approximately 1500–2000  $Re_{Dh}$ . The trend seems to remain linear past that Reynolds number but with a reduced slope. These results suggest that there is little effect in changing the pyramid base angle at a given fin height on the thermal conductance of the fin array. However, a decrease of the convective heat transfer coefficient values can be observed in Fig. 12 as the base angle increases. Sample 1.0 has the highest convective coefficient value, which is reduced with each increase in initial fin height up to sample M2.4. This effect is moderated by an increase in the total heat transfer area available, explaining why the fin array thermal conductance can be considered independent of the fin base angle. The decrease of convective heat transfer coefficient observed is believed to be caused by a difference in the flow structure around the fins, with steeper base angles leading to a reduction of the amount of fluid mixing, and consequently of the convective heat transfer coefficient. The performance of a pyrami-

dal type fin would approach that of a completely rectangular pin fin as the base angle is increased.

Examining the results from the rectangular fin array, one can observe that its thermal conductance is significantly lower than those of the pyramidal/trapezoidal fin arrays. This is expected due to the different flow structures between continuous geometries such as plain rectangular fins and pin fin geometries, which are discontinuous. These discontinuities are believed to cause turbulent wakes behind each column of fins, which increases fluid mixing and consequently increases the convective heat transfer coefficient. This is supported by Fig. 12, which shows a significant increase in the convective heat transfer coefficient between the two types of fins. This difference in convective coefficient reaches up to 300% increase at 2500  $Re_{Dh}$  between the pyramidal sample 1.0 and the rectangular fin array, with increasing fin angle diminishing this difference to a 200% increase when comparing to the M2.4 sample at the same Reynolds number. A plateau of thermal performance is noted for the plain rectangular fin sample, which begins at 1000  $Re_{Dh}$  and extends to 3000  $Re_{Dh}$ . This plateau has a



**Fig. 12.** Convective heat transfer coefficient comparison between 1.0 (star), ground from 1.3 (circle), ground from 1.5 (diamond), ground from 1.8 (cross), ground from 2.0 (square), ground from 2.4 (triangle) and rectangular fin array (line) while the unfinned surface is used as reference (X).



**Fig. 13.** Fin differential pressure comparison between 1.0 (star), ground from 1.3 (circle), ground from 1.5 (diamond), ground from 1.8 (cross), ground from 2.0 (square), ground from 2.4 (triangle) and rectangular fin array (line) while the unfinned surface is used as reference (X).

similar rate of increase of convective coefficient as an unfinned surface, which is to be expected as the major advantage of adding plain rectangular fins is not the increase in convective coefficient, which can usually be neglected, but the increase in total heat transfer area [8].

#### 4.2. Hydrodynamic performance

The pressure loss through the arrays of the different fin geometries is shown in Fig. 13, while the total pressure loss including the entrance and exit effects is shown in Fig. 14. There is little difference between the hydrodynamic performances of the trapezoidal samples, which have a very similar pressure loss over the range of Reynolds number studied. The pyramidal sample, however, shows a slightly lower pressure loss than the trapezoidal samples. This is attributed to the slight clearance between the top of the fins and the shroud of the testing set-up, which stems from the variability in the manufacturing process of this type of fin array. The ground (trapezoid) fins have a very uniform height due to the

material removal process, while the variability between the maximum and the minimum height of the pyramidal sample is more significant as shown in Table 1. This variability is believed to cause a clearance between the average fins and the shroud, which can reduce the pressure loss significantly. The results suggest that the convective heat transfer coefficient is not affected as much, however. The total pressure loss follows the same trends as the fin array pressure losses, which indicates that these observations are not due to entrance and exit effects.

The pressure loss of the rectangular fin sample is much lower than that of the pin fins tested, which is expected. The reduction in fluid mixing and turbulence zones allows a smooth flow along the length of the fins. Furthermore, pin fins have a large form drag pressure loss due to their discontinuity along the flow length, a feature which is not present for plain rectangular fins, which are continuous along the flow direction. The pressure losses due to the inlet and outlet effects are consistently measured to be approximately 30% of the total pressure loss through the pin fin arrays.



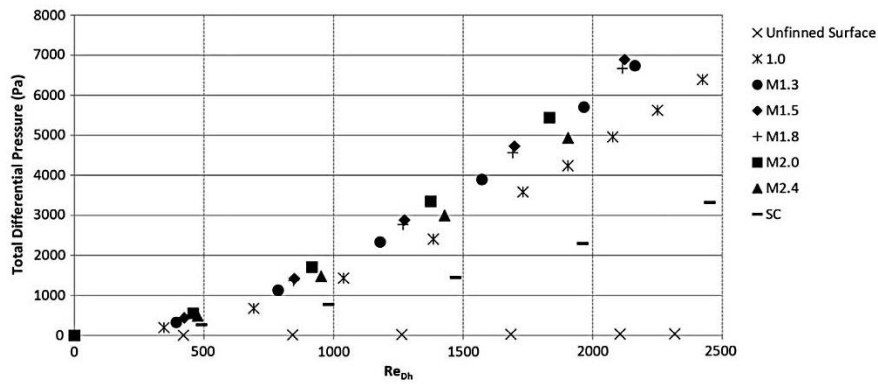


Fig. 14. Total differential pressure comparison between 1.0 (star), ground from 1.3 (circle), ground from 1.5 (diamond), ground from 1.8 (cross), ground from 2.0 (square), ground from 2.4 (triangle) and rectangular fin array (line) while the unfinned surface is used as reference (X).

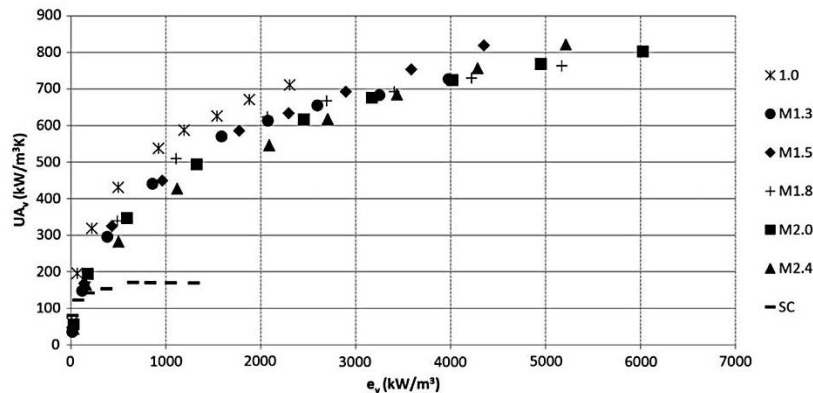


Fig. 15. Heat transfer surface efficiency comparison between 1.0 (star), ground from 1.3 (circle), ground from 1.5 (diamond), ground from 1.8 (cross), ground from 2.0 (square), ground from 2.4 (triangle) and rectangular fin array (line).

#### 4.3. Heat transfer surface efficiency

To determine which fin array transfers heat more efficiently, an efficiency indicator must be selected. Based on an extensive literature review performed, the method proposed in Sahiti's work [12] was selected. This method is based on plotting the thermal and hydrodynamic data as a curve of thermal conductance per unit of fin array volume as a function of the required pumping power per unit volume for different flow conditions. This yields curves which can be directly compared one to another, with the most efficient fin array having a higher thermal conductance at a given pumping power.

Fig. 15 shows the efficiency indicator plotted for the different types of fin arrays tested. The performances of the pin fins are very similar over the range of test conditions, which indicates that little performance is lost when pyramidal fins are ground down to a trapezoid shape. An inflection point is noted for the pin fin samples, located near 500 kW/m³. This inflection point denotes the zone where this type of fin array performs best, i.e. where the maximum amount of heat is transferred per unit pumping power. Further increasing the Reynolds number past this point leads to increasingly large pressure losses for little heat transfer gain. These facts lead to the conclusion that in this case manufacturing concerns should be accommodated, with the following production method: Fins should be produced several hundred microns higher than the target height and the top of the fins should be removed by grinding, which will allow the quality of the array to better be controlled. This process will ensure that all the fins over a sample are

at the same height and that there is no bypass flow between the top of the fins and the shroud, which could have detrimental effects on fin array performance.

It can be observed that the performance of the plain rectangular fins is similar to that of pyramidal and trapezoidal fins at low Reynolds number, but rapidly peaks once their maximal thermal performance is reached. Indeed, at 1000 kW/m³, a 300% increase in performance is attained when using pyramidal pins fins instead of plain rectangular fins for the same heat exchanger volume. This is due to the plateau of thermal performance that begins at about 1000  $Re_{Dh}$  for rectangular fins, which was detailed in Section 4.1. The same type of inflection point is found in the performance curve of the rectangular fin array as in the pin fin arrays. This inflection, however, is located much earlier in the performance curve due to low pumping powers required by the rectangular fin array.

#### 5. Summary and conclusion

This study investigates the thermal and hydrodynamic performance of pyramidal and trapezoidal fin arrays. The new geometries proposed are produced using Cold Gas Dynamic Spraying, with the former type being left as-sprayed while the latter type was sprayed and subsequently ground down to size. This operation removed the top part of the pyramid to yield fins with a trapezoid prism shape. This investigation concentrated on samples of 12 fins per inch at 1.0 mm height. The performances of these fins are compared using the thermal conductivity over the required pumping



power per unit volume. The performances of these new types of fin arrays are also compared to traditional plain rectangular fin arrays. The conclusions of this work are summarized as:

1. There is a small difference in the thermal conductance when fins are ground down to the same height as an as-sprayed sample. This small difference is due to a decreasing convective heat transfer as the pyramid base angle increases, while the total available heat transfer surface area increases as the base angle increases. Since these two parameter offer a conflicting effect but change at a similar rhythm with the base angle parameter, there is little net change in the thermal conductance.
2. The hydrodynamic performance of the new types of fin is similar over the range of Reynolds numbers studied, with the pyramidal sample having the lowest pressure loss. This is believed to be caused by clearance induced by the manufacturing process used, and will be confirmed in a later publication which will focus on the flow structures found in these types of fin arrays.
3. The global performance of the trapezoidal and pyramidal fins was found to be very similar over the range of pumping power studied. This result confirms that removing the top of the fins by grinding is a viable way of controlling the quality of the end product without compromising the performance of the fin array. Using an industrial process for grinding could also tighten the tolerances and further ensure repeatability of production.
4. The new types of fins have a better performance value than traditional plain rectangular fins over the range of Reynolds number studied ( $100 < Re_{Dh} < 3500$ ). This is due to the large increase in convective heat transfer coefficient due to the turbulence between the rows of fins, which increase fluid mixing. This gain is offset by the increased pressure loss through the discontinuous fin array. The increased thermal performance, however, still ensures that the total performance of the new type of fin array is superior to that of a comparable rectangular fin array.

## Acknowledgements

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## References

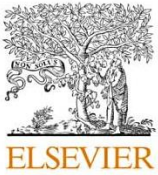
- [1] D. Bohn, Micro gas turbine and fuel cell – a hybrid energy conversion system with high potential, NATO Research and Technology Organisation, 2005.
- [2] Process Technology, TherMax<sup>2</sup> Heat Exchangers, <http://process-technology.com>, Consulted April 2013.
- [3] R.L. Shaner, US Patent No. 4840228, 1989.
- [4] J. Chisholm, US Patent No. 4843693, 1989.
- [5] J. Assaad, A. Corbeil, P. Richer, B. Jodoin, Novel stacked wire mesh compact heat exchangers produced using cold spray, *J. Therm. Spray Technol.* 20 (6) (2011) 1192–1200.
- [6] H.R. Salimi Jazi, J. Mostaghimi, S. Chandra, L. Pershin, T. Coyle, Spray-formed metal-foam heat exchangers for high temperature applications, *J. Therm. Sci. Eng. Appl.* 1 (3) (2009) 1–7.
- [7] F. Azarmi, J. Saaedi, T.W. Coyle, J. Mostaghimi, Microstructure characterization of alloy 625 deposited on nickel foam using air plasma spraying, *Adv. Eng. Mater.* 10 (5) (2008) 459–465.
- [8] F.P. Incropera, D.P. DeWitt, T.L. Bergman, A.S. Lavine, Fundamentals of Heat and Mass Transfer, sixth ed., Wiley, New York, 2006.
- [9] W.M. Kays, A.L. London, Compact Heat Exchangers, third ed., McGraw-Hill, New York, 1984.
- [10] A. Žukauskas, Heat transfer from tubes in crossflow, *Adv. Heat Transfer* 8 (1972) 93–160.
- [11] N. Sahiti, F. Durst, A. Dewan, Heat transfer enhancement by pin elements, *Int. J. Heat Mass Transfer* 48 (2005) 4738–4747.
- [12] N. Sahiti, F. Durst, A. Dewan, Strategy for selection of elements for heat transfer enhancement, *Int. J. Heat Mass Transfer* 49 (2006) 3392–3400.
- [13] N. Sahiti, A. Lemouedda, D. Stojkovic, F. Durst, E. Franz, Performance comparison of pin fin in-duct flow arrays with various pin cross-sections, *Appl. Therm. Eng.* 26 (2006) 1176–1192.
- [14] B.A. Brigham, G.J. Vanfossen, Length to diameter ratio and row number effects in short pin fin heat transfer, *ASME J. Eng. Gas Turbines Power* 104 (1984) 241–244.
- [15] G.J. Vanfossen, Heat-transfer coefficients for staggered arrays of short pin fins, *ASME J. Eng. Gas Turbines Power* 102 (1982) 268–274.
- [16] D.E. Metzger, C.S. Fan, S.W. Haley, Effects of pin shape and array orientation on heat transfer and pressure loss in pin fin arrays, *ASME J. Eng. Gas Turbines Power* 106 (1984) 252–257.
- [17] H.I. You, C.H. Chang, Numerical prediction of heat transfer coefficient for a pin-fin channel flow, *ASME J. Heat Transfer* 119 (1997) 840–843.
- [18] T.M. Jeng, S.C. Tzeng, Pressure drop and heat transfer of square pin-fin arrays in in-line and staggered arrangements, *Int. J. Heat Mass Transfer* 50 (2007) 2364–2375.
- [19] Y. Cormier, P. Dupuis, B. Jodoin, A. Corbeil, Net shape fins for compact heat exchanger produced by cold spray, *J. Therm. Spray Technol.* 22 (7) (2013) 1210–1221.
- [20] J.R. Davis (Ed.), Handbook of Thermal Spray Technology, ASM International, 2004.
- [21] H. Herman, S. Sampath, R. McCune, Thermal spray: current status and future trends, *Mater. Res. Soc. Bull.* 25 (7) (2000) 17–25.
- [22] A.P. Alkhimov, A.N. Papyrin, V.F. Dosarev, N.I. Nesterovich, M.M. Shuspanov, US Patent No. 5302414, 1994.
- [23] A.O. Tokarev, Structure of aluminum powder coatings prepared by cold gas-dynamic spraying, *Metal Sc. Heat Treat.* 38 (3–4) (1996) 136–139.
- [24] M. Grujicic, C.L. Zhao, W.S. DeRosset, D. Helfrich, Adiabatic shear instability based mechanism for particles/substrate bonding in the cold-gas dynamic-spray process, *Mater. Des.* 25 (2004) 681–688.
- [25] R.C. Dykhuizen, M.F. Smith, D.L. Gilmore, R.A. Neiser, X. Jiang, S. Sampath, Impact of high velocity cold spray particles, *J. Therm. Spray Technol.* 8 (4) (1999) 559–564.
- [26] Plasma Giken, <<http://www.plasma.co.jp/en/>>, Consulted November 2012.
- [27] J. Tian, T. Kim, T.J. Lu, H.P. Hodson, D.T. Queheillalt, D.J. Sypeck, H.N.G. Wadley, The effects of topology upon fluid-flow and heat-transfer within cellular copper structures, *Int. J. Heat Mass Transfer* 47 (14–16) (2004) 3171–3186.
- [28] S. Naik, S.D. Probert, M.J. Shilston, Forced convective steady-state heat transfers from shrouded vertical fin arrays aligned parallel to an undisturbed air-stream, *Appl. Energy* 26 (1987) 137–158.
- [29] J.J. Hwang, T.M. Liou, Heat transfer and friction in a low-aspect-ratio rectangular channel with staggered perforated ribs on two opposite walls, *ASME J. Heat Transfer* 117 (11) (1995) 843–850.

### **5.3 Paper #3: Height and Fin Density Effect**

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The focus of this paper, entitled “Additive manufacturing of pyramidal pin fins: Height and fin density effects under forced convection”, was to determine the effect of varying the fin height and fin density on the thermal and hydrodynamic performance of the fin array, as well as to build correlations on the Nusselt number for the industrial partner. It was found that increasing the fin height or the fin density promotes the thermal conductance but at the expense of a higher pressure loss. Empirical correlations linking the Nusselt number, the Reynolds number and the fin height or the fin density were produced and validated. This peer-reviewed article addresses the objective of the effect of fin height and fin density presented in section 3.4.3.





# Additive manufacturing of pyramidal pin fins: Height and fin density effects under forced convection

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Pressure loss

Heat exchanger

Forced convection

Cold spray

## ABSTRACT

This paper investigates the effects of varying the fin height and the fin density of pyramidal pin fins produced using cold spray technology. The thermal and hydrodynamic performance of this new type of fin array is evaluated using a forced convection heat transfer apparatus. The heat transfer efficiency of this type of extended surface is determined and correlations linking the Nusselt number, the Reynolds number, the fin height and the fin density are produced and validated. The geometric and thermo-hydraulic parameters affecting the thermal conductance of the pyramidal pin fin arrays are discussed. It is found that increasing either the fin height or the fin density also increases the total thermal conductance at the expense of a higher pressure loss.

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## 1. Introduction

Increasing heat transfer has been a frontline concern in the last fifty years in many fields such as the cooling of sensitive electronic equipment. With the miniaturization of electronic packages, the power density has increased while the amount of heat transfer area on such components is decreasing, making the development of effective heat sinks a priority [1]. In such applications, the heat sinks should dissipate as much heat as possible to ensure the reliability and life of critical components, while allowing for the most compact package size possible. The development of more efficient heat transfer surfaces could also bring benefits to other areas such as the cooling of gas turbine blades or the production of compact heat exchangers, where the same challenge of increasing the amount of heat transfer per unit volume at a given head loss is encountered [2–4].

To this end, pin fins have replaced traditional continuous fin shapes such as rectangular or wavy fins in critical applications due to their higher heat transfer rate [5,6]. The increased heat transfer attained by pin fins is often partially offset by higher pressure losses through the fin array, but yields a better overall

performance than their continuous counterparts [7–10]. Sahiti et al. [8–10] have demonstrated that pin fins offer the best performance for a given pumping power and heat exchanger volume. This was explained by the fact that unlike continuous heat transfer features, pin fins not only increase the available heat transfer area, but also the convective heat transfer coefficient of the fin array.

Various pin fin array parameters have been investigated over the past thirty years. The relative height parameter ( $H/d$ , where  $H$  is the fin height and  $d$  is the pin fin diameter) of circular pin fins was one of the first parameters to be detailed by Brigham and Van Fossen [11,12]. Their study revealed that for cylindrical staggered fins, shorter pin fins ( $H/d < 4$ ) tend to transfer less heat than longer ones. Sparrow et al. [13,14] and Metzger et al. [15] have also extensively studied the heat transfer characteristics of cylindrical pin fin in the inline and staggered configurations. A major conclusion from these studies is that the pin fin surface convective heat transfer coefficient was approximately 100% larger than that of the end wall. Different pin fin geometries have been investigated by Li et al. and Chen et al. [16,17]. It was determined that the heat transfer of drop shaped and ellipse shaped pin fins is increased with both types presenting a decrease in pressure loss when comparing to similar circular cross-section pin fins. Furthermore, Grannis and Sparrow [18] have developed correlations for the friction factor of fluid flow through diamond shaped pin fins. Square base pin fins have also received some attention by You and Chang [19], who used numerical simulations to study the performance of

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## Nomenclature

|                     |                                                  |             |                                                             |
|---------------------|--------------------------------------------------|-------------|-------------------------------------------------------------|
| $\Delta P_{outlet}$ | outlet differential pressure [Pa]                | $e_v$       | pumping power per unit volume [kW/m <sup>3</sup> ]          |
| $\Delta P_{fin}$    | fin differential pressure [Pa]                   | $FD$        | fin density [fin/in]                                        |
| $\Delta P_{inlet}$  | inlet differential pressure [Pa]                 | $H$         | fin height [m]                                              |
| $\Delta P_{tot}$    | total differential pressure [Pa]                 | $h$         | convective heat transfer coefficient [W/(m <sup>2</sup> K)] |
| $\Delta T_1$        | inlet temperature difference [K]                 | $k_f$       | fluid thermal conductivity [W/(m K)]                        |
| $\Delta T_2$        | outlet temperature difference [K]                | $\dot{m}$   | mass flow rate [kg/s]                                       |
| $\Delta T_{lm}$     | log mean temperature difference [K]              | $Nu_{Dh}$   | Nusselt number based on hydraulic diameter                  |
| $\eta_f$            | individual fin efficiency                        | $P_{flow}$  | flow perimeter [m]                                          |
| $\eta_o$            | overall fin efficiency                           | $q$         | heat flux [W/m <sup>2</sup> ]                               |
| $\mu$               | dynamic viscosity [Pa s]                         | $Re_{Dh}$   | Reynolds number based on hydraulic diameter                 |
| $\rho$              | fluid density [kg/m <sup>3</sup> ]               | $R_{eq}$    | equivalent thermal resistance [K/W]                         |
| $A_b$               | base area [m <sup>2</sup> ]                      | $T_b$       | base temperature [K]                                        |
| $A_f$               | fin heat transfer area [m <sup>2</sup> ]         | $T_{in}$    | inlet fluid temperature [K]                                 |
| $A_{flow}$          | net flow area [m <sup>2</sup> ]                  | $T_{out}$   | outlet fluid temperature [K]                                |
| $A_{tot}$           | total heat transfer area [m <sup>2</sup> ]       | $U_{max}$   | maximum fluid velocity [m/s]                                |
| $C_p$               | fluid specific heat capacity [kJ/(kg K)]         | $UA$        | thermal conductance [W/K]                                   |
| $D_{mean}$          | mean base fin length [m]                         | $UA_b$      | thermal conductance per unit area [kW/(m <sup>2</sup> K)]   |
| $D_h$               | hydraulic diameter [m]                           | $UA_v$      | thermal conductance per unit volume [kW/(m <sup>3</sup> K)] |
| $e$                 | pumping power [kW]                               | $V$         | volume [m <sup>3</sup> ]                                    |
| $e_b$               | pumping power per unit area [kW/m <sup>2</sup> ] | $\dot{V}_f$ | volumetric flow rate [m <sup>3</sup> /s]                    |

inline square base pin fins in a rectangular channel and have provided correlations for this type of geometry. Şara [20] has also studied several geometric parameters of staggered rectangular base pin fins such as fin spacing and shroud clearance and provided correlations for these parameters. Jeng and Tzeng [21] also experimentally confirmed the correlations found in the literature for square pin fins. It was found that the performance of some configurations of square base fins can exceed that of cylindrical pin fins under the same flow and geometric conditions.

Previous studies of pin fin geometries have focused on fins whose geometry is constant along the height of the fin. No literature was encountered for variable cross-section fins such as pyramidal shaped fins, until very recently. This lacuna in the literature is believed to be attributable to the lack of manufacturing methods for this type of fin. Traditional production methods (casting, machining, etching, etc.) cannot make economically viable fin arrays which are not constant along their height on an industrial scale. The manufacturability of square base pyramidal shape pin fin arrays produced using Cold Gas Dynamic Spraying (CGDS or simply cold spray) was demonstrated by Cormier et al. [22]. This type of manufacturing process opens new possibilities in terms of fin array geometries and configurations. This work [22] also presented information regarding the current limits of the production parameters. The fin density is limited to up to 24 fins per inch (for the specific aluminum powder used in this study), while the maximal attainable fin height is inversely proportional to the fin density. For 12 fins per inch samples, this limit is about 2.5 mm using the equipment described. Short pyramidal fins were produced and tested for thermal and hydrodynamic performance, which was detailed by Dupuis et al. [23]. It was demonstrated that flattening the top part of the pyramids could ensure better production consistency while retaining the fin performance for a specific fin height. It was also shown that the thermal performance is insensitive to the base angle of the pyramid at a given fin height. Many parameters still require investigation for the optimization of this new pin fin configuration.

In this work, the thermal and hydrodynamic performance of inline short aluminum square base pyramidal pin fin arrays produced with cold spray is assessed. The effect of varying the fin height is characterized, with the  $H/D_{mean}$  ratio varying between 1.24 and 2.62. The effect of increasing the fin density is also

characterized, with this parameter varying between 12 and 24 fins per linear inch. The convective heat transfer coefficient and thermal conductance parameters are evaluated using a forced flow convective heat transfer apparatus at low Reynolds numbers (less than 3500). This fixture is also used to determine the pressure loss through the fin array. Investigation of the fin height effect is limited to one configuration of uniformly distributed pin fins with a fin density of 12 fins per inch while investigation of the effect of the fin density is done for a fin height of 1.0 mm.

## 2. Fin production technique description

### 2.1. CGDS process and masking technique

The cold spray process was developed in the late 1980s at the Institute of Theoretical and Applied Mechanics of the Siberian Branch of the Russian Academy of Sciences [24,25]. This process is based on the addition of material to a substrate by the deposition of solid powder particles. This deposition is enabled by the acceleration of the powder particles by a high pressure carrier gas flowing at supersonic speed. Upon impact, the particles plastically deform and adhere to the substrate and to the already deposited particles, creating dense coatings [26,27]. More detailed discussions about the cold spray process can be found elsewhere [25].

The technique used to create square based pyramidal fin arrays using cold spray is shown in Fig. 1. It consists of a mask shielding parts of the substrate from the deposition of the particles accelerated by the cold spray system. Since this process is an additive manufacturing method, the selective masking of the substrate allows the user to build features of various shapes and dimensions. The operation principles of the spray process also encourage preferential build-up of material in the center of the mask openings, allowing the construction of features with reducing cross-sections along the height such as pyramids or cones [22]. Commercially available, plain woven, steel wire mesh (McMaster-Carr, Aurora, OH, USA) was used to mask the substrate from deposition of the cold spray system. This configuration creates a pyramidal fin array (Fig. 2) which closely mimics the reversed geometry of the chosen wire mesh, with peaks aligning with the holes of the screened area. Cold spray parameters were chosen to ensure that the deposited geometry is as close to the idealized geometry as possible. Addi-



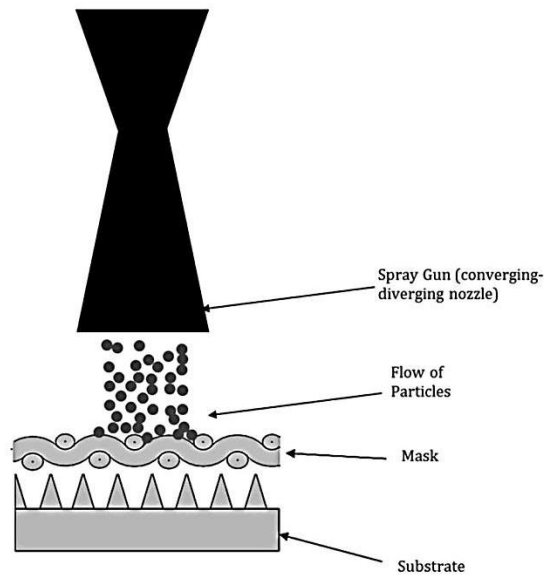


Fig. 1. Schematic of the spray process with masking technique.

tional details on the production of pyramidal pin fins can be found elsewhere [22,23].

The feedstock powder material used is commercially pure aluminum (99.8% Al, Centerline (Windsor) Ltd.) with an elongated shape and a mean particle diameter of 25.35 microns. Details about the powder morphology and size distribution are given elsewhere [22].

### 2.2. Test sample preparation

Fins were deposited on 5.1 cm by 5.1 cm aluminum alloy substrates (Al6061 T6). The substrates were cleaned and degreased using acetone after being cut to size. No further surface preparation steps were performed prior to spraying. The samples were sprayed slightly taller than the desired height, and were subsequently ground down to size to ensure production consistency, as detailed in Dupuis et al. [23].

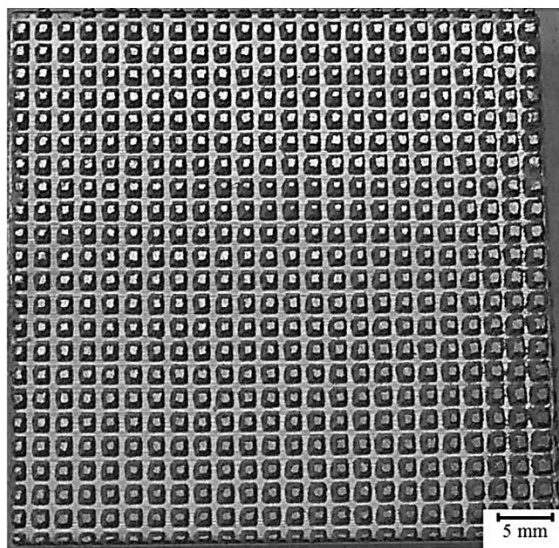


Fig. 2. Overview of a typical fin array.

## 3. Testing procedure

### 3.1. Performance test apparatus

Performance of the various fin arrays produced was assessed using an in-house heat transfer and pressure drop test apparatus, adapted from the literature [23,28]. A schematic of the apparatus is shown in Fig. 3.

The main components of this apparatus are the air supply, the test section, the heating pad and the data acquisition system. The air is supplied by a compressed air line equipped with a filter and dehumidifier. The air inlet flow rate is controlled by a needle valve and is measured with a flow meter. This air is smoothly expanded into a horizontal rectangular channel with a diffuser and a honeycomb flow straightener, with the shroud touching the top of the fins in the test section for every trial. An unobstructed 457 mm long section ensures that the flow is fully developed before reaching the test section. The test sample is subjected to a constant heat flux provided by a strip heater.

A stagnation pressure measurement hole is located 25.4 mm upstream from the test section, while a set of differential pressure measurement holes are located 12.7 mm upstream (high pressure side) and downstream (low pressure side) from the middle of the test section. Two thermocouples are located 6.0 mm ahead and afterward the test section at mid-height to record the fluid local average temperature. Four thermocouples are attached to the fin array's base, with two on the side facing the incoming flow, and two on the side facing the exiting flow. On each face, a thermocouple is located on the centerline, while the other thermocouple is located 6.0 mm away from the wall. All the thermocouples used are T-type, butt-bonded, thin gauge (28 gauge) thermocouples. Radiative heat transfer losses are neglected as they would account for less than 0.5% of the total heat input for similar test surfaces [29,30].

### 3.2. Heat transfer and pressure loss testing procedure

All measurements were performed at steady-state and were repeated until statistical significance was achieved. Heat transfer data samples were taken at 2 Hz over a 30 s interval. For both types of tests, five sets of data samples were recorded for each flow rate tested. Measurements were taken at flow rates of 20–70 SLPM, which corresponds to Reynolds numbers varying from 200 to 3500 depending on sample geometry. Deviation of the measurements from the mean value was less than 5%, which led to error bars that were smaller than the markers. Since the error bars were not visible, they were not included in the graphs of Section 4. A detailed description of the apparatus' accuracy and of the instrument's uncertainties is given in Cormier et al. [22]. For hydrodynamic performance tests, a set of pressure taps is used to obtain the fin array friction losses, which include skin friction and form drag.

### 3.3. Data analysis method

To analyze the raw data obtained, flow conditions are expressed using the Reynolds number based on the flow hydraulic diameter. Heat transfer performance evaluations are done by comparing convective heat transfer coefficients and thermal conductances. The thermal conductance shows the real thermal performance of the fin array while the convective coefficient serves to differentiate between thermal conductance gained from the increase in heat transfer area and the increase from fluid mechanics based phenomena.

The Reynolds number based on the hydraulic diameter is given by:



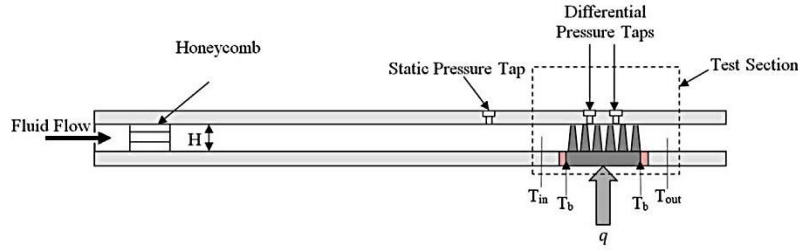


Fig. 3. Schematic of the heat transfer test fixture.

$$Re_{Dh} = \frac{\rho U_{max} D_h}{\mu} \quad (1)$$

where  $\rho$  is the fluid's density,  $U_{max}$  is the maximal fluid velocity in the fin array,  $D_h$  is the hydraulic diameter of the fin array flow passage and  $\mu$  is the fluid's dynamic viscosity. The variable  $D_h$  is given by:

$$D_h = \frac{4 \cdot A_{flow}}{P_{flow}} \quad (2)$$

where  $A_{flow}$  and  $P_{flow}$  are the flow channel's face area and perimeter, respectively.

The thermal conductance can be calculated using:

$$UA = \frac{1}{R_{eq}} = h \cdot A_{tot} \cdot \eta_0 \quad (3)$$

where  $R_{eq}$  is the thermal circuit's equivalent resistance,  $h$  is the convective heat transfer coefficient,  $\eta_0$  is the overall surface efficiency and  $A_{tot}$  is the total heat transfer area of the exposed fin and base surface. The overall surface efficiency is calculated using the following equation:

$$\eta_0 = 1 - \frac{A_f}{A_{tot}} (1 - \eta_f) \quad (4)$$

where  $A_f$  is the exposed fin area and  $\eta_f$  is the individual fin efficiency, defined as the heat transferred by the fin over the maximum possible heat that can be transferred, which is adapted from the procedure detailed in Incropera's work [5]. More details can be found in [22,23], but for the samples described in this work the overall surface efficiency is computed to be over 0.99.

The fin array's convective heat transfer coefficient and Nusselt number are calculated using the following relations:

$$h = \frac{q}{\Delta T_{lm} A_{tot} \eta_0} \quad (5)$$

$$Nu_{Dh} = \frac{h \cdot D_h}{k_f} \quad (6)$$

In these equations,  $k_f$  is the thermal conductivity of the fluid,  $q$  is the heating rate and  $\Delta T_{lm}$  is the log-mean temperature difference between the fluid and the fin array. The log-mean temperature difference is calculated with:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (7)$$

where  $\Delta T_1$  and  $\Delta T_2$  are the temperature difference between the fin surface and the flow at the inlet and outlet, respectively. The total heat input rate to the system is calculated using:

$$q = \dot{m} \cdot (T_{out} - T_{in}) \cdot C_p \quad (8)$$

where  $\dot{m}$  is the fluid mass flow rate,  $(T_{out} - T_{in})$  is the fluid temperature difference between the entrance and the exit of the test section,  $C_p$  is the fluid's thermal capacitance.

To determine which fin array transfers heat more efficiently, an efficiency indicator must be selected. Based on the literature review performed, the method proposed in Sahiti's work [8] was selected. This method is based on plotting the thermal and hydrodynamic data with a curve of thermal conductance as a function of the required pumping power for different flow conditions. This yields curves which can be directly compared one to another, with the most efficient fin array having a higher thermal conductance at a given pumping power. The required pumping power input was calculated according to Sahiti's definition [9]:

$$e = \frac{\dot{V}_f \cdot \Delta P_{fin}}{\eta} \quad (9)$$

In the previous relation,  $e$  is the pumping power,  $\dot{V}_f$  is the volumetric flow rate of the fluid while  $\eta$  is the fan efficiency. A fan efficiency of 0.8 was chosen as a reasonable value to perform these calculations [9]. The pumping power is divided either by the fin array base ( $A_b$ ) to yield the pumping power per unit area or by the volume ( $V$ ) of the fin array to obtain the pumping power per unit volume.

### 3.4. Test sample geometric data

The fin array geometric variables were measured using a depth of field microscope (Keyence model VHX-2000, Mississauga, Ontario, Canada) which allows building three dimensional images of the given samples. The 12 fins per inch pyramidal fin samples were used to investigate the fin height effect while the samples used to investigate the fin density effect have a height of 1.0 mm. Table 1 shows the important geometric variables required for the analysis of the data obtained from the pin fin samples. In this table, the prefix  $H$  indicates a sample used for the height effect study, with the following number representing the height of the sample. Similarly, the prefix  $FD$  represents a sample used for the fin density study, with the number following this prefix designating the amount of fins per linear inch of this sample.

## 4. Results

### 4.1. Heat transfer performance

Fig. 4 indicates that the pyramidal pin fin array's thermal conductance increases with the Reynolds number for all fin height tested. It also shows that the thermal conductance of the fin array increases with fin height. This is expected as Brigham and Van Fossen [11,12] demonstrated the same trend for circular pins fins. The convective heat transfer coefficient values are very similar between the 1.2, 1.4, 1.6, 1.8 and 2.0 mm height samples, while the 1.0 and 2.2 mm samples seem to differ from the rest of the group, as it can be seen in Fig. 5. This nearly constant convective heat transfer coefficient indicates that the increase of the total available heat transfer area governs the increase in thermal conductance observed in Fig. 4 for these cases. It is hypothesized

**Table 1**  
Important geometric variables of pyramidal samples.

| Sample | Hydraulic diameter mm | Total heat transfer area mm <sup>2</sup> | $D_{mean}$ mm | $H/D_{mean}$ ratio | Height mm | Fin density fpi |
|--------|-----------------------|------------------------------------------|---------------|--------------------|-----------|-----------------|
| H1.0   | 1.02                  | 3637                                     | 0.81          | 1.24               | 1.0       | 12              |
| H1.2   | 1.19                  | 3834                                     | 0.77          | 1.56               | 1.2       | 12              |
| H1.4   | 1.30                  | 4128                                     | 0.76          | 1.83               | 1.4       | 12              |
| H1.6   | 1.24                  | 5184                                     | 1.05          | 1.53               | 1.6       | 12              |
| H1.8   | 1.35                  | 5389                                     | 0.98          | 1.84               | 1.8       | 12              |
| H2.0   | 1.45                  | 5525                                     | 0.91          | 2.20               | 2.0       | 12              |
| H2.2   | 1.56                  | 5592                                     | 0.84          | 2.62               | 2.2       | 12              |
| FD12   | 1.02                  | 3637                                     | 0.81          | 1.24               | 1.0       | 12              |
| FD16   | 0.90                  | 4570                                     | 0.72          | 1.38               | 1.0       | 16              |
| FD20   | 0.73                  | 5524                                     | 0.66          | 1.51               | 1.0       | 20              |
| FD24   | 0.66                  | 6346                                     | 0.55          | 1.81               | 1.0       | 24              |

that the samples between the 1.2 and 2.0 mm heights share a similar flow structure, while the 1.0 and 2.2 samples potentially have a slightly different flow structure that would lead to the observed difference in convective heat transfer coefficients.

Similar trends can be seen when the fin density is varied, as illustrated in Figs. 6 and 7. The thermal conductance follows the expected trend, with samples having an increasing thermal conductance with fin density. One can observe a large increase of thermal conductance between samples FD12 and FD16, with smaller increases between FD16, FD20, and FD24. The convective coefficient of these samples is shown in Fig. 7, where all samples except FD12 have the same curve of convective coefficient as a function of Reynolds. These observations suggest that there is a change in flow structure as fin density is increased from 12 to 16 but that this flow structure is similar for the 16, 20 and 24 fin density cases. Further investigations will be performed using advanced localized flow visualization techniques to validate this hypothesis and shall be the topic of a dedicated study.

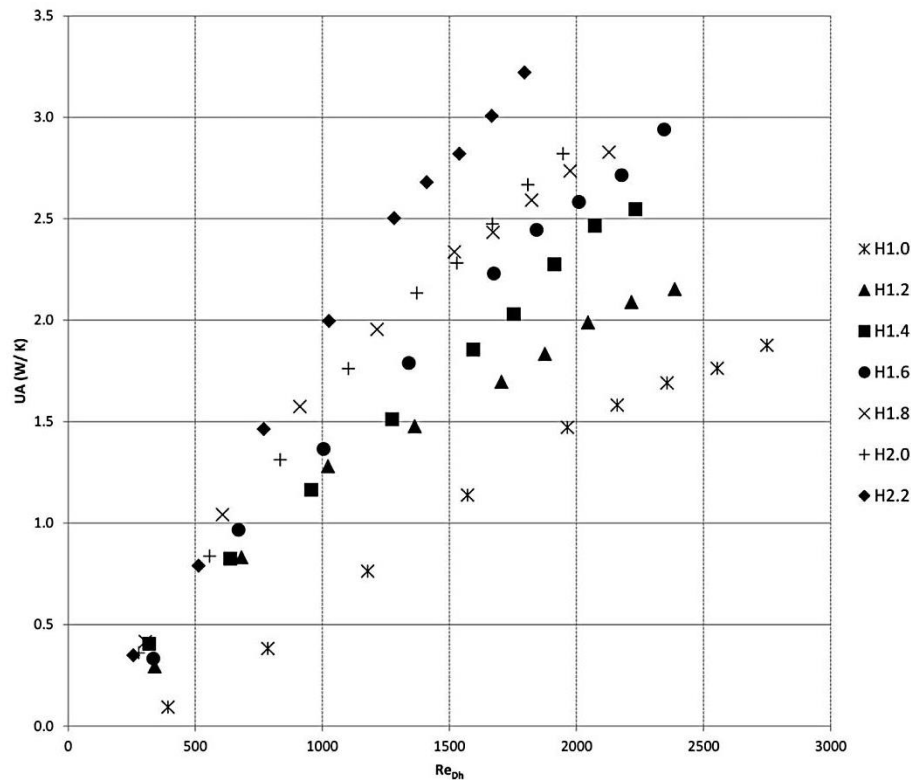
## 4.2. Hydrodynamic performance

Fig. 8 shows that the pressure losses through the fin arrays seem to be independent of the fin height over the range of Reynolds number studied. This fact, combined with the observations made in Section 4.1 regarding the convective coefficient, leaves one to believe that the flow structure is not affected much by varying the fin height between 1.2 and 2.0 mm. The pressure loss has a quadratic relation with Reynolds number for all samples, which is expected due to the fact that this dimensionless number varies linearly with flow velocity while pressure losses vary with the square of fluid velocity. The 1.0 and 2.2 mm height samples also have similar pressure loss compared to the other samples. This indicates that further investigations of the flow structure are required to understand the differences in heat transfer performance of these geometries.

The pressure losses as a function of the Reynolds number for the fin density samples are shown in Fig. 9. An increase in pressure loss is observed as fin density increases. This phenomenon is expected, as pressure losses in pin fin array increase with the number of fin rows in the flow direction. Indeed, the number of flow recirculation zones is increased as the fin density is increased, which creates more friction losses.

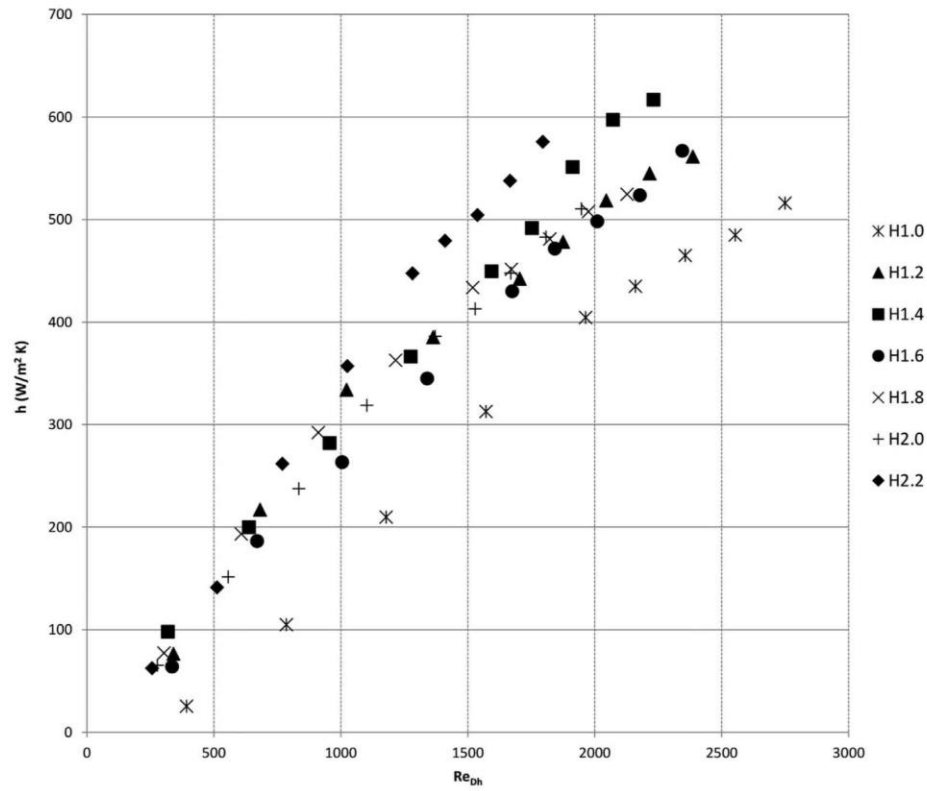
## 4.3. Heat transfer efficiency

Observing Fig. 10, one notices that for the range of fin heights studied there is very little variation in heat transfer efficiency per unit volume between the different sample heights. The efficiency curves all have a similar shape, with a sharp initial increase in thermal conductance with pumping power, with inflection points located near 250 kW/m<sup>3</sup>. After this inflection point, a tapering off of the slope is noted. For some applications, however, the important

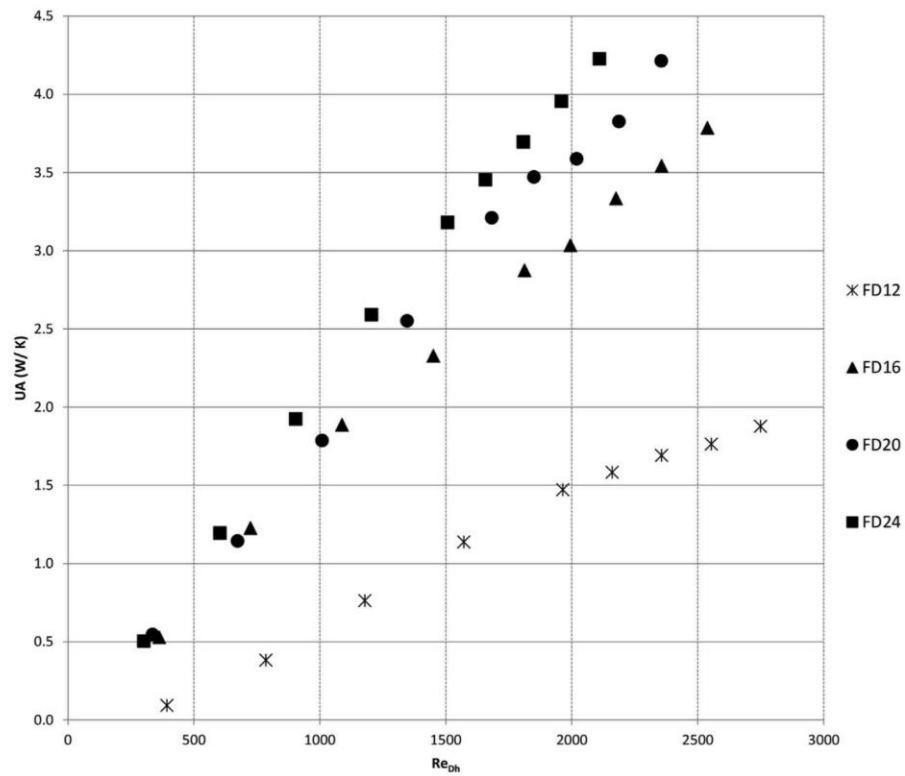


**Fig. 4.** Thermal conductance comparison between various fin heights: 1.0 mm (star), 1.2 mm (triangle), 1.4 mm (square), 1.6 mm (circle), 1.8 mm (X), 2.0 mm (cross) and 2.2 mm (diamond).





**Fig. 5.** Convective heat transfer coefficient comparison between various fin heights: 1.0 mm (star), 1.2 mm (triangle), 1.4 mm (square), 1.6 mm (circle), 1.8 mm (X), 2.0 mm (cross) and 2.2 mm (diamond).



**Fig. 6.** Thermal conductance comparison between various fin densities: 12 fins per inch (star), 16 fins per inch (triangle), 20 fin per inch (circle) and 24 fins per inch (square).

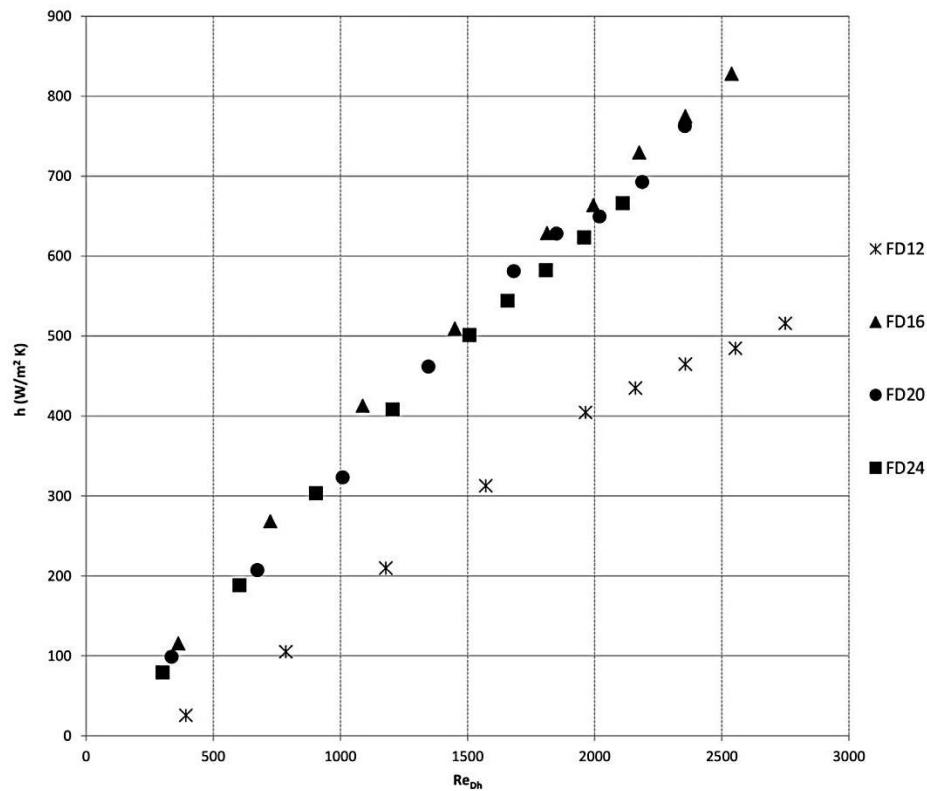


Fig. 7. Convective heat transfer coefficient comparison between various fin densities: 12 fins per inch (star), 16 fins per inch (triangle), 20 fin per inch (circle) and 24 fins per inch (square).

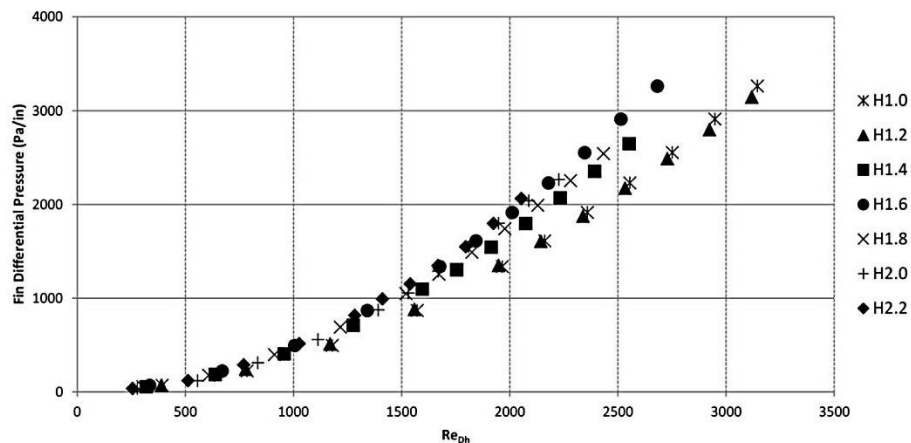


Fig. 8. Fin differential pressure comparison between various fin heights: 1.0 mm (star), 1.2 mm (triangle), 1.4 mm (square), 1.6 mm (circle), 1.8 mm (X), 2.0 mm (cross) and 2.2 mm (diamond).

parameter is not the total volume of the heat exchanger, but the footprint area of such devices. In this case, Fig. 11 reveals that using longer fins yields a much better performance than shorter ones for a given footprint area.

Fig. 12 shows the volume based surface efficiency for the different fin density samples. Once again, the curves have a sharp increase in thermal performance at low pumping powers, followed by an inflection point and a zone where there is a large increase of the required pumping power for every unit of thermal conductance. This figure also demonstrates that the increases in thermal conductance are offset by the increased pressure losses for the FD16, FD20 and FD24 samples. This would imply that there is little

advantage to using fin arrays with more than 16 fins per inch due to the fact that these arrays are as efficient as the 20 and 24 fins per inch samples tested, while being easier to manufacture using the cold spray process. The same conclusions are reached when using the base area surface efficiency, as the curve is identical to the volume based efficiency.

#### 4.4. Performance correlations

To be able to predict the heat transfer performance of fin arrays at various fin heights and fin densities and use the data generated in a design/predictive environment, general correlations should

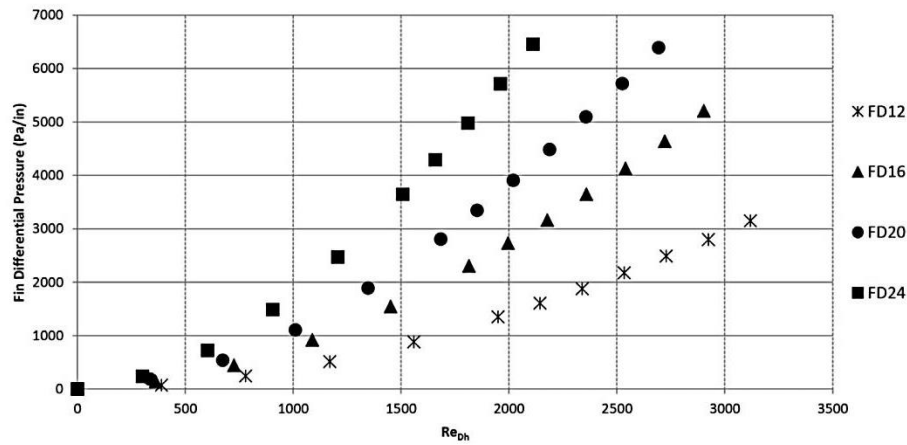


Fig. 9. Fin differential pressure comparison between various fin densities: 12 fins per inch (star), 16 fins per inch (triangle), 20 fin per inch (circle) and 24 fins per inch (square).

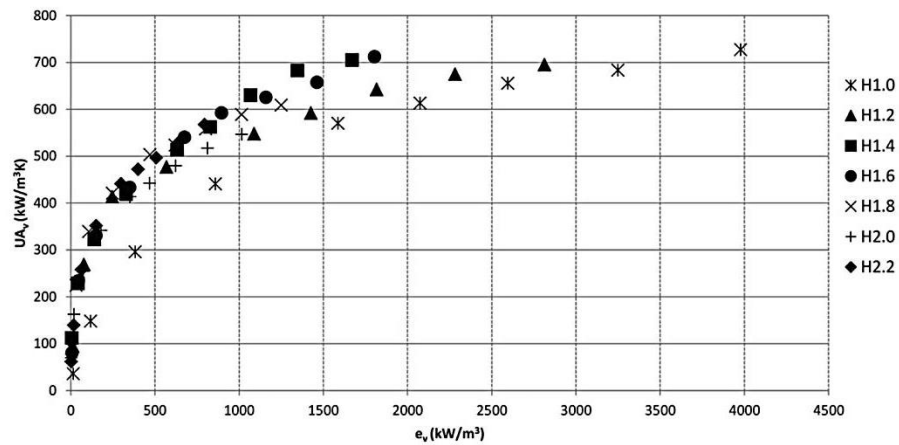


Fig. 10. Volume based heat transfer surface efficiency comparison between various fin heights: 1.0 mm (star), 1.2 mm (triangle), 1.4 mm (square), 1.6 mm (circle), 1.8 mm (X), 2.0 mm (cross) and 2.2 mm (diamond).

ideally be developed for the Nusselt number of the fin arrays. This would allow one to perform predictive modeling and to compare different heat exchanger geometries without having to perform the corresponding experimental work. In this case, the Nusselt number acts as a dimensionless heat transfer performance scaling parameter. To this end, the height samples between 1.2 and

2.0 mm height were selected to build correlations because they are believed to have similar flow structures. The fin density samples with the most similar flow structures (FD16, FD20 and FD24) were also selected to build another set of correlations. Due to the small number of samples tested, no attempt was made to produce cross-correlations between the height effect and the fin

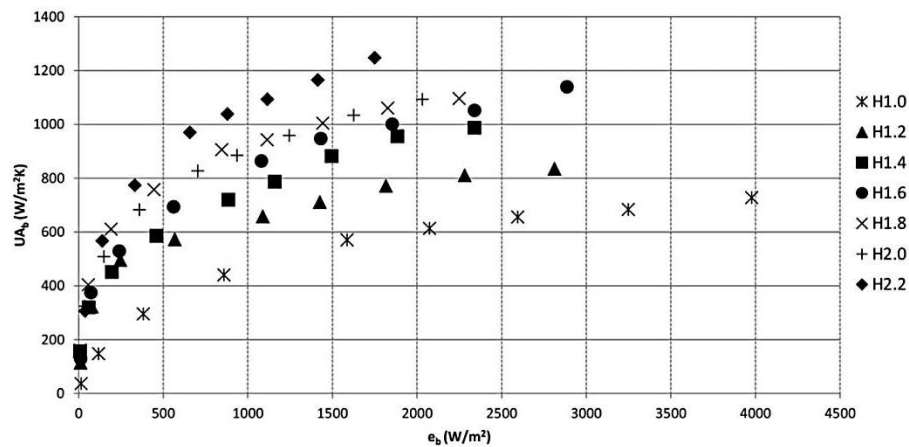


Fig. 11. Footprint area based heat transfer surface efficiency comparison between various fin heights: 1.0 mm (star), 1.2 mm (triangle), 1.4 mm (square), 1.6 mm (circle), 1.8 mm (X), 2.0 mm (cross) and 2.2 mm (diamond).



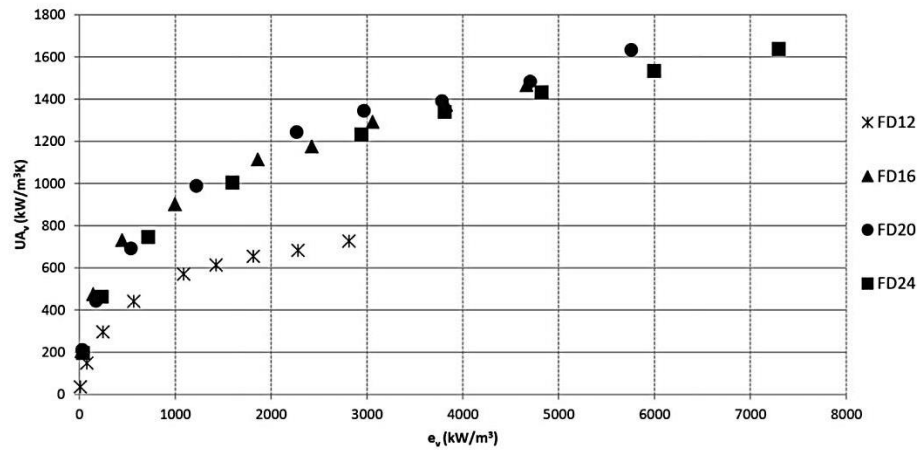


Fig. 12. Volume based heat transfer surface efficiency comparison between various fin densities: 12 fins per inch (star), 16 fins per inch (triangle), 20 fin per inch (circle) and 24 fins per inch (square).

density effect. Two linearly proportional relationships (regimes) between the Reynolds number and the Nusselt number at a given fin height have been identified. The trends agree with the theory proposed by Žukauskas [7] for banks of tubes where two regimes dictate the flow structure at low Reynolds numbers. The first regime (laminar regime) would extend from very low Reynolds to approximately  $1000 Re_{Dh}$ , where the flow is dominantly laminar with large scale recirculation regions behind the fins. The second regime (sub-critical regime) would start near  $500 Re_{Dh}$  and extend to  $200,000 Re_{Dh}$ , with the flow structure being largely laminar but becoming increasingly turbulent in the fin's wake. For  $Re_{Dh}$  between 500 and 1000, the flow is in transition between the two regimes. In this case, for the pyramidal fins, the transition between the two slopes is found in this range of Reynolds number, which confirms the hypothesis that the flow behavior is similar between banks of tubes and pin fins.

The procedure for obtaining a correlation for the Nusselt number as a function of the fin height is detailed as follows. Two curves of the Nusselt number as a function of the Reynolds number were plotted using the numerical average of the five samples in question. Each curve has a domain of application, with the first located in the laminar regime, while the second lies in the sub-critical regime. The separation of the domain into two curves is warranted due to the change in flow structure between these two regimes. Linear regressions were found which correlate the results appropriately ( $R^2$  value of 0.9997 and 0.9977, respectively). The average Nusselt value lines was then reconverted to thermal conductance values with the geometric data to evaluate the error between the correlation and the actual experimental data. The error averages are less than 15% for all samples which validates the proposed approach. This would indicate that using the equations:

$$Nu_{Dh} = 0.0173 Re_{Dh} - 1.5099 \quad \text{for } 350 < Re_{Dh} < 750 \quad (10)$$

$$Nu_{Dh} = 0.0120 Re_{Dh} + 2.8463 \quad \text{for } 750 < Re_{Dh} < 2000 \quad (11)$$

would appropriately model the thermal performance of 12 fin per inch pyramidal fin arrays with heights varying between 1.2 and 2.0 mm over the range of Reynolds number studied.

A different approach was used to create correlations between the Nusselt number, Reynolds number and the fin density. The Nusselt number was found to be a quadratic function of the fin density, with a  $R^2$  value of 1 for samples FD16, FD20 and FD24. Correlations were built using a first order term for Reynolds number, with first and second order terms for the fin density. Once again, the domain was separated in two zones corresponding to the

two flow regimes identified: laminar and sub-critical. The correlation curves and the experimental curves are found to have good agreement. The errors were also evaluated using values that were converted back to thermal conductance. The error averages are less than 5% for these samples. The following correlations are then proposed for fin densities between 16 and 24 fins per inch:

$$Nu_{Dh} = 1.81 \cdot 10^{-4} Re_{Dh} FD^2 - 0.0486 FD^2 - 0.00793 Re_{Dh} FD + 2.02 FD + 0.0954 Re_{Dh} - 21.2 \quad \text{for } 350 < Re_{Dh} < 750 \quad (12)$$

$$Nu_{Dh} = 2.61 \cdot 10^{-5} Re_{Dh} FD^2 + 0.0171 FD^2 - 0.00133 Re_{Dh} FD - 1.05 FD + 0.0238 Re_{Dh} + 17.0 \quad \text{for } 750 < Re_{Dh} < 2000 \quad (13)$$

to appropriately model the thermal performance of 1.0 mm height pyramidal fin arrays over the range of Reynolds number studied.

## 5. Summary and conclusion

This study investigates the thermal and hydrodynamic performance of several heights of pyramidal fin arrays. This investigation concentrated on samples with fin densities of 12 fins per inch between 1.0 and 2.2 mm heights and samples of 1.0 mm height with fin densities varying between 12 and 24 fins per inch. The performances of these fins are compared using the thermal conductivity over the required pumping power per unit volume to assess the efficiency of these fin arrays for a given heat sink geometry. The conclusions of this work are summarized as:

1. The thermal conductance of the fin arrays tested increases with height and with fin density at a given Reynolds number, which is consistent with other work produced in the literature for other fin geometries. The convective coefficient is very similar over the range of sample heights studied, which leads to the conclusion that the parameter governing the increase in thermal conductivity with fin height is purely geometrical. The increased height allows for more available heat transfer area, which subsequently increases UA. A similar explanation could also be applied to the fin density samples, as the 16, 20 and 24 fin per inch sample have the same curve of convective coefficient with Reynolds number, which would indicate that the growth of the thermal conductance is due to the increased heat transfer area for these samples.



2. The hydrodynamic performance of the fin arrays studied follows the expected distribution, with the pressure losses varying very slightly with fin height. This supports the theory that the flow structure is largely similar, especially for fins between the heights of 1.2–2.0 mm. For the fin density samples, a linear increase in pressure loss is found as the fin density is increased. This is also expected as the pressure loss of pin fins is directly proportional to the number of rows in the fin array.
3. The global performance of the height study pyramidal fin arrays is largely similar over the entire range of flow conditions studied when the results are compared for a unit volume. If the fin height is not an obstacle to the design, increasing this parameter yields better thermal performances for a given pumping power per unit area, however. The efficiencies of samples with fin densities between 16 and 24 fins per inch are shown to be very similar over the range of Reynolds number studied, which implies that manufacturing samples with 16 fins per inch is preferable due to the limits of the production technique.
4. The flow regimes are believed to be different between the 1.0, 2.2 mm and the other samples for the height effect study due to the aforementioned observations. Furthermore, the results obtained from the fin density study suggest that there is also a different flow structure between the samples produced at 12 fins per inch and the samples with higher fin densities. These hypotheses will be detailed in a later communication focusing on the observation of the different flow structures at different fin heights, fin densities and Reynolds numbers using the Particle Image Velocimetry (PIV) flow observation technique.
5. Correlations were proposed to model the heat transfer performance of pyramidal fins with geometries such as those studied in this work. Relations linking the Nusselt number, Reynolds number, fin height and the fin densities are found in Section 4.4. The maximal error obtained using the procedure detailed in this section is 20%, with average errors at less than 15%, which validates the proposed approach.

#### Conflict of interest

None declared.

#### Acknowledgments

Acknowledgments are due to the MITACS Accelerate program for its financial support of this project.

#### References

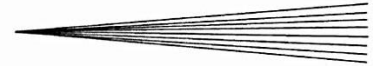
- [1] Cornell Aeronautics Laboratory, Heat Transfer in Miniaturized Electronic Equipment, Bureau of ships, Navy Department, US Army, 1955.
- [2] S.G. Kandlikar, W.J. Grande, Evaluation of single phase flow in microchannels for high flux chip cooling – thermohydraulic performance enhancement and fabrication technology, *Heat Transfer Eng.* 25 (8) (2004) 5–16.
- [3] R.L. Webb, Enhancement of single-phase heat transfer, in: S. Kakac, K. Ramesh, W. Aung (Eds.), *Handbook of Single-Phase Convective Heat Exchanger Design*, Wiley, New York, 1987, pp. 17–1–17–8.
- [4] J.C. Han, Turbine blade cooling studies at Texas A&M university: 1980–2004, *J. Thermophys. Heat Transfer* 20 (2) (2006).
- [5] F.P. Incropera, D.P. DeWitt, T.L. Bergman, A.S. Lavine, *Fundamentals of Heat and Mass Transfer*, 6th ed., Wiley, New York, 2006.
- [6] W.M. Kays, A.L. London, *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York, 1984.
- [7] A. Žukauskas, Heat transfer from tubes in crossflow, *Adv. Heat Transfer* 8 (1972) 93–160.
- [8] N. Sahiti, F. Durst, A. Dewan, Heat transfer enhancement by pin elements, *Int. J. Heat Mass Transfer* 48 (2005) 4738–4747.
- [9] N. Sahiti, F. Durst, A. Dewan, Strategy for selection of elements for heat transfer enhancement, *Int. J. Heat Mass Transfer* 49 (2006) 3392–3400.
- [10] N. Sahiti, A. Lemouedda, D. Stojkovic, F. Durst, E. Franz, Performance comparison of pin fin in-duct flow arrays with various pin cross-sections, *Appl. Therm. Eng.* 26 (2006) 1176–1192.
- [11] B.A. Brigham, G.J. Vanfossen, Length to diameter ratio and row number effects in short pin fin heat transfer, *ASME J. Eng. Gas Turbines Power* 104 (1984) 241–244.
- [12] G.J. Vanfossen, Heat-transfer coefficients for staggered arrays of short pin fins, *ASME J. Eng. Gas Turbines Power* 102 (1982) 268–274.
- [13] E.M. Sparrow, J.W. Ramsey, Heat transfer and pressure drop for a staggered wall-attached array of cylinders with tip clearance, *Int. J. Heat Mass Transfer* 21 (1978) 1369–1377.
- [14] E.M. Sparrow, J.W. Ramsey, C.A.C. Altamiani, Experiments on in-line pin fin arrays and performance comparison with staggered arrays, *ASME J. Heat Transfer* 102 (1980) 44–50.
- [15] D.E. Metzger, C.S. Fan, S.W. Haley, Effects of pin shape and array orientation on heat transfer and pressure loss in pin fin arrays, *ASME J. Eng. Gas Turbines Power* 106 (1984) 252–257.
- [16] Q. Li, Z. Chen, U. Flechtner, H.-J. Warnecke, Heat transfer and pressure drop characteristics in rectangular channels with elliptic pin fins, *Int. J. Heat Fluid Flow* 19 (1998) 245–250.
- [17] Z. Chen, Q. Li, D. Meier, H.-J. Warnecke, Convective heat transfer and pressure loss in rectangular ducts with drop-shaped pin fins, *Heat Mass Transfer* 33 (1997) 219–224.
- [18] V.B. Grannis, E.M. Sparrow, Numerical simulation of fluid flow through an array of diamond-shaped pin fins, *Numer. Heat Transfer (Part A)* 19 (1991) 381–403.
- [19] H.I. You, C.H. Chang, Numerical prediction of heat transfer coefficient for a pin-fin channel flow, *ASME J. Heat Transfer* 119 (1997) 840–843.
- [20] O.N. Şara, Performance analysis of rectangular ducts with staggered square pin fins, *Energy Convers. Manage.* 44 (2003) 1787–1803.
- [21] T.M. Jeng, S.C. Tzeng, Pressure drop and heat transfer of square pin-fin arrays in in-line and staggered arrangements, *Int. J. Heat Mass Transfer* 50 (2007) 2364–2375.
- [22] Y. Cormier, B. Dupuis, B. Jodoin, A. Corbeil, Net shape fins for compact heat exchanger produced by cold spray, *J. Therm. Spray Technol.* 22 (7) (2013) 1210–1221.
- [23] P. Dupuis, Y. Cormier, A. Farjam, B. Jodoin, A. Corbeil, Performance evaluation of near-net pyramidal shaped fin arrays manufactured by cold spray, *Int. J. Heat Mass Transfer* 69 (2014) 34–43.
- [24] A.P. Alkhimov, A.N. Papyrin, V.F. Dosarev, N.I. Nesterovich, M.M. Shuspanov, US Patent No. 5302414, 1994.
- [25] E. Irissou, J.-G. Legoux, A.N. Ryabinin, B. Jodoin, C. Moreau, Review on cold spray process and technology: part I-intellectual property, *J. Therm. Spray Technol.* 17 (4) (2008) 495–516.
- [26] J.R. Davis (Ed.), *Handbook of Thermal Spray Technology*, ASM International, 2004.
- [27] H. Herman, S. Sampath, R. McCune, *Thermal Spray: Current Status and Future Trends*, Materials Research Society, Bulletin, 2000, pp. 17–25. vol. 25, issue 7.
- [28] J. Tian, T. Kim, T.J. Lu, H.P. Hodson, D.T. Queheillalt, D.J. Syceck, H.N.G. Wadley, The effects of topology upon fluid-flow and heat-transfer within cellular copper structures, *Int. J. Heat Mass Transfer* 47 (14–16) (2004) 3171–3186.
- [29] S. Naik, S.D. Probert, M.J. Shilton, Forced convective steady-state heat transfers from shrouded vertical fin arrays aligned parallel to an undisturbed air-stream, *Appl. Energy* 26 (1987) 137–158.
- [30] J.J. Hwang, T.M. Liou, Heat transfer and friction in a low-aspect-ratio rectangular channel with staggered perforated ribs on two opposite walls, *ASME J. Heat Transfer* 117 (11) (1995) 843–850.



## **5.4 Paper #4: Shear Strength and Bond Coat**

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The paper presented in this section details the shear strength of the pyramidal fins. The effects of the fin density, fin height and substrate surface roughness on the fin's shear strength were evaluated. Also, the feasibility of manufacturing these fins on various bond coats was investigated. The effect of adding a bond coat was evaluated in terms of thermal and hydrodynamic performance and shear strength. This paper addresses mainly the objective 3.5 which is the adhesion strength of the near-net pyramidal shaped fin arrays manufactured by CGDS.



# Mechanical Properties of Cold Gas Dynamic-Sprayed Near-Net-Shaped Fin Arrays

Yannick Cormier, Philippe Dupuis, Bertrand Jodoin, and Antoine Corbeil

(Submitted September 4, 2014; in revised form November 7, 2014)

This work focuses on the study of the adhesion and thermal performance of near-net-shaped pyramidal fin arrays manufactured by cold spray on aluminum alloy substrate coated with various bond coats: a cold-sprayed bond coat as well as nitrogen- and air-propelled arc-sprayed bond coats. Furthermore, the effects of the fin density, fin height, and substrate surface roughness on the adhesion strength of the fins deposited on Al6061 substrates were characterized. It was found that the fin density, the fin height, and the substrate roughness have little impact on the adhesion strength of this system. The adhesion strength was found to be inversely proportional to the surface hardness when investigating these parameters for the different thermal-spray bond coatings, with all the fin systems having a much greater strength than the theoretical application stresses. Finally, it was found that the increase in the fin's base layer's roughness increases the overall heat transfer, with the bond coat material having a negligible effect on the thermal resistance for this type of heat-exchanger configuration.

**Keywords** additive manufacturing, cold spray, heat transfer, pin fin array, shear strength

## 1. Introduction

Localized power production systems have been the subject of a substantial number of researches over the last two decades (Ref 1, 2). Low power production capacity systems (less than 10 MW) typically rely on cycles such as reciprocating gasoline or diesel engines to convert the fuel's chemical energy to mechanical work (Ref 3). These systems are cumbersome and complex due to their size and the number of mechanical parts required for the system to operate. Research has been undertaken to replace such systems by gas turbine generator systems, due to the combined advantages of low emissions and high reliability found in a compact gas turbine package (Ref 3).

The simplest gas turbine generator designs often have efficiencies below 20% due to the large amount of heat that remains unused in the exhaust flow, while diesel and gasoline engines typically operate at thermal efficiencies between 30 and 35% over the same power production range (Ref 3). However, gas turbine generators equipped with a recuperator preheating the inlet gas using the heat recovered from the hot exhaust gases have the potential to reach thermal efficiency of >50% (Ref 3). Consequently, recuperators play a crucial role in a successful design of gas turbine generator, and often represent 20–25% of the cost and 20–35% of the volume of the turbine-recuperator

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## Nomenclature

|                  |                                                               |
|------------------|---------------------------------------------------------------|
| $\Delta P_{fin}$ | Fin differential pressure [Pa]                                |
| $\Delta T_1$     | Inlet temperature difference [K]                              |
| $\Delta T_2$     | Outlet temperature difference [K]                             |
| $\Delta T_{lm}$  | Log mean temperature difference [K]                           |
| $\varepsilon$    | Relative surface roughness                                    |
| $\eta$           | Fan efficiency                                                |
| $\rho$           | Fluid density [kg/m <sup>3</sup> ]                            |
| $\tau_d$         | Fluid drag shear stress [Pa]                                  |
| $A_{base}$       | Base area [m <sup>2</sup> ]                                   |
| $A_{cond}$       | Conduction area [m <sup>2</sup> ]                             |
| $A_{conv}$       | Convection area [m <sup>2</sup> ]                             |
| $A_{front}$      | Frontal area [m <sup>2</sup> ]                                |
| $C_d$            | Drag coefficient                                              |
| $C_p$            | Fluid's specific heat capacity [kJ/(kg K)]                    |
| $D_h$            | Hydraulic diameter [m]                                        |
| $e_v$            | Pumping power per unit volume [kW/m <sup>3</sup> ]            |
| $H$              | Fin height [m]                                                |
| $h$              | Convective heat-transfer coefficient [W/(m <sup>2</sup> · K)] |
| $k$              | Thermal conductivity [W/(m · K)]                              |
| $L$              | Base length (streamwise direction) [m]                        |
| $\dot{m}$        | Mass-flow rate [kg/s]                                         |
| $q$              | Heat flux [W/m <sup>2</sup> ]                                 |
| $R_{cond}$       | Conduction thermal resistance [K/W]                           |
| $R_{conv}$       | Convection thermal resistance [K/W]                           |
| $R_{eq}$         | Equivalent thermal resistance [K/W]                           |
| $r$              | Surface roughness [m]                                         |
| $T_{in}$         | Inlet fluid temperature [K]                                   |
| $T_{out}$        | Outlet fluid temperature [K]                                  |
| $t$              | Conduction length [m]                                         |
| $U$              | Fluid velocity [m/s]                                          |
| $UA_{conv}$      | Convection thermal conductance [W/K]                          |
| $UA_v$           | Thermal conductance per unit volume [kW/(m <sup>3</sup> K)]   |

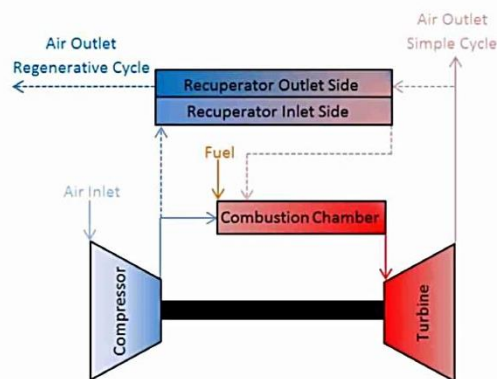


|             |                                                |
|-------------|------------------------------------------------|
| $V$         | Volume [ $\text{m}^3$ ]                        |
| $\dot{V}_f$ | Volumetric flow rate [ $\text{m}^3/\text{s}$ ] |

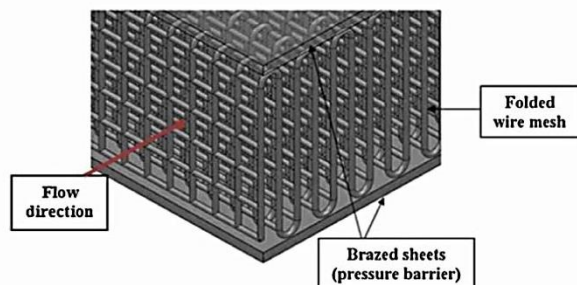
package (Ref 4). Furthermore, increasing recuperator efficiencies from 5 to 10% can improve the overall thermal efficiency of the power generation system by the same amount (Ref 3). To this end, decreasing the size and increasing the efficiency of such components become important. Figure 1 shows the difference between the simple cycle and the regenerative gas turbine cycle.

State-of-the-art recuperators operate at thermal efficiencies from 75 to 85%. These compact heat exchangers typically use packed bed or metallic foam as heat-transfer media; however, they are prone to problems such as leakage, high-pressure and axial conduction losses, and high manufacturing costs (Ref 4, 5).

A new type of compact heat exchangers, referred to as a Wire Mesh Heat Exchanger (WMHE), was developed in the 1980s as a solution for the issues commonly faced by other types of heat-transfer-enhancing media (Ref 6, 7). WMHEs are based on the concept of using metallic wire mesh sheets as extended surfaces inside a cellular metallic sandwich structure. The initial configuration developed using this concept was the folded wire mesh geometry—shown in Fig. 2. This geometry is produced by successively folding wire mesh textiles and brazing metallic



**Fig. 1** Schematic of a gas turbine cycle (full line) and regenerative cycle (dashed line)



**Fig. 2** Wire mesh heat exchanger using folded wire mesh as heat-transfer medium

sheets to the tips of the folds to create the fully enclosed heat-exchanger cell. The small amount of contact area between the wire mesh and the brazed sheets, combined with a limit in the number of folds of the wire mesh achievable per unit area, are important restrictions inherent in this production method (Ref 8). These limitations call for an improved production method that can increase the contact area between the heat-transfer medium and the brazed parting plate.

A new method for producing WMHEs has recently been developed which addresses the above concern. Rather than being folded, the wire mesh textile is stacked and cut, and then sealed with a thin layer of arc-sprayed coating (bond coat) followed by a much thicker layer produced by means of Cold Gas Dynamic Spraying (CGDS) process. Plate fins are then machined into this cold-sprayed layer to create a heat-transfer-enhancing surface. These steps are illustrated in Fig. 3.

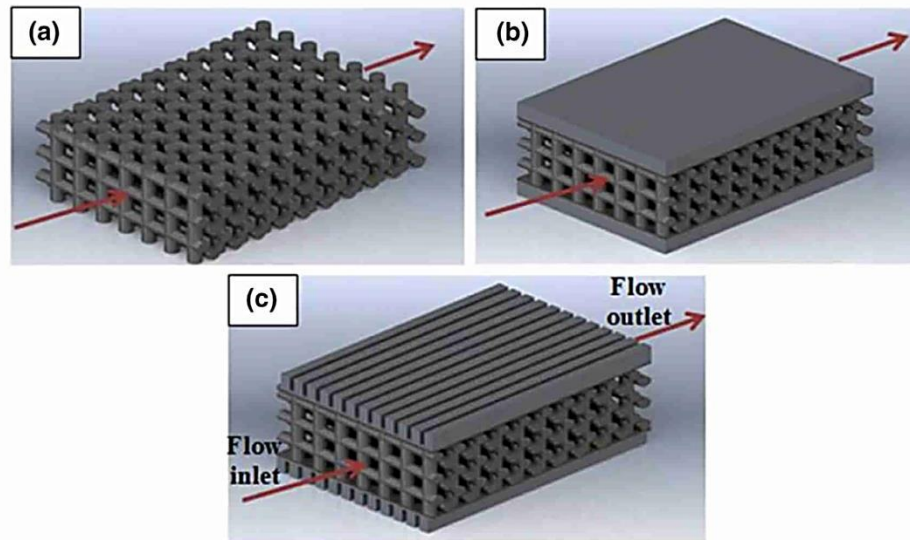
The WMHE design presented so far offers some areas for improvement due to the high cost arising from the labor intensiveness of the fin manufacturing process. Further developments of WMHE production methods have yielded the concept of a new additive manufacturing process to produce near-net-shaped fin arrays (using the CGDS process) on a thin bond coat layer produced using arc spray to seal the WMHE, as illustrated in Fig. 4. These heat-exchanger unit cells can then be stacked and attached to manifolds to form a full heat exchanger. The manufacturability of the near-net-shaped fin arrays used for this type of heat exchanger on aluminum bulk substrates has been detailed in the work of Cormier et al. (Ref 9), while the resulting enhanced thermal and hydrodynamics properties have been investigated and detailed in subsequent works (Ref 10, 11). The performance gains are mostly attributed to the turbulent wakes created behind the pin fins, which increase the convective heat-transfer coefficient.

Other interesting and promising types of heat exchangers are currently being developed using a similar principle, such as metal foam heat exchangers encased in a metallic skin deposited using wire-arc or plasma spray, as investigated by Salimi Jazi et al. (Ref 12, 13). In this work, wire arc- or plasma-sprayed processes are used to deposit Inconel 625 to seal commercially available nickel foam bricks and to create fully enclosed heat-exchanger unit cells.

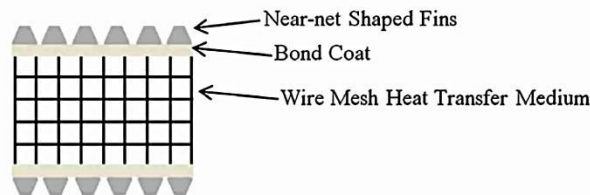
For compact heat exchangers produced using thermal-spray processes to become commercially viable, the adhesion strength and the cohesion between the various layers (heat-exchanger media, sprayed metallic skin, and net-shaped fins) must be sufficient to ensure that failure does not occur due to shear stress associated with the drag from the fluid being pumped through the mesh core and through the fin array. This parameter has yet to be investigated and detailed in the literature.

The adhesion between coatings produced using either cold-sprayed or arc-sprayed processes and their substrate has been studied extensively over the past many years (Ref 14–18). Failures of the coatings are usually attributed to either to the adhesive failure of the coating/substrate





**Fig. 3** Successive steps to manufacture a wire mesh heat exchanger: (a) Stacked wire mesh; (b) Deposited coatings; (c) Machined external fins



**Fig. 4** Schematic cross section of the improved Wire Mesh Heat Exchanger (WMHE)

interface, where the coating would detach from the substrate, or to the cohesive failure of the coating itself, where the coating would detach from itself through the propagation of cracks between coating particles (Ref 19). A mix of adhesive and cohesive failure modes can also be observed (Ref 19). Cohesive failures can generally be delayed by increasing the coating's tolerance to deformation, while adhesive failures can be delayed by improving the quality of the bonding at the interface between the substrate and the coating.

In this work, the characterization of adhesion as well as the heat-transfer performance of fin arrays produced using the CGDS fin production technique on a layer of arc-sprayed bond coat are detailed. Experiments have been performed to decouple the thermal and mechanical performances attributed to the bond coat's roughness from the effect attributable to the change of bond coat material by changing the roughness of the bond coat using a given material and by changing the bond coat material while keeping the same roughness value. Furthermore, the performance of a fin array sample with a cold-sprayed bond coat (instead of the typical arc-sprayed bond coat) was evaluated to determine the viability of this type of system to replace the arc-sprayed bond coat/cold spray structure currently used. The thermal performance inves-

tigations were performed for one configuration of uniformly distributed fins with a fin density of 12 fins per inch with a fin height of 1.0 mm, at Reynolds numbers representative of typical flow regime used for these heat exchangers.

## 2. Fin Array's Production Process Description

### 2.1 Sample Preparation and Bond Coat Production

Thin bond coats were produced on aluminum substrates of selected samples to assess the performance differences between fin arrays deposited directly on substrates and those deposited on different types of bond coats. The deposition was done on 6061 aluminum substrates instead of a wire mesh core allowing for the segregation of the bond coat/fin system from the heat-exchanger core. A commercially available Jet Force Arc Spray System 57456HD (Thermion Inc., Silverdale, Washington, United States) was used to produce nickel/aluminum alloy bond coats. The aluminum substrates were grit-blasted before the bond coat deposition using 24 grit alumina particles at 120 psi before depositing the arc-sprayed bond coats.

An SST-EP CGDS system (CenterLine (Windsor) Limited, Windsor, Ontario, Canada) was used to produce a cold-sprayed pure aluminum bond coat and to produce the pyramidal fin arrays. A polymer nozzle with a 3.7-mm throat diameter and a 6.4-mm exit diameter was used to spray pure aluminum, to prevent nozzle clogging. No surface preparation was performed on the substrates used for the production of the cold-sprayed bond coats.

**Table 1 Bond coat spray parameters**

| Sample           | Bond coat type         | Carrier gas    | Bond coat material | Inlet pressure (MPa) | Nozzle inlet gas temperature (°C) | Standoff distance (mm) | Traverse speed (mm/s) | Voltage (V) | Number of passes |
|------------------|------------------------|----------------|--------------------|----------------------|-----------------------------------|------------------------|-----------------------|-------------|------------------|
| ASN <sub>2</sub> | Arc spray              | N <sub>2</sub> | BondArc® Wire75B®  | 0.6                  | N/A                               | 100                    | 1500                  | 39.5        | 10               |
| ASAir            | Arc spray              | Air            | BondArc® Wire75B®  | 0.6                  | N/A                               | 100                    | 1500                  | 38.5        | 10               |
| CGDS             | Cold gas dynamic spray | N <sub>2</sub> | Al SST-A5001       | 1.7                  | 350                               | 15                     | 30                    | N/A         | 1                |


**Fig. 5** SEM image of as-received pure aluminum feedstock powder

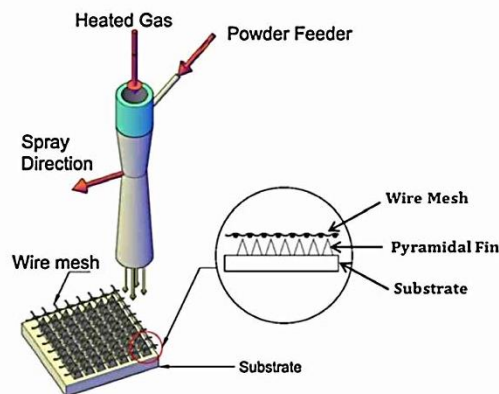
Table 1 presents the spray parameters used for the production of the bond coats by both spray techniques. Bond coats were produced using air-propelled arc spray to determine if there is a mechanical or thermal performance loss due to the added impurities (such as oxides) in the coating, compared with the coating produced using nitrogen as a propellant.

Bond coats and fins were deposited on 5.1 cm by 5.1 cm aluminum alloy substrates (Al6061 T6). The substrates were cleaned and degreased using acetone after being cut to size. Selected substrates were grit blasted to determine the effect of adding surface roughness on both adhesion and thermal performance. The fins were sprayed taller than the desired height, and were subsequently ground down to size to ensure production consistency, as detailed in Dupuis et al. (Ref 10).

## 2.2 Feedstock Material

Pure aluminum (99.8% Al), distributed commercially under the name SST-A5001 (CenterLine (Windsor) Limited, Windsor, ON, Canada) was used. The powder size is characterized by a D10 value of 13.29 microns, a D50 value of 25.35 microns, and a D90 value of 49.58 microns, as measured using a laser diffraction analysis apparatus (Microtrac model S3500, Montgomeryville, Pennsylvania, USA). The powder presents an elongated, nonspherical geometry, as seen in Fig. 5.

The wire used for the arc-sprayed process is a commercially available BondArc® Wire75B from Praxair

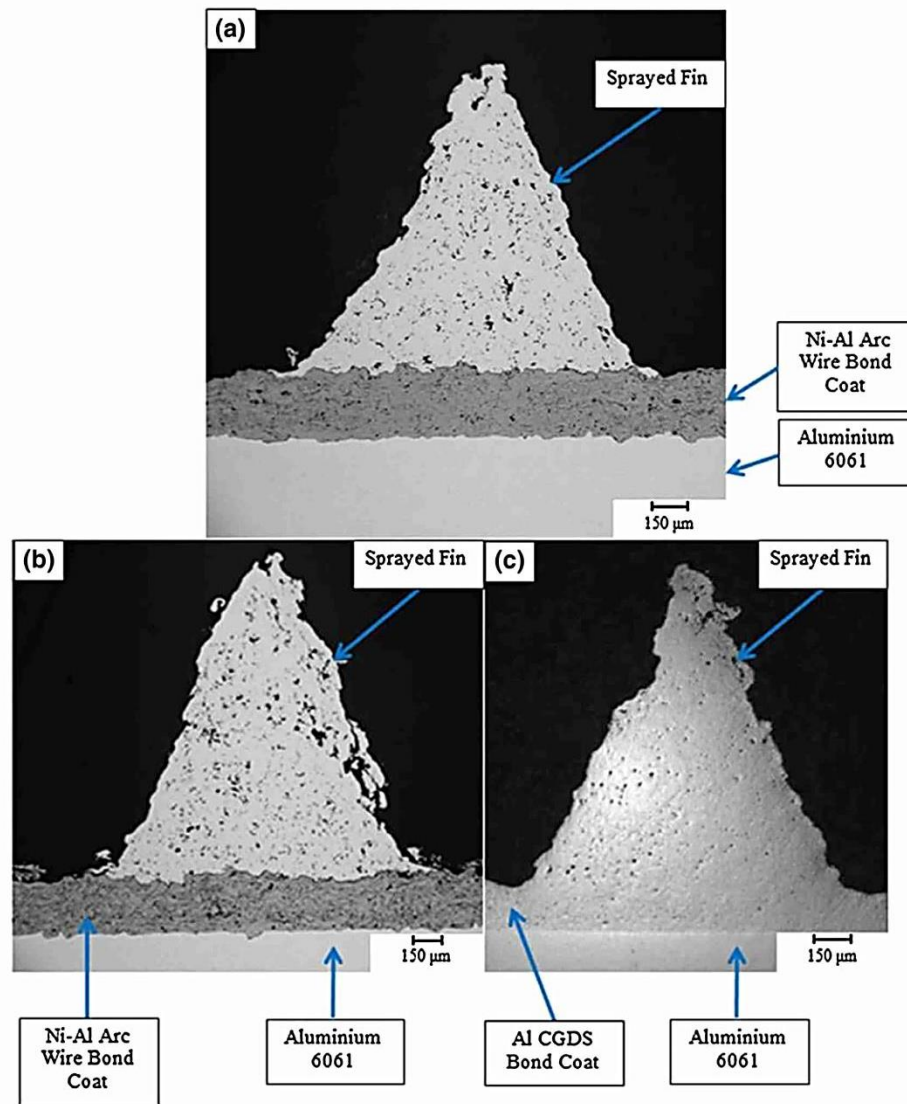

**Fig. 6** Schematic of the masking technique utilized to produce pyramidal fin arrays

(Danbury, Connecticut, USA). The wire composition is 95/5 %wt nickel/aluminum alloy.

## 2.3 CGDS Fin Production Technique

The production method used to create square-based pyramidal fin arrays using cold spray is illustrated in Fig. 6. It consists of using a mask to shield parts of the substrate from deposition of the particles accelerated by the cold-sprayed system. This allows the user to build features of various shapes and dimensions on the substrate. The operating principles of CGDS result in the preferential build-up of material in the center of the mask openings, allowing for the construction of features with reducing cross sections along the height, such as pyramids or cones (Ref 9). Commercially available, plain woven, steel wire mesh (McMaster-Carr, Aurora, OH, USA) was used as the mask. This configuration creates a pyramidal fin array (Fig. 6) which closely mimics the reversed geometry of the chosen wire mesh, with peaks aligning with the holes of the screened area. Cold-spray process parameters were chosen to ensure that the deposited geometry is as close to the idealized geometry as possible. Additional details on the production of pyramidal pin fins can be found elsewhere (Ref 9, 10). The wire mesh mask chosen for the thermal performance study has 12 wires per inch with a wire diameter of 0.035" (0.89 mm) in both the longitudinal and transverse directions. Selected specimens for the adhesion strength tests used a 16 wire per inch and 0.028" (0.71 mm) wire mesh mask, and a 24 wire per inch and 0.014" (0.36 mm) mask to study the fin density's effect. The spray conditions used to deposit the fins are a gas temperature of





**Fig. 7** Cross sections of pyramidal fins deposited on diverse bond coats: (a) Arc-sprayed bond coat propelled with nitrogen (HT1-ASN<sub>2</sub>); (b) Arc-sprayed bond coat propelled with air (HT2-ASAir); (c) Cold-sprayed bond coat (HT3-CGDS)

350 °C, gas pressure of 0.6 MPa, and a stand-off distance between the nozzle and the mesh of 15.0 mm.

Typical fin array specimens that were deposited on arc-sprayed Ni-Al bond coats produced using either nitrogen or air as propellant gas are shown in Fig. 7(a) and (b), as well as a specimen deposited on a CGDS pure aluminum bond coat (Fig. 7c).

### 3. Experimental Procedure

#### 3.1 Sample Surface and Fin Characterization

Standard metallographic procedures were used to obtain cross sections of the sprayed samples. Control speci-

mens were sprayed to obtain a standard for comparison and to establish a porosity level baseline for both types of arc-sprayed bond coats. Image analysis was performed using Clemex Vision Lite (Clemex Technologies Inc., Longueuil, Quebec, Canada) image analysis software. Porosity levels are reported with a mean value of 10 measurements for each dataset. The porosity levels reported for the air-propelled arc-sprayed coatings also include the oxide content due to the high-temperature oxidation occurring during the spray process, while the nitrogen-propelled porosity levels accurately represent only the void contents in the coating, due to the negligible quantity of oxides found in this type of coating (Ref 20). The inclusion of the oxide content in the reported porosity values stems from the inability of accurately discriminat-



ing between the two types of imperfections when using the image analysis software. Since both these imperfections have a similar and negative impact on heat transfer, there is little value in expending resources in determining their relative proportions.

To determine if there was any metallurgical bonding between the particles, Keller's reagent was used to etch the surface of a mounted sample, which was then observed using optical microscopy.

The fin array's geometric variables were measured using a depth of field microscope (Keyence model VHX-2000, Mississauga, ON, Canada) which allows building three-dimensional (3D) images of the given samples.

To determine the hardness values of the bond coats produced, Vickers microhardness tests were performed on the cross section of the coatings using a 0.1-kg (HV0.1) load and a dwell time of 10 s using a Duramin-1 microhardness tester (Struers Inc., Cleveland, OH, USA). To avoid the effect of the stress field generated by previous indentations, the distance between two indentations and that between an indentation and an edge were kept greater than three times the length of the indentation diagonal (Ref 21). The reported values are the average of five indentations for each sample, taken from random locations of polished coatings, at least three times the diagonal away from the interfaces between the various layers.

The surface roughness of the samples was evaluated using SRG-4000 Surface Roughness Tester profilometer (Phase II, Upper Saddle River, NJ, United States). Five measurements per sample were collected; each was characterized by the arithmetic average of the absolute values of the profile height deviations from the mean line ( $R_a$ ) value of the surface average.

Table 2 shows the thickness, porosity, and roughness of the different types of bond coats and of the uncoated substrates. Table 3 shows the important parameters that differentiate the pin fin arrays used for the adhesion study (AD samples), while Table 4 presents the important parameters of the fin arrays for the thermal performance tests (HT samples). The sample height was not strictly controlled for the adhesion studies other than the one focusing on fin height since this parameter was determined earlier not to have a significant effect on the adhesion strength of the specimens in question, as presented in section 4.1. For the thermal performance studies, pyramidal fin arrays were produced with a fin height of 1.0 mm using a wire mesh mask of 12 wires per inch and a wire diameter of 0.035" (0.89 mm).

Table 5 shows the grit blasting parameters used to produce selected samples for the study of the surface roughness effect. The grit blasting was performed at a 45° angle. The substrates were cleaned and degreased using acetone prior to this operation, and were cleaned in an ultrasonic bath for 10 min after the grit blasting.

### 3.2 Shear Test Apparatus

Adhesion strength analysis of the sprayed coatings was performed using the shear test standard EN15340-2007

(Ref 19). This test standard better describes the behavior of the coating when subjected to shearing loads which are encountered in industrial applications of heat exchangers rather than purely tensile testing typically used in thermal-spray-coating applications. These shearing loads arise from the fluid drag on the pin fins. The load is applied using an indenter [geometry prescribed by the cited standard (Ref 19)] attached to a commercially available tensile testing machine (Instron model 4482, High Wycombe, Bucks, UK) equipped with a static load cell of 100 kN, until failure occurs. The tests were performed on samples with a single row of 24 fins (50.8 by 5.0 mm). The shear distance was set to 50  $\mu$ m from the coating/substrate or the coating/bond coat interface, as required in EN15340-2007. The adhesion strength is obtained by dividing the measured shear force by the pin fins' sheared area. The reported values are the average of 15 specimens for each group. A schematic of the apparatus used for this type of testing is shown in Fig. 8(a). Figure 8(b) shows the sharp changes in load/displacement curve that occur when the indenter initially contacts the fins and when the failure of the fins occurs.

### 3.3 Fin Performance Testing

**3.3.1 Thermal Performance Test Apparatus.** Performance of the various fin arrays produced was assessed using an in-house heat-transfer and pressure-drop test apparatus. A schematic of the apparatus is shown in Fig. 9. The main components of this system are the air supply, the test section, the heating pad, and the data acquisition system. The air is supplied by a compressed air line equipped with a filter and dehumidifier. The air inlet flow rate is controlled by a needle valve and is measured using a flow meter. This air is smoothly expanded into a horizontal rectangular channel with a diffuser and a honeycomb flow straightener. An unobstructed 457-mm-long rectangular channel ensures that the flow is fully developed before reaching the test section. The test sample is subjected to a constant heat flux provided by a strip heater.

A set of differential pressure measurement taps are spaced 12.7 mm upstream and downstream from the middle of the test section along the centerline of the apparatus. A thermocouple is located 6.0 mm ahead of the test section and another 6.0 mm after the test section at mid-height to record the local average temperature of the fluid. Four thermocouples are attached to the fin array's base, with two on the side facing the incoming flow, and two on the side facing the exiting flow. On each face, a thermocouple is located on the centerline, while the other thermocouple is located 6.0 mm away from the wall. All the thermocouples used are T-type, butt-bonded, 28-gauge thermocouples.

**3.3.2 Thermal Performance Testing Procedure.** All measurements were performed at steady state and were repeated until statistical significance was achieved. Heat-transfer data samples were taken at 2 Hz over a 30-s interval. For both the hydrodynamic and thermal tests, five sets of data samples were recorded for each flow rate



**Table 2 Surface characterization of the bond coats**

| Sample           | Material          | Bond coat thickness (μm) | Porosity (%) <sup>a</sup> | Roughness (Ra) (μm) |
|------------------|-------------------|--------------------------|---------------------------|---------------------|
| ASN <sub>2</sub> | BondArc® Wire75B® | 242 ± 11                 | 2.0 ± 1.0                 | 13.9 ± 1.5          |
| ASAir            | BondArc® Wire75B® | 229 ± 19                 | 8.3 ± 2.2                 | 10.5 ± 1.2          |
| CGDS             | Al SST-A5001      | 310 ± 15                 | 1.8 ± 0.8                 | 18.7 ± 2.2          |
| Bulk             | Al 6061           | N/A                      | N/A                       | 1.4 ± 0.1           |

<sup>a</sup> Porosity includes oxide layer for the bond coat sprayed with Air (see text for more details)

**Table 3 Important variables of pyramidal fin array samples used for adhesion (AD) study**

| Sample number           | Fin density (Fins per in) | Wire diameter (in) | Fin height (mm) | Base Layer's roughness (μm) | Fin material | Study                         |
|-------------------------|---------------------------|--------------------|-----------------|-----------------------------|--------------|-------------------------------|
| AD1 - Bulk              | 12                        | 0.035              | 1.2             | 1.4 ± 0.1                   | Al SST-A5001 | Reference sample              |
| AD2 - Bulk              | 12                        | 0.035              | 1.4             | 1.4 ± 0.1                   | Al SST-A5001 | Fin height effect             |
| AD3 - Bulk              | 12                        | 0.035              | 2.1             | 1.4 ± 0.1                   | Al SST-A5001 |                               |
| AD4 - Bulk              | 12                        | 0.035              | 2.4             | 1.4 ± 0.1                   | Al SST-A5001 |                               |
| AD5 - Bulk              | 16                        | 0.028              | 1.6             | 1.4 ± 0.1                   | Al SST-A5001 | Fin density effect            |
| AD6 - Bulk              | 24                        | 0.014              | 1.1             | 1.4 ± 0.1                   | Al SST-A5001 |                               |
| AD7 - Bulk              | 12                        | 0.035              | 1.3             | 5.9 ± 0.4                   | Al SST-A5001 | Base Layer's roughness effect |
| AD8 - Bulk              | 12                        | 0.035              | 1.2             | 8.9 ± 1.4                   | Al SST-A5001 |                               |
| AD9 - Bulk              | 12                        | 0.035              | 1.2             | 11.0 ± 1.0                  | Al SST-A5001 |                               |
| AD10 - ASN <sub>2</sub> | 12                        | 0.035              | 1.4             | 13.9 ± 1.5                  | Al SST-A5001 | Bond coat effect              |
| AD11 - ASAir            | 12                        | 0.035              | 1.8             | 10.5 ± 1.2                  | Al SST-A5001 |                               |
| AD12 - CGDS             | 12                        | 0.035              | 1.7             | 18.7 ± 2.2                  | Al SST-A5001 |                               |

**Table 4 Important variables of pyramidal fin array samples (12 fin per inch, ground to 1.0 mm) used for heat-transfer (HT) performance study**

| Sample number          | Total heat-transfer area (mm <sup>2</sup> ) | Hydraulic diameter (mm) | Base Layer's roughness (μm) | Base Layer's relative roughness |
|------------------------|---------------------------------------------|-------------------------|-----------------------------|---------------------------------|
| HT1 - ASN <sub>2</sub> | 4106                                        | 0.94                    | 13.9 ± 1.5                  | 0.011                           |
| HT2 - ASAir            | 4207                                        | 0.92                    | 10.5 ± 1.2                  | 0.015                           |
| HT3 - CGDS             | 4037                                        | 0.98                    | 18.7 ± 2.2                  | 0.019                           |
| HT4 - Bulk             | 4133                                        | 0.94                    | 1.4 ± 0.1                   | 0.002                           |
| HT5 - Bulk             | 3947                                        | 0.99                    | 10.1 ± 1.0                  | 0.011                           |

**Table 5 Grit blasting parameters used for the substrate surface preparation for the study of substrate roughness effect**

| Sample number | Type of grit           | Blasting pressure (MPa) |
|---------------|------------------------|-------------------------|
| AD7 - Bulk    | Copper Slag: 20 grit   | 0.6                     |
| AD8 - Bulk    | Copper Slag: 20 grit   | 1.4                     |
| AD9 - Bulk    | Angular Steel: 25 grit | 1.4                     |
| HT5 - Bulk    | Angular Steel: 25 grit | 1.4                     |

tested. Measurements were taken at flow rates ranging from 10 to 70 SLPM, which corresponds to Reynolds numbers varying from 200 to 3500 depending on sample geometry. Deviation of the measurements from the mean value was less than 5%, which led to error bars that were smaller than the markers. Since the error bars were not visible, they were not included in the thermal performance graphs of section "Heat Transfer Performance". A detailed description of the apparatus' accuracy and of the instrument's uncertainties is given in Cormier et al. (Ref 9). For hydrodynamic performance tests, a set of pressure taps is used to obtain the fin array's friction losses, which include skin friction and form drag. Radiative heat-trans-

fer losses are neglected as they would account for less than 0.5% of the total heat input (Ref 22, 23).

**3.3.3 Data Reduction.** To determine if the fin arrays can survive the stress involved with the industrial applications, the shear stress imposed by the fluid is computed using the following relation:

$$\tau_d = \frac{\frac{1}{2} C_d \times \rho \times U^2 \times A_{\text{front}}}{A_{\text{base}}} \quad (\text{Eq 1})$$

where  $\tau_d$  is the shear stress due to fluid drag,  $\rho$  is the fluid's density,  $U$  is the fluid's velocity,  $A_{\text{base}}$  is the fin's base area,  $A_{\text{front}}$  is the fin frontal area, and  $C_d$  is the drag coefficient of the surface, with the last two parameters assumed, in the worst case scenario, to be equal to that of a rectangular prism with a width and length equal to the base dimensions of the pyramid with the same height as the pyramidal fins ( $C_d=1.05$ ) (Ref 24). This reduction yields the following equation:

$$\tau_d = 0.525 \rho \times U^2 \times \frac{H}{L} \quad (\text{Eq 2})$$

where  $H$  is the fin height, and  $L$  is the base length (in the streamwise direction).

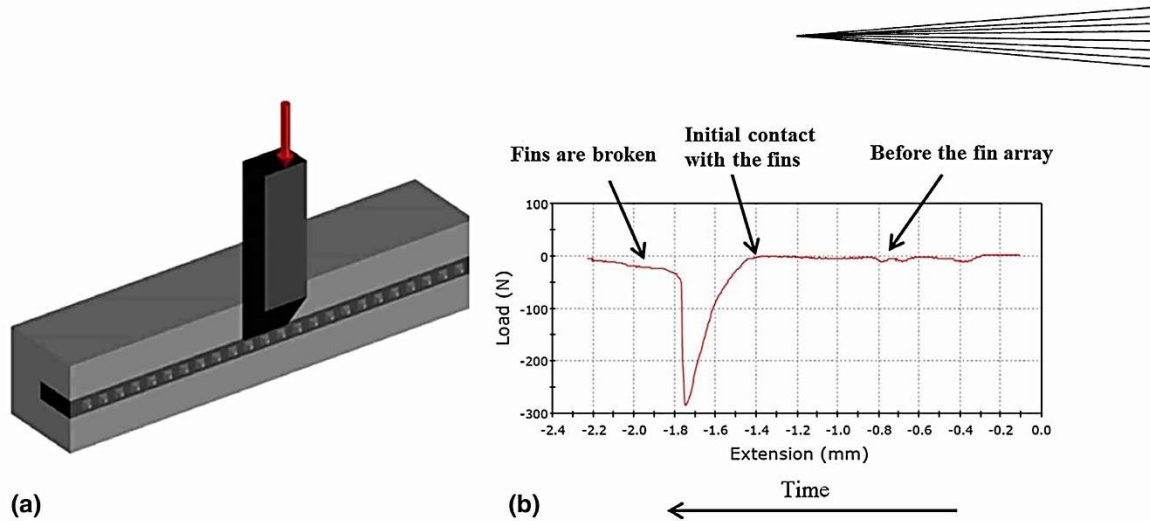


Fig. 8 (a) Diagram of the adhesion test apparatus; (b) Typical load-displacement curve obtained

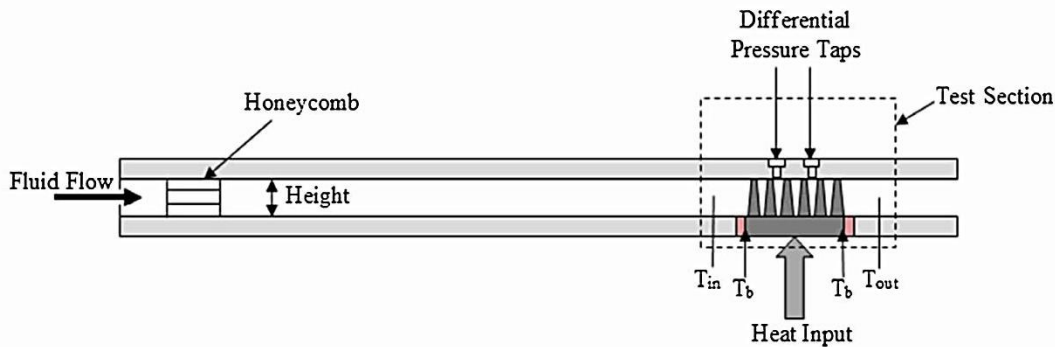


Fig. 9 Heat-transfer fixture utilized to evaluate the fin arrays' heat-transfer performance

The thermal performance comparison is done according to the procedure detailed in Sahiti's work (Ref 25). This method is based on relating the thermal and hydrodynamic data plotting thermal conductance as a function of the required pumping power for different flow conditions for a unit volume of heat exchanger. This yields curves which can be directly compared one with another, with the most efficient fin array having a higher thermal conductance at a given pumping power.

The required pumping power input was calculated according to Sahiti's definition (Ref 26):

$$e_v = \frac{\dot{V}_f \times \Delta P_{fin}}{\eta \times V} \quad (\text{Eq 3})$$

In the previous relation,  $e_v$  is the pumping power per unit volume,  $\dot{V}_f$  is the volumetric flow rate of the fluid,  $\Delta P_{fin}$  is the pressure drop measured,  $\eta$  is the fan efficiency, and  $V$  is the sample heat exchanger's volume. A fan efficiency of 0.8 was chosen as a reasonable value to perform these calculations (Ref 26).

The thermal conductance associated with convection is given by

$$UA_{conv} = \frac{1}{R_{conv}} = h \times A_{conv} \quad (\text{Eq 4})$$

where  $R_{conv}$  is the resistance to convective heat transfer,  $h$  is the convective heat-transfer coefficient, and  $A_{conv}$  is the area where the convective heat transfer occurs.

The thermal conductance associated with conduction can be calculated using

$$UA_{cond} = \frac{1}{R_{cond}} = \frac{k \times A_{cond}}{t} \quad (\text{Eq 5})$$

where  $R_{cond}$  is the resistance to heat transfer by conduction,  $k$  is the thermal conductivity,  $t$  is the length of the conduction path, and  $A_{cond}$  is the heat-transfer area associated with conduction.

The equivalent thermal conductance per unit volume for the heat-exchanger specimen can be calculated using

$$UA_v = \frac{1}{R_{cq} \times V} = \frac{q}{\Delta T_{lm} \times V} \quad (\text{Eq 6})$$

where  $R_{cq}$  is the thermal circuit's equivalent resistance,  $q$  is the total heat input rate to the system, and  $\Delta T_{lm}$  is the log-mean temperature difference between the fluid and the fin array.



The total heat input rate to the system is calculated using

$$q = \dot{m} \times (T_{\text{out}} - T_{\text{in}}) \times C_p \quad (\text{Eq 7})$$

where  $\dot{m}$  is the fluid mass flow rate,  $(T_{\text{out}} - T_{\text{in}})$  is the fluid temperature difference between the entrance and the exit of the test section, and  $C_p$  is the fluid's thermal capacitance.

The log-mean temperature difference is calculated using

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (\text{Eq 8})$$

where  $\Delta T_1$  and  $\Delta T_2$  are the temperature difference between the fin surface and the flow at the inlet and outlet, respectively.

To determine the relative roughness of the base layer's surface for the heat-transfer samples, the following relation is used:

$$\varepsilon = \frac{r}{D_h} \quad (\text{Eq 9})$$

where  $\varepsilon$  is the relative roughness of the surface,  $r$  is the surface roughness, and  $D_h$  is the fin hydraulic diameter.

The Reynolds number based on the hydraulic diameter is given by

$$\text{Re}_D = \frac{\rho U D_h}{\mu} \quad (\text{Eq 10})$$

where  $\rho$  is the fluid density and  $\mu$  is the fluid's dynamic viscosity.

## 4. Results

### 4.1 Shear Tests

For all the tests performed in this section, the only failure mode observed was adhesive failure at the bond coat/fin interface or at the substrate/fin interface. The theoretical shear stress value obtained from Eq 1 in section "Data Reduction" for pure water flowing at a Reynolds number of 3500 (typical of the commercial application for this type of WMHE) is found to be 10.0 kPa. This is much lower than the lowest strength value measured (18.4 MPa).

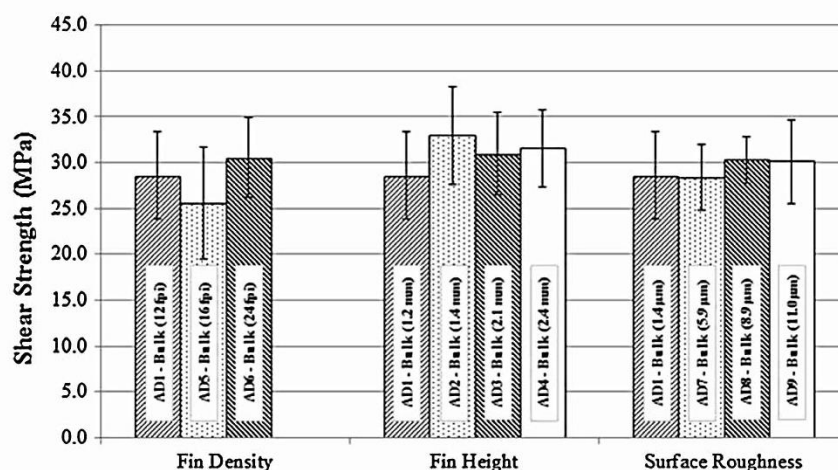
Figure 10 shows the effect of varying the fin density, the fin height, and the surface roughness of the substrate on the shear strength of pure aluminum fins deposited directly on the Al6061 substrate. No significant differences can be observed between the different samples, which have an average of 27.2 MPa in shear strength. These differences fall well within the experimental error margin, which leads to the conclusion that none of these parameters has a significant effect on the adhesion strength between pure aluminum and an Al6061 substrate. This leads to three different conclusions. First, the changes induced

in the flow structure of the CGDS propellant through the different type of wire mesh mask used in this work minimally affect particle velocity, as a denser mask causes the resulting fins to have the same adhesion for the three fin densities tested. Consequently, one can deduce that the particle impact velocity is not affected by the mask's mesh size used, as it would otherwise lead to reduced adhesion strength. Second, the impingement effect that is commonly noted in cold spray does not increase the adhesion of the first few layers of particles for pin fins with heights between 1.2 and 2.4 mm. Finally, increasing the roughness of the base layer does not increase the amount or the quality of the mechanical anchoring between the fins and the bond coat. This leads one to believe that the adhesion strength is not affected by these parameters since they do not change the type or the quality of bond formation between the substrate and the first few layers of the deposited material.

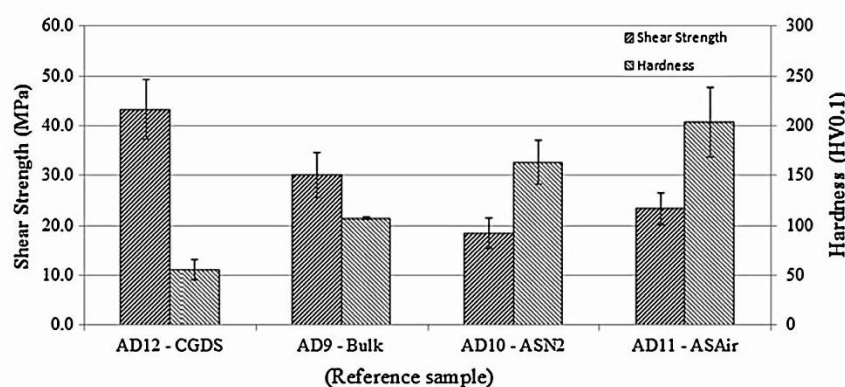
The adhesion between pure aluminum pyramidal fins and diverse types of bond coat surfaces is shown in Fig. 11, with the adhesion to the substrate (without a bond coat) as a reference. It also presents the hardness values of the material on which the fins have been deposited. The best adhesion is found to be with the CGDS-produced bond coat, while the lowest adhesion is obtained with the arc-sprayed bond coat produced using nitrogen. A relationship between the adhesion strength and the bond coat (or substrate) hardness can be observed as increasing the hardness of bond coat layer lowers the adhesion strength. This reduction in adhesion strength does not seem to be attributable to the decrease in surface roughness, as the control sample on Al6061 has the same roughness as the nitrogen-propelled arc-sprayed coating. It is thus surmised that the reduction in the adhesion strength with the increasing substrate top coat's hardness is due to the cold-sprayed particles becoming less embedded in the higher hardness material. Since the embedment is the source of the mechanical anchoring of the particles, it follows logically that less embedment equals a lower adhesion value. It is useful to note that although the air arc-sprayed coating does not seem to follow the same trend where increasing hardness yields lower adhesion strength, the observed effect for this sample is most likely due to the presence of oxides which yields a larger range of hardness values than those for the other samples. Furthermore, when looking at the error bars it seems that the increase in hardness and the increase in adhesion strength, compared with the nitrogen-propelled arc-sprayed sample, are marginal.

The fin adhesion to the substrates is maintained purely by mechanical anchoring, as supported by Fig. 12. This figure shows typical cross sections of the control sample (AD-1), one of the substrate roughness sample (AD-9), the nitrogen-propelled arc-sprayed bond coat sample (AD-10), the air-propelled arc-sprayed bond coat sample (AD-11), and the cold-sprayed bond coat sample (AD-12). In all the cross sections of the samples investigated, there is no evidence of metallurgical bonding between the fin material and the bond coat or the substrate. This,





**Fig. 10** Effect of fin density, fin height, and base layer's surface roughness on the shear strength of pyramidal fin arrays produced on bulk Al 6061 substrates



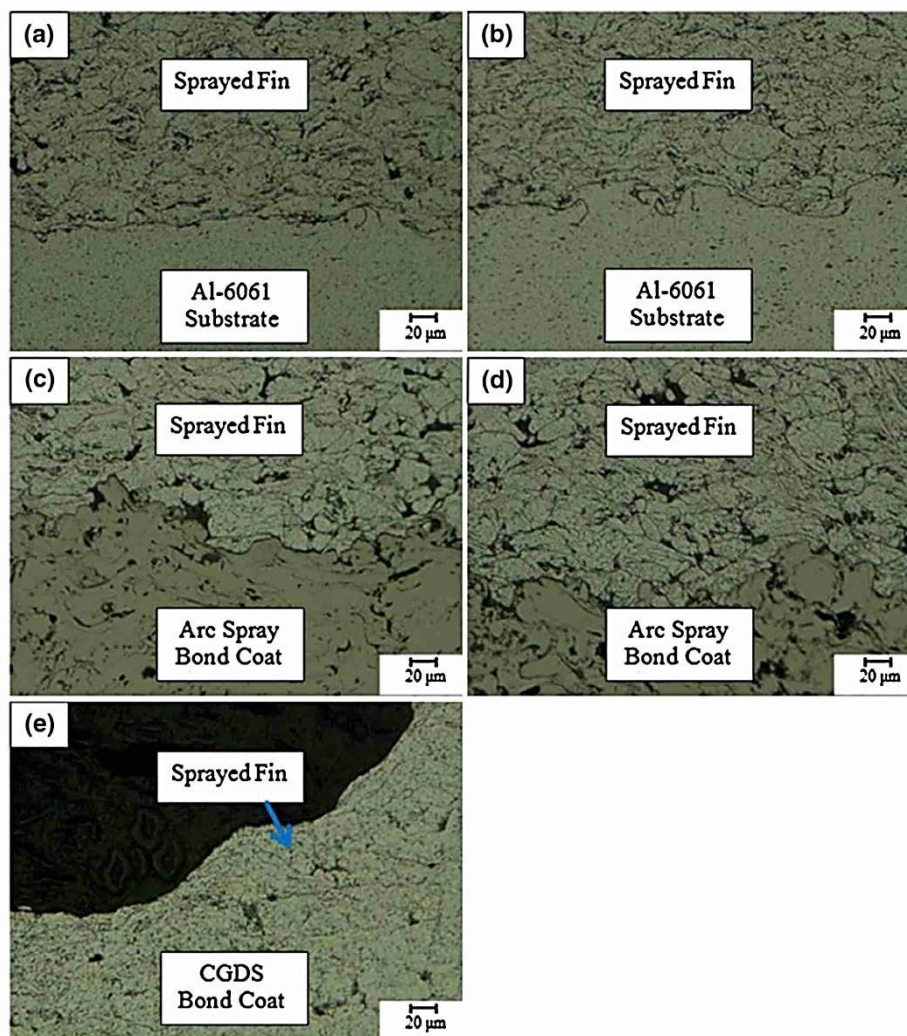
**Fig. 11** Shear strength of pure aluminum pyramidal fin arrays deposited on bond coats and their respective hardness values

combined with the relatively low shear strength values, leads one to believe that the increase in the shear strength observed is purely due to a better mechanical anchoring at the fin/substrate or the fin/bond coat interface. This is substantiated by Fig. 13(a) and (b), which show typical fracture surfaces for fin samples on substrates and on bond coats, respectively. The absence of shear bands indicates that there is no clear evidence of metallurgical bonding at the interface.

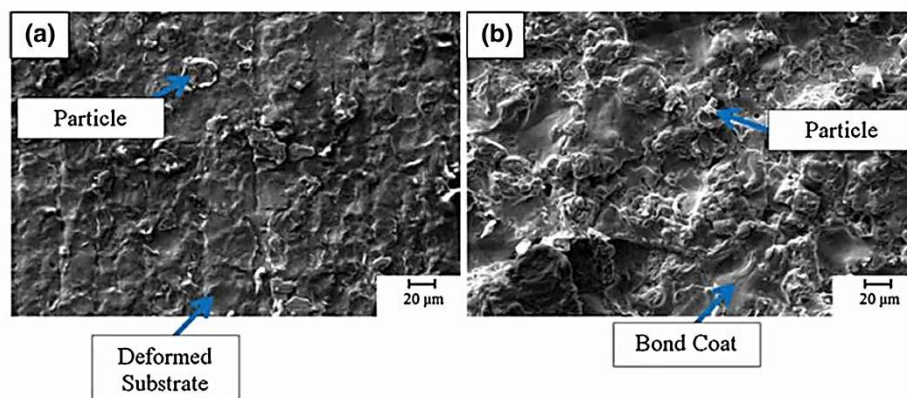
## 4.2 Heat-Transfer Performance

The thermal performance results are presented in Fig. 14, showing the thermal efficiency of the fin arrays produced on the three types of bond coats and two control samples, one with the same relative roughness ( $\epsilon$ ) as that of the HT1 sample (nitrogen-propelled arc-sprayed sample) and the other with a relative roughness roughly one order of magnitude lower than those of the four other samples. The addition of the bond coat and thus the resulting increase in the base layer's thermal resistance

seem to have a negligible effect on the thermal performance of the fin array, irrespectively of the nature of the bond coat. It can be noted that the efficiency of the four fin arrays with a higher relative roughness have approximately 20% higher thermal conductance values at the same pumping power than the sample with the smoother base (1230 kW/m<sup>3</sup>K compared to 1020 at 4000 kW/m<sup>3</sup> pumping power, for example), regardless of the bond coat's presence or material, for pumping powers larger than 2000 kW/m<sup>3</sup>. It is hypothesized that in this type of system, the base layer's roughness contributes to making the boundary layer attached to the base of the fin arrays more turbulent than for the smoother specimen. This increase in turbulence leads to an increase in the convection coefficient, which translates to a decreased thermal resistance to convection. For the particular heat-exchanger system proposed, the thermal resistance associated with convection from the fin array and the base to the fluid is much higher than the thermal resistance associated with conduction through the fin array's base layer. It is thus surmised that the resistance to conduction through the

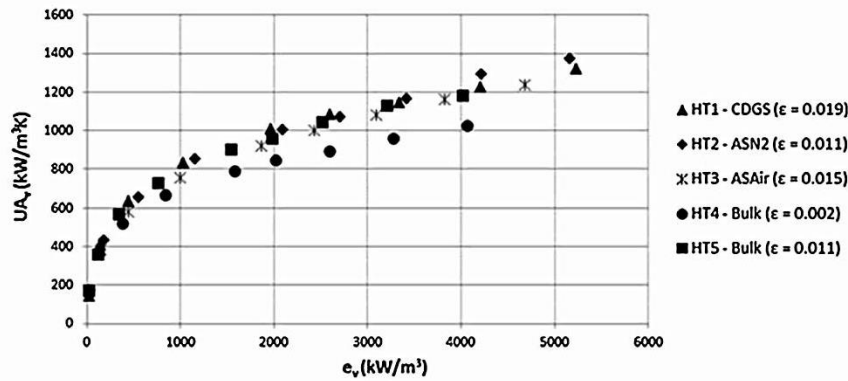


**Fig. 12** Etched cross section of the fins-substrate/bond coat interface showing purely mechanical bonding for samples: (a) AD1-Bulk; (b) AD9-Bulk; (c) AD10-ASN<sub>2</sub>; (d) AD11-ASAir; (e) AD12-CGDS



**Fig. 13** Scanning Electron Microscope (SEM) images of fracture surface for (a) AD1-Bulk; (b) AD10-ASN<sub>2</sub>





**Fig. 14** Volume-based heat-transfer performance comparisons between HT1-CGDS (triangle), HT2-ASN<sub>2</sub> (diamond), HT3-ASAir (star), HT4-Bulk (circle), and HT5-Bulk (square) with the respective sample relative roughness values indicated in the legend

bond coat is a negligible term in this heat-transfer system due to the thinness of this layer. The net effect of adding a bond coat, in this scenario, would result in a performance increase due to the increased roughness of the top layer compared with an untreated Al6061 substrate.

## 5. Summary and Conclusion

This paper investigated the effects on the adhesion strength and the thermal efficiency of compact WMHE's external fin systems produced using the masked CGDS technique with various types of bond coat layers. In addition, experiments have been performed to decouple the effect attributable to the material properties of the bond coat from the effect of the bond coat's roughness on both the adhesion and the thermal performance of the system. The following conclusions are proposed:

1. The shear strength of the proposed fin system is largely sufficient to sustain the theoretical drag pressures encountered in typical application scenarios.
2. The fin height, fin density, and surface roughness of the substrate do not have a significant influence on the adhesion strength of the tested specimens.
3. An inversely proportional relationship has been identified between the bond coat hardness and the adhesion strength of the pyramidal fin specimens, indicating that samples with the lowest hardness bond coats had the highest adhesion strength.
4. The bond coats produced have a positive effect on the heat-transfer efficiency of the pin fin arrays produced due to the increased roughness of the base layer's surface compared with the as-received Al6061 substrate. The gain is explained by an increase in the level of turbulence in the boundary layer for rougher surfaces.
5. The added resistance to conduction through the bond coat layer is found to be negligible due to the fact that the bond coat's layer is relatively thin (250  $\mu\text{m}$ ).

## Acknowledgment

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## References

1. A.B. Lovins, *Small is Profitable: The Hidden Economic Benefits of Making Electrical Resources the Right Size*, Rocky Mountain Institute, 2002
2. S.M. Kaplan and F. Sissine, *Smart Grid: Modernizing Electric Power Transmission and Distribution; Energy Independence, Storage and Security; Energy Independence and Security Act ... and Resiliency*; Integra, TheCapitol.Net Inc., Alexandria, 2009
3. D. Bohn, *Micro Gas Turbine and Fuel Cell: A Hybrid Energy Conversion System with High Potential*, NATO Research and Technology Organisation, 2005
4. J. Tian, T. Kim, T.J. Lu, H.P. Hodson, D.T. Queheillalt, D.J. Sypek, and H.N.G. Wadley, The Effects of Topology Upon Fluid-Flow and Heat-Transfer Within Cellular Copper Structures, *Int. J. Heat Mass Transf.*, 2004, **47**(14-16), p 3171-3186
5. Z. Anxionnaz, M. Cabassud, C. Gourdon, and P. Tochon, Heat Exchanger/Reactors (HEX Reactors): Concepts, Technologies: State-of-the-Art, *Chem. Eng. Process.*, 2008, **47**(12), p 2029-2050
6. R.L. Shaner, *Heat Exchanger Having Metal Wire Screens, and Method of Making Stack of Screens Therefor*, US4840228, Year of Priority (Issued): 1985 (1989)
7. J. Chisholm, *Method of Making a Crimped Wire Mesh Heat Exchanger/Sink*, US4843693, Year of Priority (Issued): 1988 (1989)
8. J. Assaad, A. Corbeil, P. Richer, and B. Jodoin, Novel Stacked Wire Mesh Compact Heat Exchangers Produced Using Cold Spray, *J. Therm. Spray Technol.*, 2011, **20**(6), p 1192-1200
9. Y. Cormier, P. Dupuis, B. Jodoin, and A. Corbeil, Net Shape Fins for Compact Heat Exchanger Produced by Cold Spray, *J. Therm. Spray Technol.*, 2013, **22**(7), p 1210-1221
10. P. Dupuis, Y. Cormier, A. Farjam, B. Jodoin, and A. Corbeil, Performance Evaluation of Near-Net Pyramidal Shaped Fin Arrays Manufactured by Cold Spray, *Int. J. Heat Mass Transf.*, 2014, **69**, p 34-43
11. Y. Cormier, P. Dupuis, A. Farjam, B. Jodoin, and A. Corbeil, Additive Manufacturing of Pyramidal Pin Fins: Height and Fin Density Effects Under Forced Convection, *Int. J. Heat Mass Transf.*, 2014, **75**, p 235-244
12. H.R. Salimi Jazi, J. Mostaghimi, S. Chandra, L. Pershin, and T. Coyle, Spray-Formed, Metal-Foam Heat Exchangers for High Temperature Applications, *J. Thermal Sci. Eng. Appl.*, 2009, **1**(3), p 1-7
13. F. Azarmi, J. Saaedi, T.W. Coyle, and J. Mostaghimi, Microstructure Characterization of Alloy 625 Deposited on Nickel

- Foam Using Air Plasma Spraying, *Adv. Eng. Mater.*, 2008, **10**(5), p 459-465
14. J.G. Legoux and S. Dallaire, Adhesion Mechanisms of Arc-Sprayed Zinc on Concrete, *J. Therm. Spray Technol.*, 1995, **4**(4), p 395-400
15. X. Wang, J. Heberlein, E. Pfender, and W. Gerberich, Effect of Nozzle Configuration, Gas Pressure, and Gas Type on Coating Properties in Wire Arc Spray, *J. Therm. Spray Technol.*, 1999, **8**(4), p 565-575
16. R. Drehmann, T. Grund, T. Lampke, B. Wielage, K. Manyoats, T. Schucknecht, and D. Rafaja, Splat Formation and Adhesion Mechanisms of Cold Gas-Sprayed Al Coatings on  $Al_2O_3$  Substrates, *J. Therm. Spray Technol.*, 2014, **23**(1-2), p 68-75
17. D.J. Varacalle, D.P. Guillen, D.M. Deason, W. Rhodaberger, and E. Sampson, Effect of Grit-Blasting on Substrate Roughness and Coating Adhesion, *J. Therm. Spray Technol.*, 2006, **15**(3), p 348-355
18. M. Yandouzi, A.J. Böttger, R.W.A. Hendrikx, M. Brochu, P.F. Richer, A. Charest, and B. Jodoin, Microstructure and Mechanical Properties of  $B_4C$  Reinforced Al-Based Matrix Composite Coatings Deposited by CGDS and PGDS Processes, *Surf. Coat. Technol.*, 2010, **205**, p 2234-2246
19. EN 15340, *European Standard Method-Determination of Shear Load Resistance of Thermally Sprayed Coatings*, 2007, p. A91
20. J.R. Davis, Ed., *Handbook of Thermal Spray Technology*, ASM International, 2004
21. P.H. Shipway, D.G. McCartney, and T. Sudprasert, Sliding Wear Behaviour of Conventional and Nanostructured HVOF Sprayed WC-Co Coatings, *Wear*, 2005, **259**(7-12), p 820-827
22. S. Naik, S.D. Probert, and M.J. Shilston, Forced Convective Steady-State Heat Transfers From Shrouded Vertical Fin Arrays Aligned Parallel to an Undisturbed Air-Stream, *Appl. Energy*, 1987, **26**, p 137-158
23. J.J. Hwang and T.M. Liou, Heat Transfer and Friction in a Low-Aspect-Ratio Rectangular Channel with Staggered Perforated Ribs on Two Opposite Walls, *ASME J. Heat Transfer*, 1995, **117**(11), p 843-850
24. R.W. Fox, P.J. Pritchard, and A.T. McDonald, *Introduction to Fluid Mechanics*, 7th ed., Wiley, Hoboken, NJ, 2008
25. N. Sahiti, F. Durst, and A. Dewan, Heat Transfer Enhancement by Pin Elements, *Int. J. Heat Mass Transf.*, 2005, **48**, p 4738-4747
26. N. Sahiti, F. Durst, and A. Dewan, Strategy for Selection of Elements for Heat Transfer Enhancement, *Int. J. Heat Mass Transf.*, 2006, **49**, p 3392-3400

## **5.5 Paper #5: Streamwise Anisotropic Materials and Fin Roughness Effect**

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This peer reviewed article investigates the possibility to produce a fin array that has different materials along its length. Aluminum, nickel and stainless steel 304 pyramidal fins were successfully deposited. Thermal and hydrodynamic performance for each type of fin was evaluated as well as for a multi-material sample. The thermal conductivities of cold sprayed samples of the materials used to produce the fins were determined. Also, the surface roughness of the fins and the flow passages were studied and used in conjunction with the thermal conductivity results to explain the differences in performance. This paper addresses the objective of the fin arrays with streamwise anisotropic materials presented in section 3.6.





# Pyramidal Fin Arrays Performance Using Streamwise Anisotropic Materials by Cold Spray Additive Manufacturing

Yannick Cormier, Philippe Dupuis, Bertrand Jodoin, and Antoine Corbeil

(Submitted April 16, 2015; in revised form May 28, 2015)

This work evaluates the thermal and hydrodynamic performance of pyramidal fin arrays produced using cold spray as an additive manufacturing process. Near-net-shaped pyramidal fin arrays of pure aluminum, pure nickel, and stainless steel 304 were manufactured. Fin array characterization such as fin porosity level and surface roughness evaluation was performed. The thermal conductivities of the three different coating materials were measured by laser flash analysis. The results obtained show a lower thermal efficiency for stainless steel 304, whereas the performances of the aluminum and nickel fin arrays are similar. This result is explained by looking closely at the fin and substrate roughness induced by the cold gas dynamic additive manufacturing process. The multi-material fin array sample has a better thermal efficiency than stainless steel 304. The work demonstrates the potential of the process to produce streamwise anisotropic fin arrays as well as the benefits of such arrays.

**Keywords** additive manufacturing, cold spray, heat transfer, pin fin array, thermal conductivity

## 1. Introduction

Research focusing on the enhancement of heat exchanger performance has become a priority in areas such as gas turbine development, electronic cooling, and power generation (Ref 1–4). Indeed, heat exchangers have become common components in various engineering disciplines due to the importance and the challenges associated with thermal management. Some of the most widely used heat exchanger designs rely on fin arrays as heat transfer enhancing surfaces to be able to minimize the size and weight of such components for a given heat exchange rate (Ref 1, 4). Enhancing thermal conductance of fin arrays while decreasing the size of heat exchangers is also especially important for the aerospace and automotive industries, for which the weight and volume of components are important factors.

To increase the ratio of heat transfer area per unit volume, fins have tended to become longer and slimmer

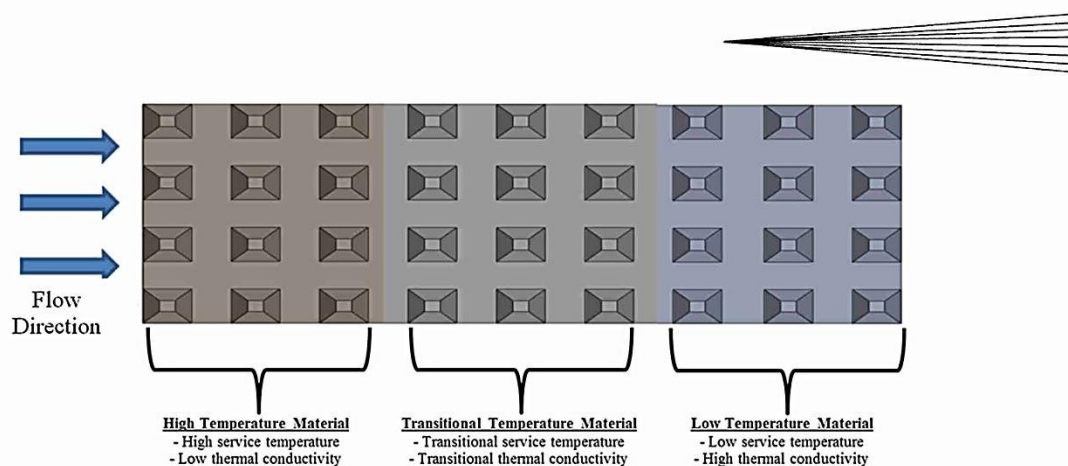
combined with a higher fin density (amount of fins per unit base area). This densification of the fin arrays has some practical limitations based on the manufacturing techniques used to produce these features. Extrusion and machining have typically been used to produce both continuous fin and pin fin arrays, respectively (Ref 5). Casting has also been used as an alternative production method for these arrays (Ref 5). All these manufacturing techniques suffer from the same drawback of increased cost as the complexity and density of the fin array increase (Ref 5). The cost of casting and extrusion increases rapidly due to the precision required for the production of the mold or die (Ref 1, 6), while for machining, the production speed decreases with tool size, which is the limiting factor for feature size and the associated tolerances (Ref 1, 7). Metal mold casting is further restrained as the design must accommodate features with taper angles and without concave details so that the part can be removed from the rigid mold after casting (Ref 1, 8). When the fin array features are too small for conventional manufacturing techniques, lithographic techniques can be used to produce fin arrays but require a significant investment in equipment (Ref 1).

Additionally, for heat exchangers with fins, the optimal geometry to reduce manufacturing costs and concerns does not necessarily correspond to the ideal geometry for thermal performance. The fin geometry has a significant effect on fin efficiency. Indeed, increasing the height to cross-section area ratio can dramatically lower the fin efficiency, thus yielding a lower overall heat exchanger thermal performance. Furthermore, fin efficiency is also dependent on the fin material thermal conductivity. For low service temperature applications (less than 200 °C), materials having a high thermal conductivity such as copper or aluminum can be used. For higher service temperature (600 °C) or highly corrosive environments,

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**Fig. 1** Schematic of a multi-material fin array separated into three sections

materials such as nickel-based alloys or stainless steel must be used. These materials suffer from the drawbacks of having lower thermal conductivities and are usually denser than the aforementioned materials, leading to fin arrays with lower thermal performance per unit mass (Ref 9). The thermal performance is further decreased when high convective coefficients are present, which makes the design of high service temperature heat exchangers using pin fin arrays a challenge (Ref 10, 11).

Another limitation of conventional fin array manufacturing methods is that they typically restrict a fin array to sections made of one homogeneous material. These sections can be joined together to create mixed material fin arrays, but this rapidly becomes uneconomical or impractical due to reliability issues with the bond between the sections. Additionally, the bonding agent itself represents a thermal barrier for the heat flow. This type of process is usually limited to plain rectangular fin geometries due to manufacturing concerns (Ref 5).

A new method for producing near-net-shaped pin fin arrays has been proposed, which uses Cold Gas Dynamic Spraying (CGDS) and a mask to selectively deposit material onto a substrate (Ref 12, 13). The adhesion of the particles to the substrate is attributed to the large plastic deformation of the particle and substrate upon impact, which yields intimate metal to metal contact, where metallic bonds and mechanical anchoring occur (Ref 14, 15). CGDS is especially well suited for the near-net-shaped production of fin arrays using substrate masking due to the low process temperatures preventing melting during the deposition process. These advantages allow for the process to be used effectively, without significant clogging, distortion, or degradation of the metallic mask.

The pin fins produced with the masked cold spray technique have a square-based pyramidal structure, with the fin density varying between 12 and 24 fins per inch and the height varying between 1.0 and 2.5 mm. The hydrodynamic and thermal performance of this type of fin array has also been studied and was found to be superior to similar size continuous fins (plain rectangular fins) (Ref 12, 13, 16). The mechanical properties of fin arrays produced using CGDS were also determined to ensure that

the stresses arising from the drag due to fluid flow could be sustained by these features without failure (Ref 17).

Since it has been proven that using CGDS as an additive manufacturing process to produce pyramidal pin fins can yield arrays with enhanced thermal properties, attention is now given to the exploration of the different materials that could be used for manufacturing this type of fin array. In this work, the thermal performances of aluminum, stainless steel 304, and nickel fin arrays are investigated. The possibility of creating multi-material heat exchangers would allow the designer the freedom of using the material combination which not only maximizes the thermal performance, but also allows for the substitution of high cost, heavy alloys by lighter and cheaper materials in the lower service temperature sections of the heat exchanger. The heat exchanger could be separated into three sections: a lower, an intermediate, and a high service temperature sections (Fig. 1) since a transfer of energy occurs between the fins and the fluid, resulting in a decrease of local fluid temperature along the length of the heat exchanger, as seen in a typical cooling curve of temperature as a function of distance (Fig. 2). The procedure for the production of this type of hybrid material fin array using the masked CGDS process is detailed, and its thermal performance is investigated.

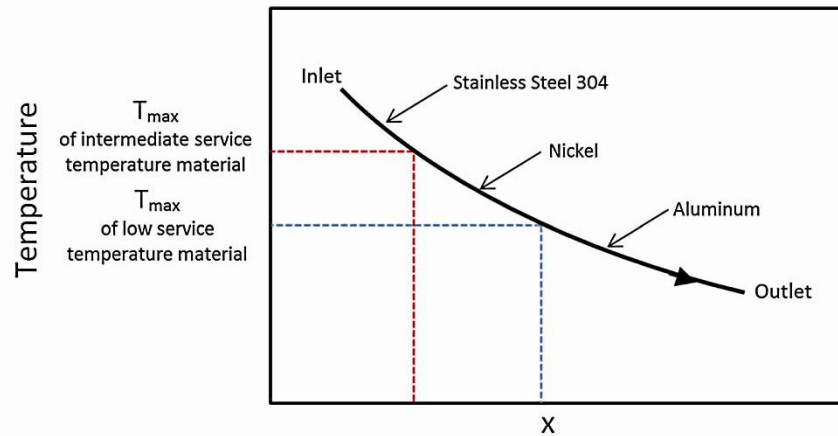
## 2. Fin Array Production

### 2.1 Feedstock Powders

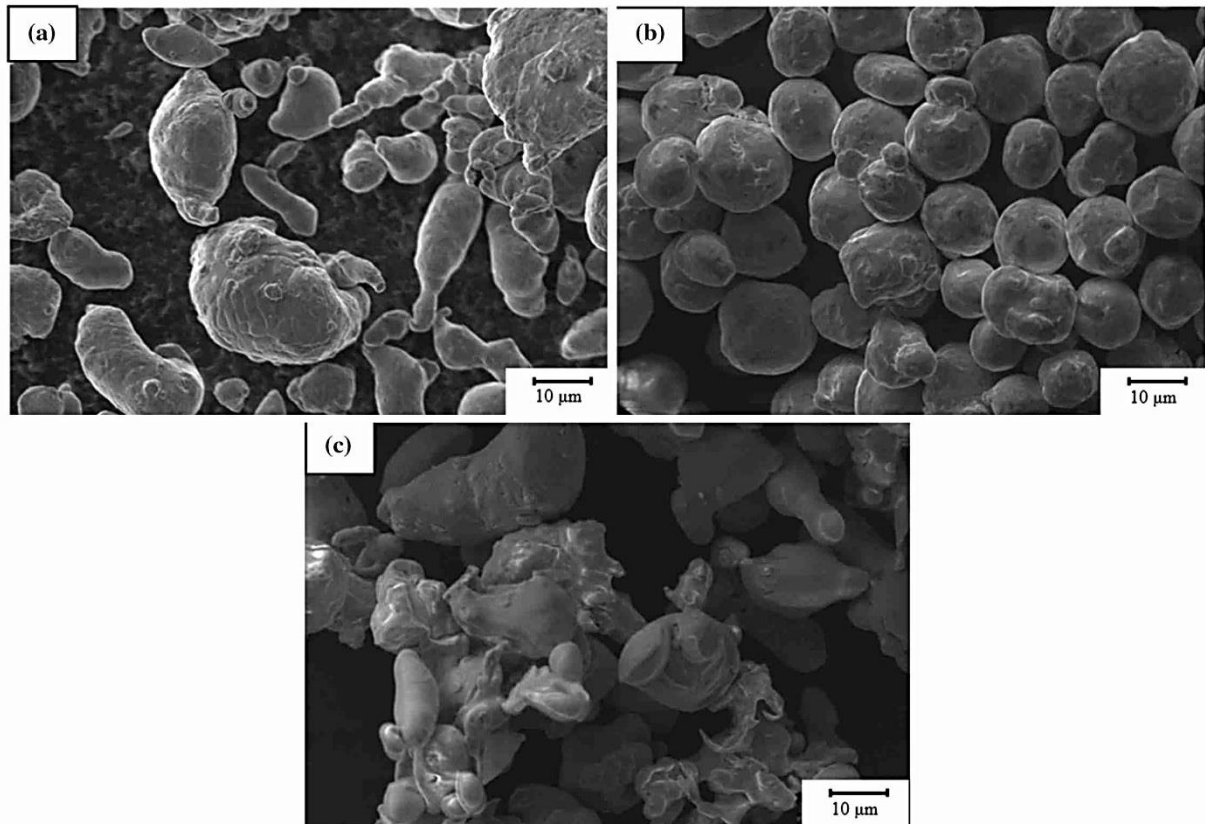
Pure commercially available aluminum (99.8% Al, CenterLine (Windsor) Limited, Windsor, ON, Canada) was used. The powder size is characterized by a D10 value of 13.29 microns, a D50 value of 25.35 microns, and a D90 value of 49.58 microns, as measured using a laser diffraction analysis apparatus (Microtrac model S3500, Montgomeryville, PA, USA). The powder presents an elongated, non-spherical geometry, as seen in Fig. 3(a).

The second powder used was a commercially pure nickel powder (99.9%, Atlantic Equipment Engineers, Upper Saddle River, NJ, USA). The powder size is characterized





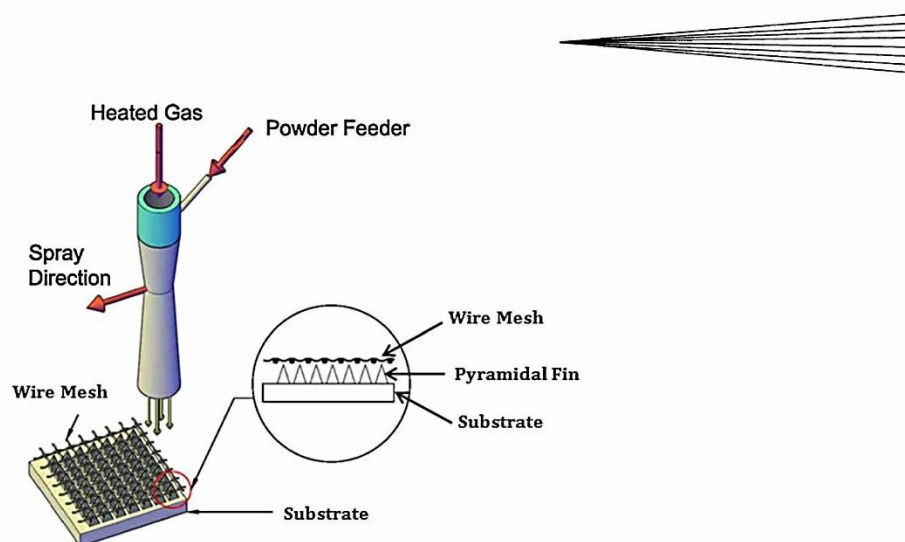
**Fig. 2** Typical cooling curve of temperature as a function of distance



**Fig. 3** SEM images of (a) As-received pure aluminum feedstock powder; (b) As-received pure nickel feedstock powder; (c) As-received stainless steel 304 feedstock powder

by a D10 value of 12.77 microns, a D50 value of 22.34 microns, and a D90 value of 40.44 microns. The powder presents a spherical geometry, as seen in Fig. 3(b).

The third feedstock powder used was a grade 304 stainless steel (Atlantic Equipment Engineers, Upper Saddle River, NJ, USA). The powder size is characterized



**Fig. 4** Schematic of the manufacturing technique utilized to produce pyramidal fin arrays (Ref 17)

**Table 1** Fin array spray parameters

| Powder type         | Nozzle inlet pressure, MPa | Nozzle inlet gas temperature, °C | Powder preheating temperature, °C | Standoff distance, mm |
|---------------------|----------------------------|----------------------------------|-----------------------------------|-----------------------|
| Pure aluminum       | 1.7                        | 350                              | N/A                               | 15                    |
| Pure nickel         | 2.8                        | 400                              | 300                               | 15                    |
| Stainless steel 304 | 3.1                        | 475                              | 300                               | 15                    |

by a D10 value of 13.50 microns, a D50 value of 29.61 microns, and a D90 value of 51.66 microns. The powder presents an elongated, non-spherical geometry, as seen in Fig. 3(c).

## 2.2 CGDS Fin Production Technique

The manufacturing method used to create the square-based pyramidal pin fin arrays using CGDS is illustrated in Fig. 4. It consists of using plain woven, steel wire mesh (McMaster-Carr, Aurora, OH, USA) to mask parts of the substrate from deposition of the particles accelerated by the cold spray process. This allows the user to build features of various shapes and dimensions on the substrate. The operating principles of CGDS result in preferential build-up of material in the center of the mask openings, creating features with reducing cross sections along the height such as pyramids (Ref 12). The CGDS deposition parameters are selected to ensure that the features closely mimic the reversed geometry of the chosen wire mesh, with peaks aligning with the holes of the screened area. Additional details on the production of pyramidal pin fins can be found elsewhere (Ref 12, 13). The wire mesh mask chosen for this study has 12 wires per inch with a wire diameter of 0.035" (0.89 mm) in both the longitudinal and transverse directions.

An SST-EP CGDS system (CenterLine (Windsor) Limited, Windsor, ON, Canada) was used to produce

the pyramidal fin arrays. A polymer nozzle was used to spray the pure aluminum powder, while a tungsten-carbide nozzle was used to deposit both the nickel and the stainless steel powders. Both nozzles have a 2.0-mm throat diameter and a 6.4-mm exit diameter. The spray parameters used to deposit the fins for each feedstock powder are presented in Table 1. The fins were deposited on 5.1 cm by 5.1 cm Al6061-T6 aluminum alloy substrates which were cleaned and degreased using acetone after being cut to size. The fins were sprayed taller than the desired height, and were subsequently ground down to size to ensure production consistency, using the procedure described in Dupuis et al. (Ref 13).

Samples have been produced with three different fin materials as well as a single fin material on a single fin array to demonstrate the possibility and the viability of creating this type array by additive manufacturing using cold spray. For the multi-material sample, since the sample is two inches long with 12 fins per inch, the fin array was divided into three sections of eight fins each. The stainless steel fins were deposited on the leading edge of the sample, while the aluminum was deposited on the trailing edge of the sample, with the nickel section separating the other two sections (in the order of their maximum service temperature). This does not represent the optimal ratio but rather serves as a proof of concept for this type of application.



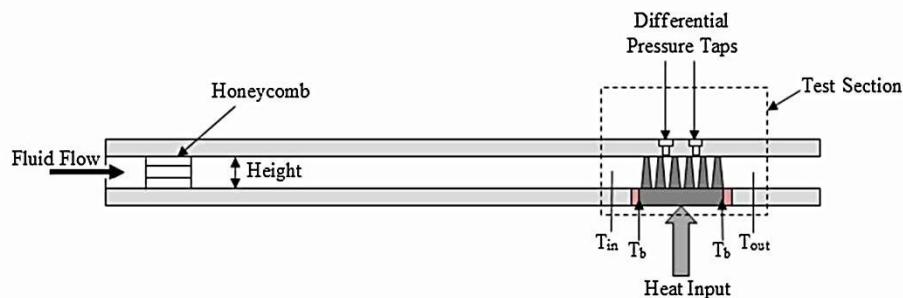


Fig. 5 Heat transfer fixture utilized to evaluate the fin arrays heat transfer performance

### 3. Experimental Procedure

#### 3.1 Sample Surface and Fin Characterization

Standard metallographic procedures were used to obtain cross sections of the sprayed samples. Control specimens were sprayed to obtain a standard for comparison and to establish a porosity level baseline for the different fin materials. Image analysis was performed using Clemex Vision Lite (Clemex Technologies Inc., Longueuil, QC, Canada) image analysis software. The porosity values are reported using the mean of 10 measurements for each dataset.

The fin array geometric variables such as base dimensions and height were measured using a depth of field microscope (Keyence model VHX-2000, Mississauga, ON, Canada) which allows building three-dimensional images of the given samples. From these dimensions, the fin base angle, the hydraulic diameter, and the total heat transfer area were computed.

The heat transfer area surface roughness was also characterized using the same depth of field microscope in a two-step approach. The first step was to obtain the roughness profile of the heat transfer passage base (between the fins), using the conventional method of depth of field microscopy. In the second step, the roughness of the fins surface was obtained by tilting the microscope to the angle matching the base angle of the pyramid to obtain a perpendicular view of the fin surface. A two-dimensional correction algorithm was used to linearly correct the slope of the sample in a plane, and the roughness data were generated directly from the input image file. The total roughness was then determined to be equal to an area weighted average of the roughness of the entire flow passage.

#### 3.2 CGDS Coating Thermal Conductivity Determination

A laser flash diffusivity testing apparatus (DLF-1300, TA Instruments, New Castle, DE, USA) was used to determine the room temperature thermal diffusivity and the specific heat capacity of 12.7-mm-diameter samples produced by cold spray at room temperature. The thermal

conductivity of the sprayed samples was determined using the definition of the thermal diffusivity:

$$\alpha = \frac{k}{\rho_m \cdot C_{pm}}, \quad (\text{Eq 1})$$

where  $\alpha$  is the thermal diffusivity of the material,  $k$  is its thermal conductivity,  $\rho_m$  is its density, and  $C_{pm}$  is its specific heat capacity. The samples were weighed, and the volume was measured by water volume displacement to obtain the coating density.

#### 3.3 Fin Thermal Performance Testing

**3.3.1 Apparatus and Test Procedure.** Performance of the various fin arrays produced was assessed using an in-house heat transfer and pressure drop test apparatus. A schematic of the apparatus is shown in Fig. 5. The main components of this system are the air supply, the test section, the heating pad, and the data acquisition system. The air is supplied by a compressed air line equipped with a filter and dehumidifier. The air inlet flow rate is controlled by a needle valve and is measured with a flow meter. This air is smoothly expanded into a horizontal rectangular channel with a diffuser and a honeycomb flow straightener. An unobstructed 457-mm-long rectangular channel ensures that the flow is fully developed before reaching the test section. The test sample is subjected to a constant heat flux provided by a strip heater.

A set of differential pressure measurement taps is spaced 12.7 mm upstream and downstream from the middle of the test section along the centerline of the apparatus to determine the fin array friction losses. The fin array friction losses include contributions from both skin friction and form drag. A thermocouple is located 6.0 mm ahead of the test section and another 6.0 mm after the test section at mid-height to record the local average temperature of the fluid. Four thermocouples are attached to the fin array's base, with two on the side facing the incoming flow, and two on the side facing the exiting flow. On each face, a thermocouple is located on the centerline, while the other thermocouple is located 6.0 mm away from the wall. All the thermocouples used are T-type, butt-bonded, 28 gage thermocouples.



All measurements were performed at steady-state and were repeated until statistical significance was achieved. Heat transfer data samples were taken at 2 Hz over a 30 s interval. For both thermal and hydrodynamic tests, five sets of data samples were recorded for each flow rate. Measurements were taken at air flow rates of 10–70 SLPM, which corresponds to Reynolds numbers varying from 400 to 3000, which are in the laminar and the sub-critical flow regimes. Deviation of the measurements from the mean value was less than 5%, which led to error bars that were smaller than the markers. For this reason, they were not included in the thermal performance graphs of “Results” section. A detailed description of the apparatus’ accuracy and of the instrument’s uncertainties is given in Cormier et al. (Ref 12). Radiative heat transfer losses are neglected as they would account for less than 0.5% of the total heat input (Ref 18, 19).

**3.3.2 Data Reduction.** The thermal performance comparison is done according to the procedure detailed in Sahiti’s work (Ref 20). This method relates the thermal and hydrodynamic data by plotting the thermal conductance as a function of the required pumping power for a unit volume of heat exchanger at different flow rates. This yields performance curves which can be directly compared one to another, with the most efficient fin array having a higher thermal conductance at a given pumping power.

The equivalent thermal conductance per unit volume for the heat exchanger specimen can be calculated using

$$UA = \frac{1}{R_{eq}} = \frac{q}{\Delta T_{lm}}, \quad (\text{Eq 2})$$

where  $R_{eq}$  is the thermal circuit’s equivalent resistance,  $q$  is the total heat input rate to the system, and  $\Delta T_{lm}$  is the log-mean temperature difference between the fluid and the fin array. The thermal conductance is either normalized using the volume of the heat transfer extended surface ( $V$ , yielding  $UA_v$ ) or using the mass of this extended surface ( $M$ , yielding  $UA_M$ ).

The total heat input rate to the system is calculated using

$$q = \dot{m} \cdot (T_{out} - T_{in}) \Delta C_{pf}, \quad (\text{Eq 3})$$

where  $\dot{m}$  is the fluid mass flow rate,  $(T_{out} - T_{in})$  is the fluid temperature difference between the entrance and the exit of the test section, and  $C_{pf}$  is the fluid’s thermal capacitance.

The log-mean temperature difference is calculated with

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}, \quad (\text{Eq 4})$$

where  $\Delta T_1$  and  $\Delta T_2$  are the temperature differences between the fin surface and the flow at the inlet and outlet, respectively.

The thermal resistance of a fin array can be computed with the following relation:

$$R_{eq} = \frac{1}{h \cdot A_t \cdot \eta_o}. \quad (\text{Eq 5})$$

In the previous equation,  $h$  is the average convective coefficient,  $A_t$  is the total heat transfer area of the fin array and the base, while  $\eta_o$  is the total surface efficiency, defined as the actual heat transfer rate over the maximal heat transfer rate that can be transferred if the surface temperature is uniform and equal to the base temperature. This parameter can be calculated as follows:

$$\eta_o = 1 - \frac{A_f}{A_t} (1 - \eta_f), \quad (\text{Eq 6})$$

where  $A_f$  is the finned area and  $\eta_f$  is the individual fin efficiency, defined as the heat transferred by the fins over the maximum possible heat that can be transferred, which can be approximated with

$$\eta_f = \frac{2 I_2(2mH)}{mH I_1(2mH)} \quad (\text{Eq 7})$$

and

$$m = \sqrt{\frac{4h}{kD}}. \quad (\text{Eq 8})$$

In the previous equations adapted from Incropera’s work (Ref 11).  $H$  denotes the height of fin (or length of protrusion of the fin from the base),  $I_1$  and  $I_2$  denote the first- and second-order Bessel functions, respectively, while  $k$  is the thermal conductivity of the fin’s metal. For the particular fin geometry used in this work,  $D$  corresponds to the fin base length ( $B$ ).

The required pumping power input was calculated according to Sahiti’s definition (Ref 21):

$$e = \frac{\dot{V}_f \cdot \Delta P_{fin}}{\eta}. \quad (\text{Eq 9})$$

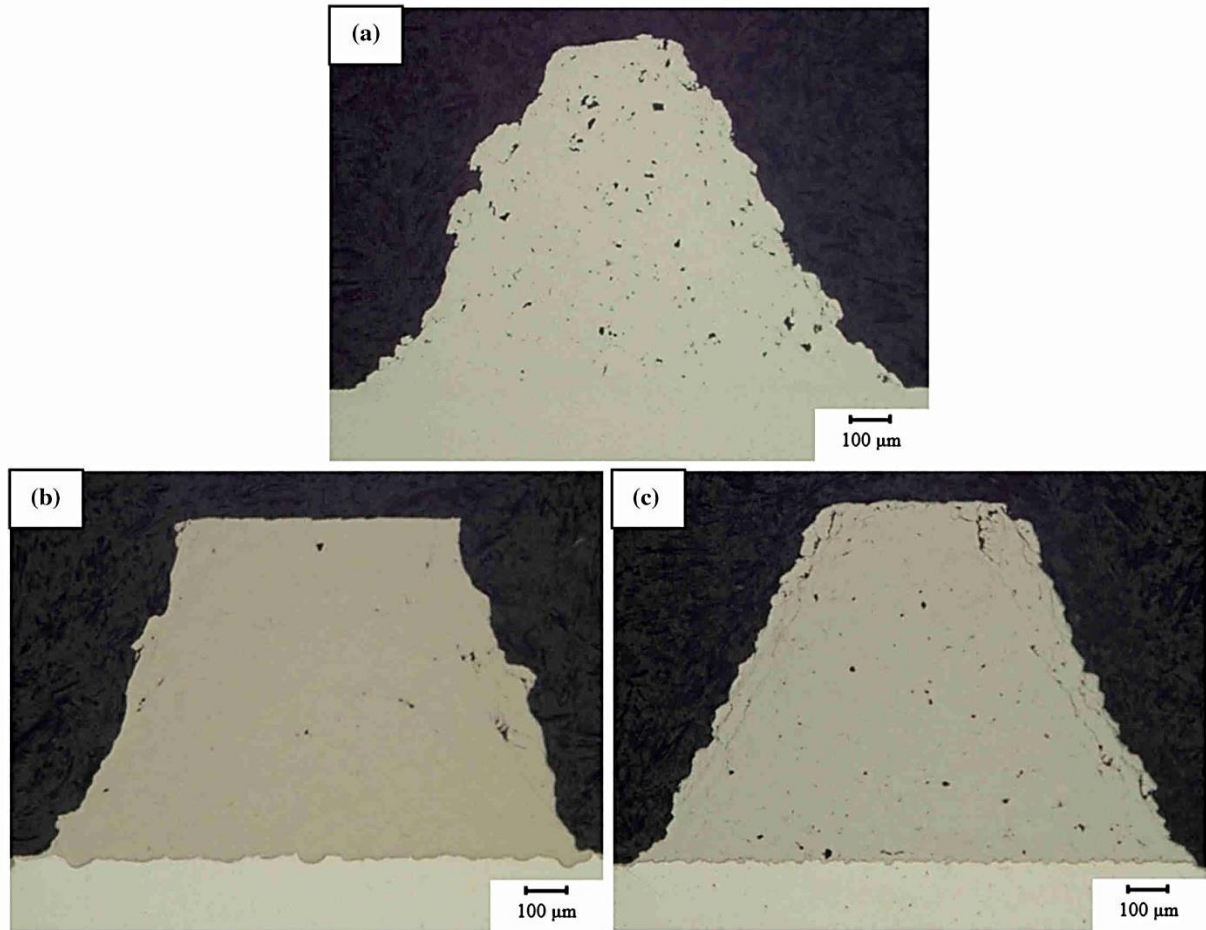
In the previous relation,  $e$  is the pumping power,  $\dot{V}_f$  is the volumetric flow rate of the fluid,  $\Delta P_{fin}$  is the pressure drop measured, and  $\eta$  is the fan efficiency. A fan efficiency of 0.8 was chosen as a reasonable value to perform these calculations (Ref 21). This pumping power is either normalized using the volume of the heat transfer extended surface ( $V$ , yielding  $e_v$ ) or using the mass of this extended surface ( $M$ , yielding  $e_M$ ).

For these experiments, the Reynolds number ( $Re$ ) of the flow is defined as

$$Re_{Dh} = \frac{\rho_f \cdot U \cdot D_h}{\mu}, \quad (\text{Eq 10})$$

where  $\rho_f$  is the fluid’s density,  $U$  is the free-stream axial velocity,  $D_h$  is the hydraulic diameter of the flow channel, and  $\mu$  is the dynamic viscosity of the working fluid.

The average roughness of the heat exchanger samples ( $Ra_A$ ) was determined using an area weighted average of the wetted surfaces’ roughness:



**Fig. 6** Cross section of different type of pyramidal fin produced (a) Pure aluminum; (b) Pure nickel; (c) Stainless steel 304

$$Ra_A = \frac{Ra_f A_f + Ra_b A_b}{A_t}, \quad (\text{Eq 11})$$

where  $Ra_f$  is the roughness value of the fin's surface,  $Ra_b$  is the roughness of the unfinned base, and  $A_b$  is the unfinned base area. The roughness of the shroud material is not considered in this equation, as its roughness is negligible when compared to that of either the base or the fin surface.

#### 4. Results

The different types of samples were characterized, with typical cross sections shown in Fig. 6 and the important geometric quantities associated with the different samples presented in Table 2. This figure shows similar shaped fins for the different samples, with some differences in the top length of the fins, attributed to over-grinding of the

specimens during sample preparation, leading to cross sections that are not exactly at the center of the fin depicted in the figure.

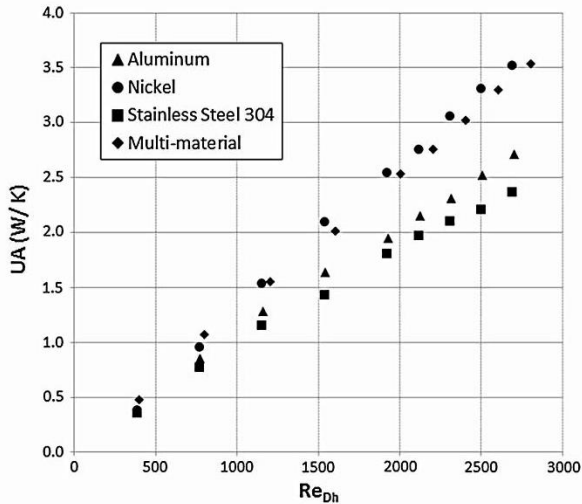
Figure 7 presents the thermal conductance as a function of Reynolds number for the different samples tested. Observing this figure, the data that stand out are the thermal performance of the nickel and multi-material samples, which have a significantly higher thermal conductance than the other samples at a given Reynolds number.

This distribution of results is not expected when looking at the thermal conductivities of the fin array materials: pure aluminum has the highest thermal conductivity, with a theoretical value of 237 W/m K, followed by nickel with an intermediate value of 91 W/m K, and finally stainless steel 304 with the lowest value of 15 W/m K (Ref 11). Solely looking at the thermal conductivity, however, gives an incomplete analysis of the data presented in Fig. 7, as it would then be expected that the aluminum sample would outperform the nickel sample. The comparable



**Table 2** Important variables of pyramidal fin array

| Sample name         | Fin material        | Fin height (H), $\mu\text{m}$ | Base (B), $\mu\text{m}$ | Total heat transfer area, $\text{mm}^2$ | Hydraulic diameter, mm | Base angle, $^\circ$ | Porosity, %   |
|---------------------|---------------------|-------------------------------|-------------------------|-----------------------------------------|------------------------|----------------------|---------------|
| Aluminum            | Pure aluminum       | $1048 \pm 12$                 | $1401 \pm 30$           | $3775 \pm 72$                           | $1103 \pm 40$          | $62.4 \pm 1.0$       | $2.8 \pm 1.5$ |
| Nickel              | Pure nickel         | $974 \pm 35$                  | $1352 \pm 28$           | $3612 \pm 66$                           | $1045 \pm 54$          | $62.5 \pm 1.7$       | $0.8 \pm 0.6$ |
| Stainless steel 304 | Stainless steel 304 | $1019 \pm 42$                 | $1351 \pm 69$           | $3687 \pm 160$                          | $1081 \pm 92$          | $62.1 \pm 1.6$       | $1.1 \pm 0.6$ |
| Multi-material      | Al/Ni/SS            | $1008 \pm 52$                 | $1372 \pm 64$           | $3903 \pm 164$                          | $1062 \pm 92$          | $64.8 \pm 2.6$       | $1.5 \pm 1.3$ |


**Fig. 7** Thermal conductance comparison between aluminum (triangle), nickel (circle), stainless steel 304 (square), and multi-material (diamond)

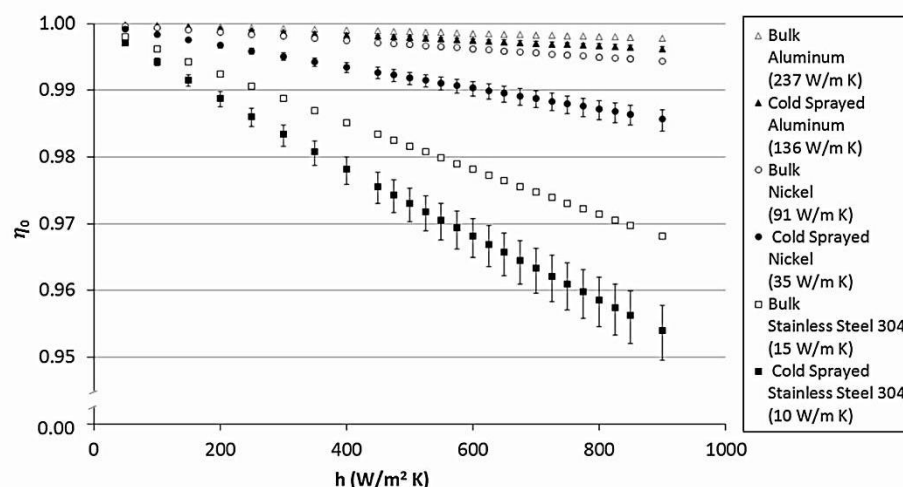
performance of these samples can then be partially explained by looking at the definition of the equivalent thermal resistance for fin arrays. Since the fin arrays have very similar geometries and are compared at the same flow conditions, it follows that the convective coefficient and the total heat transfer area are similar between samples tested at the same flow conditions. The only parameter that can vary significantly is then the total surface efficiency, which is itself dependent on the fin efficiency. Considering Eq 6, 7, and 8, the effect of the fin material, through its thermal conductivity, can be seen. For convenience, the results of Eq 6 are plotted as a function of the convective coefficient for different fin material conductivity values and are presented in Fig. 8. It should be noted that the fin efficiency is high due to their dimensions (low height to cross-section area ratio). This chart demonstrates that there is a negligible performance loss when going from bulk aluminum to bulk nickel ( $237\text{--}91\text{ W/m K}$ ) with less than 0.4% for convective coefficient values of  $800\text{ W/m}^2\text{ K}$ , but there is a more significant drop in the efficiency when using stainless steel ( $15\text{ W/m K}$ , a 3.0% reduction for convective coefficient values of  $800\text{ W/m}^2\text{ K}$ ). However, the order of thermal conductance is not fully explained since the nickel fin array has a better thermal performance than the aluminum sample; while according

to the previous explanation, their performance should be similar. Nevertheless, the lower thermal performance of stainless steel fin arrays is explained.

It has been reported that the properties of coatings produced by CGDS do not match those of the bulk material (Ref 22–25). Indeed, the mechanical, electrical, and thermal properties of a coating produced by CGDS may be widely different from the bulk, annealed material due to the large amount of plastic deformation that occurs during deposition (Ref 22). For the thermal conductivity in particular, the observed value is usually much lower than that of the bulk material due to the amount of dislocations created by the cold spray process, which shortens the mean free path for electron movement, but also due to other contributing factors such as the quality of the inter-particle bonding and the presence of surface oxides in the powder, which become imbedded in the coating (Ref 22–25). The difference between the fin efficiencies for the different materials becomes even more pronounced when the real values of thermal conductivity (presented in Table 3) are used instead of values typical of bulk materials, as seen in Fig. 8. To this end, following this discussion, it would be expected that the thermal conductance curves of the different samples would be in the following order, from most performant to least performant: aluminum, followed very closely by nickel, multi-material, and finally stainless steel.

The difference between the expected order and the order of the measured thermal conductances can then be explained by looking at the hydrodynamic performance of the samples. Figure 9 presents the measured pressure losses of the fin array samples as a function of the Reynolds number. It can be observed that the nickel and multi-material samples have the highest pressure loss, followed by the stainless steel sample, with the aluminum sample having the lowest hydrodynamic losses. This trend is likely attributable to the roughness of the flow passage (the side of the fins and the substrate area between the fins). It is well known that surface roughness increases heat transfer performance and contributes significantly to pressure losses (Ref 26–28), especially in small hydraulic diameter heat exchangers (Ref 29).

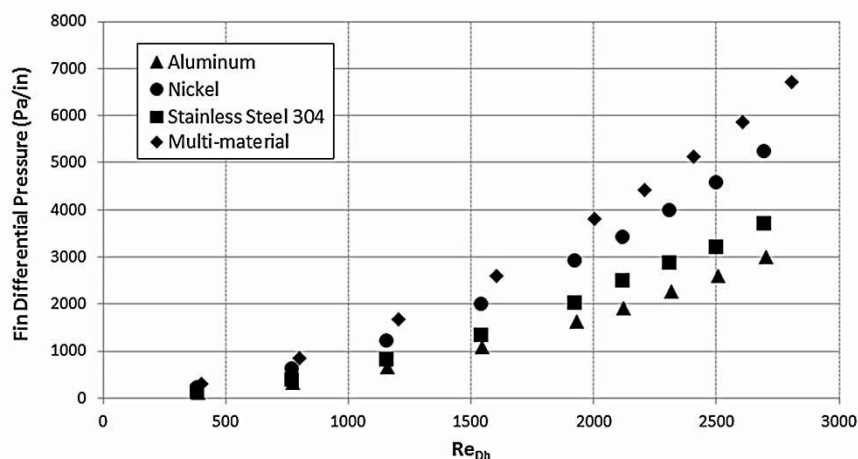
A representative image of the fin's side surface is shown in Fig. 10, which results in the roughness values ( $Ra_f$ ) presented in Fig. 11. It is possible to observe that the nickel fin's surface is the roughest, followed by the aluminum fins and finally by the stainless steel fins. The roughness of the fin surface, on its own, cannot explain the increase in fin arrays differential pressure observed in Fig. 10 as well as the rise in the thermal conductance



**Fig. 8** Total surface fin efficiency, for the shape of the fins in the present study, in function of the convective heat transfer coefficient for different thermal conductivities

**Table 3** Physical and thermal properties of the CGDS produced coatings as well as bulk thermal conductivity for comparison purpose

| Material            | Thermal diffusivity, $\text{cm}^2/\text{s}$ | Thermal capacitance, $\text{J/kg K}$ | Density, $\text{kg/m}^3$ | Thermal conductivity, $\text{W/m K}$ | Bulk thermal conductivity, $\text{W/m K}$ |
|---------------------|---------------------------------------------|--------------------------------------|--------------------------|--------------------------------------|-------------------------------------------|
| Pure aluminum       | $0.588 \pm 0.012$                           | $911 \pm 46$                         | $2535 \pm 254$           | $136 \pm 15$                         | 237                                       |
| Pure nickel         | $0.126 \pm 0.003$                           | $341 \pm 17$                         | $8137 \pm 814$           | $35 \pm 4$                           | 91                                        |
| Stainless steel 304 | $0.017 \pm 0.001$                           | $882 \pm 44$                         | $6988 \pm 699$           | $10 \pm 1$                           | 15                                        |

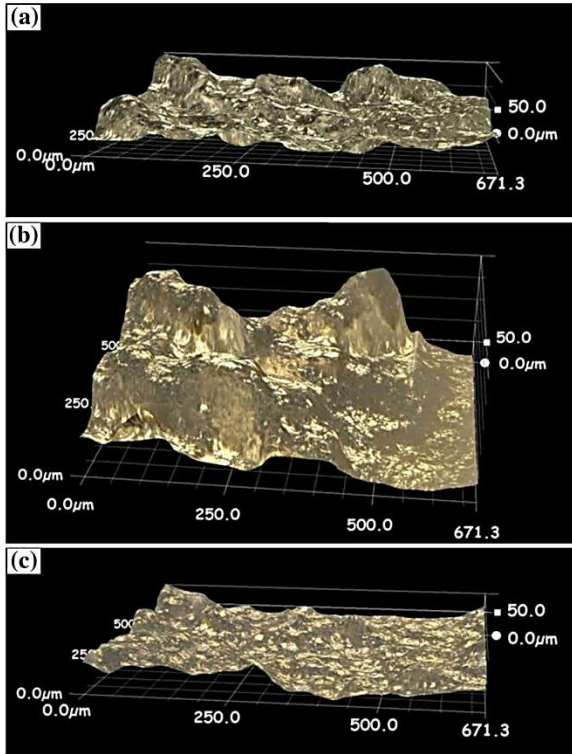


**Fig. 9** Fin differential pressure per inch comparison between aluminum (triangle), nickel (circle), stainless steel 304 (square), and multi-material (diamond)

depicted in Fig. 7 since the stainless steel sample has the lowest fin roughness, while it has a higher differential fin pressure than the aluminum. By observing the substrate area between each fin, it is possible to notice the presence of a thin layer of attached particles for the nickel and the stainless steel fin arrays (Fig. 12b and c), whereas the

aluminum does not have deposition between the fins (Fig. 12a). The arithmetic mean roughness ( $Ra_b$ ) for the base area is shown in Fig. 11 and results in a higher roughness for the stainless steel sample followed by the nickel sample and the aluminum sample, most likely attributable to the stronger deformation of the aluminum



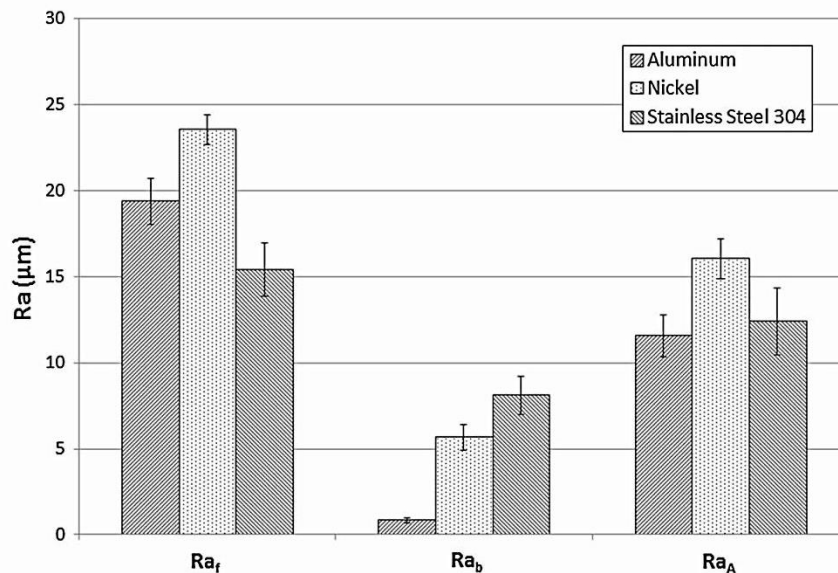


**Fig. 10** Three-dimensional image of the fin's side surface for (a) Aluminum; (b) Nickel; (c) Stainless Steel 304

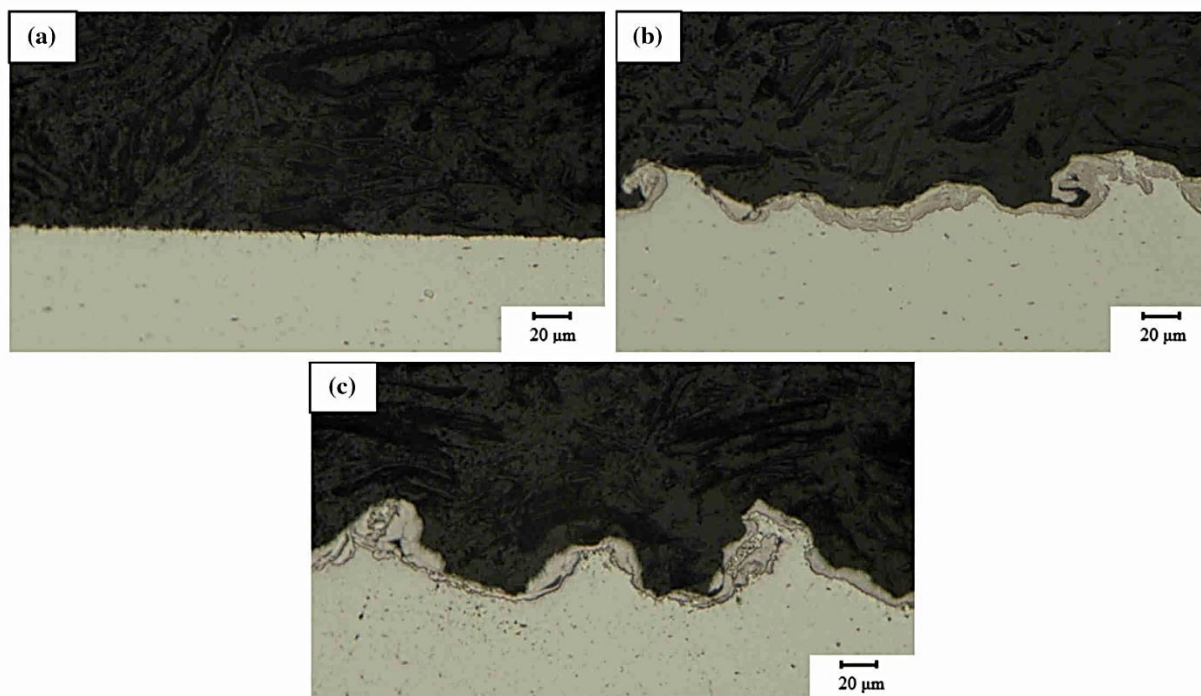
substrate by the heavier and harder stainless steel and nickel particles than by the softer, lighter aluminum particles. In the case of the stainless steel and nickel samples, the substrate deformation is evident, while for the aluminum, only small dimples (approximately 4  $\mu\text{m}$  in depth) are observed. In order to understand the performance curves, a weighted average of each component of the surface roughness (fin surface and base roughness) is computed according to Eq 11. This weighted average ( $Ra_A$ ) demonstrates that the nickel fin array has the highest average roughness, followed by the stainless steel and the aluminum fin arrays (Fig. 10), which correlates with the hydrodynamic performance trends observed in Fig. 9.

Using the hydrodynamic performance data and following the discussion regarding the thermal conductivity and fin efficiency, the differences in thermal performance of the samples shown in Fig. 7 can be explained. The nickel sample has the highest thermal conductance due to the increased convective coefficient (from the flow disruptions caused by to the surface roughness), despite the negligible decrease in fin efficiency associated with the use of nickel compared to aluminum. The multi-material sample's performance is similar to that of the nickel sample for the same reason, while the aluminum sample outperforms the stainless steel fin array due to its higher fin efficiency (despite the fact that they have a similar passage roughness).

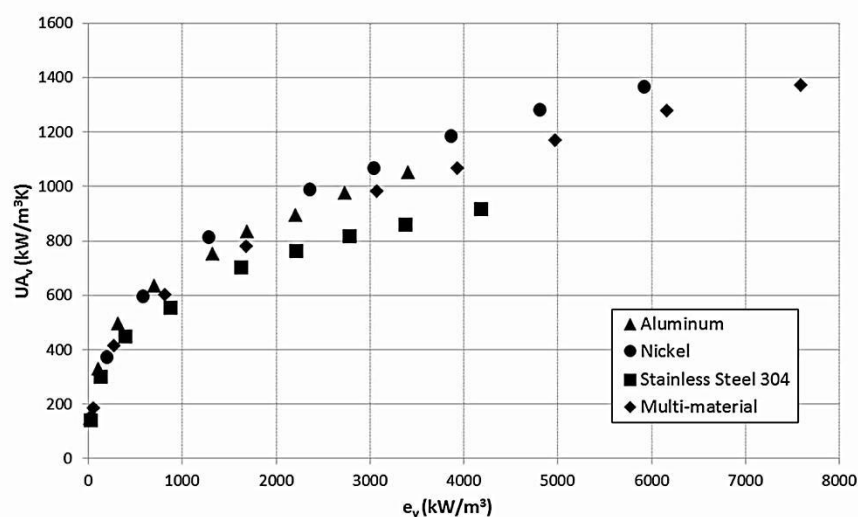
The volumetric heat transfer surface efficiency curves are presented in Fig. 13. One first notes that the volumetric efficiencies of the aluminum, the nickel, and the multi-material samples are comparable, while the perfor-



**Fig. 11** Surface roughness arithmetic average for the aluminum, stainless steel 304 and nickel for (a) fin's surface; (b) Unfinned base area; (c) Total roughness



**Fig. 12** Typical cross section of the unfinned surface (a) Aluminum; (b) Nickel; (c) Stainless steel 304

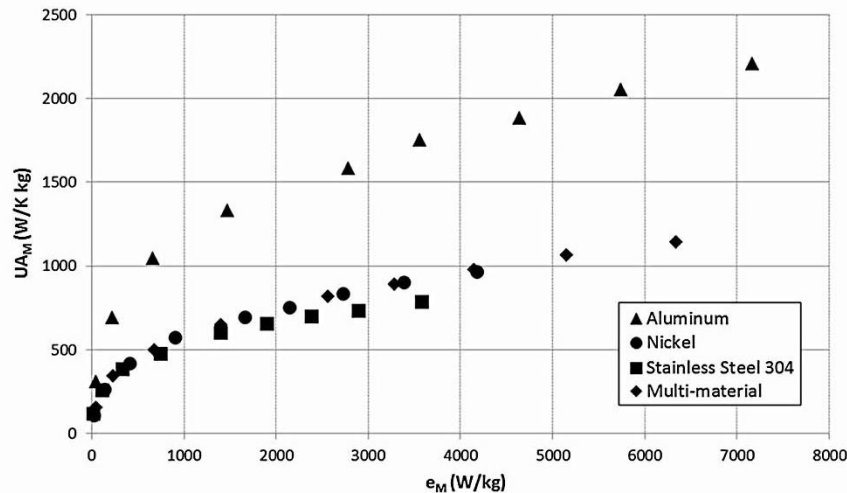


**Fig. 13** Volume-based heat transfer performance comparison between aluminum (triangle), nickel (circle), stainless steel 304 (square), and Multi-material (diamond)

mance of the stainless steel sample is lower. This trend can be expected from Fig. 7 and 9, as the increased thermal performances of the multi-material and nickel samples are offset by the higher hydrodynamic losses encountered in those fin arrays. The information presented in Fig. 13,

however, indicates that using a heat exchanger made of several materials with different service temperatures can yield better heat transfer performance when compared to one that uses a single-material with a high service temperature (in this example, stainless steel). This becomes





**Fig. 14** Mass-based heat transfer performance comparison between aluminum (triangle), nickel (circle), stainless steel 304 (square), and multi-material (diamond)

**Table 4** Summary chart of the different performance metrics between the multi-material and the stainless steel samples

| Performance metric                                              | Comparison point    | Multi-material interpolated metric value | Stainless steel interpolated metric value | Metric increase of the multi-material sample, % |
|-----------------------------------------------------------------|---------------------|------------------------------------------|-------------------------------------------|-------------------------------------------------|
| Figure 7: thermal conductance (UA) (W/K)                        | $Re_{Dh} = 1500$    | 1.9                                      | 1.4                                       | 35                                              |
|                                                                 | $Re_{Dh} = 2500$    | 3.2                                      | 2.2                                       | 42                                              |
| Figure 9: differential pressure ( $\Delta P_{fin}$ ) (Pa/in)    | $Re_{Dh} = 1500$    | 2362                                     | 1301                                      | 82                                              |
|                                                                 | $Re_{Dh} = 2500$    | 5475                                     | 3220                                      | 70                                              |
| Figure 13: volume-based performance ( $UA_v$ ) [ $kW/(m^3 K)$ ] | $e_v = 1500 kW/m^3$ | 744                                      | 676                                       | 10                                              |
|                                                                 | $e_v = 3000 kW/m^3$ | 972                                      | 831                                       | 17                                              |
| Figure 14: mass-based performance ( $UA_M$ ) [ $W/(kg K)$ ]     | $e_M = 1500 W/kg$   | 666                                      | 611                                       | 9                                               |
|                                                                 | $e_M = 3000 W/kg$   | 864                                      | 744                                       | 16                                              |

even more apparent when the comparison is made between heat exchangers on a unit mass basis, such as the results presented in Fig. 14. In this case, aluminum becomes much more efficient than the other two materials due to its low density. Nickel, having the highest density, has its performance decreased greatly, in comparison, but is still slightly superior to that of stainless steel. The multi-material sample once again has a similar performance to that of the nickel sample. This also confirms the viability of using a multi-material heat exchanger due to the increased performance when compared to typical single-material exchangers made of a high service temperature alloy such as stainless steel. When the service temperature does not restrict the use of low service temperature materials such as aluminum, it can be very advantageous to use this type of low-density material to increase the mass-specific thermal performance.

The tests for the multi-material sample were also performed in both orientations, with the stainless steel section closer to the inlet and then closer to the outlet, respectively. No discernable difference between the performances of the two configurations has been observed, leading to the conclusion

that the efficiency of this sample is mostly attributable to the thermal and hydrodynamic performance of its nickel section. This conclusion, of course, is only applicable to the particular configuration of the multi-material fin array detailed in this work. For other configurations, the performance of the hybrid fin array will reflect that of whichever section most influences the flow physics in that particular case.

## 5. Summary and Conclusion

Near-net-shaped pyramidal fin arrays of various materials were manufactured using the CGDS process as well as a streamwise anisotropic fin array. Fin characterization was performed using conventional microscopy, as well as depth of field microscopy. Fin array performance was determined using a steady-state forced air convection apparatus. The heat transfer performance analysis has revealed that the cold spray process used to produce the fin arrays has a considerable influence on the results by modifying the fins and substrate surface roughness. Results obtained show a similar efficiency based on



volume for the nickel, the aluminum, and the multi-material samples, while the stainless steel is less efficient. Using a unit mass criterion, the aluminum fin array outperforms the three other fin arrays. The proof of concept of using cold spray as an additive manufacturing process to produce a streamwise anisotropic fin array instead of one made from a single, high service temperature material has been performed. The performance metrics of the multi-material sample compared to the stainless steel sample are summarized in Table 4, showing the benefits of the anisotropic approach.

## Acknowledgments

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## References

1. B.A. Jasperson, Y. Jeon, K.T. Turner, F.E. Pfefferkorn, and W. Qu, Comparison of Micro-Pin-Fin and Microchannel Heat Sinks Considering Thermal-Hydraulic Performance and Manufacturability, *IEEE Trans. Compon. Packag. Technol.*, 2001, **33**(1), p 148-160
2. S.M. Kaplan and F. Sissine, Smart Grid: Modernizing Electric Power Transmission and Distribution; Energy Independence, Storage and Security; Energy Independence and Security Act... and Resiliency; Integra, TheCapitol.Net Inc., 2009
3. D. Bohn, "Micro Gas Turbine and Fuel Cell—A Hybrid Energy Conversion System with High Potential." NATO Research and Technology Organisation, 2005
4. I.A. Popov, Y.F. Gortyshov, and V.V. Olimpiev, Industrial Applications of Heat Transfer Enhancement: The Modern State of the Problem (A Review), *Therm. Eng.*, 2012, **59**(1), p 1-12
5. M. Iyengar and A. Bar-Cohen, Design for Manufacturability of SISE Parallel Plate Forced Convection Heat Sinks, *IEEE Trans. Compon. Packag. Technol.*, 2001, **24**(2), p 150-158
6. M. Geiger, M. Kleiner, R. Eckstein, N. Tiesler, and U. Engel, Microforming, *CIRP Ann. Manuf. Technol.*, 2001, **50**(2), p 445-462
7. Y. Jeon and F.E. Pfefferkorn, Effect of Laser Preheating Workpiece on Micro end Milling of Metals, *J. Manuf. Sci. Eng.*, 2008, **130**(1), p 1-9
8. K.A. Moores and Y. Joshi, Effect of Tip Clearance on the Thermal and Hydrodynamic Performance of a Shrouded Pin Fin Array, *J. Heat Transfer*, 2003, **125**, p 999-1006
9. K. Thulukkanam, *Heat Exchanger Design Handbook*, 2nd ed., CRC Press, Boca Raton, 2013
10. W.M. Kays and A.L. London, *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York, 1984
11. F.P. Incropera, D.P. DeWitt, T.L. Bergman, and A.S. Lavine, *Fundamentals of Heat and Mass Transfer*, 6th ed., Wiley, New York, 2006
12. Y. Cormier, P. Dupuis, B. Jodoin, and A. Corbeil, Net Shape Fins for Compact Heat Exchanger Produced by Cold Spray, *J. Therm. Spray Technol.*, 2013, **22**(7), p 1210-1221
13. P. Dupuis, Y. Cormier, A. Farjam, B. Jodoin, and A. Corbeil, Performance Evaluation of Near-Net Pyramidal Shaped Fin Arrays Manufactured by Cold Spray, *Int. J. Heat Mass Transf.*, 2014, **69**, p 34-43
14. H. Assadi, F. Gärtner, T. Stoltenhoff, and H. Kreye, Bonding Mechanism in Cold Gas Spraying, *Acta Mater.*, 2003, **51**, p 4379-4394
15. M. Grujicic, C.L. Zhao, W.S. DeRosset, and D. Helfrich, Adiabatic Shear Instability Based Mechanism for Particles/Substrate Bonding in the Cold-Gas Dynamic-Spray Process, *Mater. Des.*, 2004, **25**, p 681-688
16. Y. Cormier, P. Dupuis, A. Farjam, A. Corbeil, and B. Jodoin, Additive Manufacturing of Pyramidal Pin Fins: Height and Fin Density Effects under Forced Convection, *Int. J. Heat Mass Transf.*, 2014, **75**, p 235-244
17. Y. Cormier, P. Dupuis, B. Jodoin, and A. Corbeil, Mechanical Properties of Cold Gas Dynamic-Sprayed Near-Net-Shaped Fin Arrays, *J. Therm. Spray Technol.*, 2015, **24**(3), p 476-488
18. S. Naik, S.D. Probert, and M.J. Shilston, Forced Convective Steady-state Heat Transfers from Shrouded Vertical Fin Arrays Aligned Parallel to an Undisturbed Air-stream, *Appl. Energy*, 1987, **26**, p 137-158
19. J.J. Hwang and T.M. Liou, Heat Transfer and Friction in a Low-aspect-ratio Rectangular Channel with Staggered Perforated Ribs on Two Opposite Walls, *ASME J. Heat Transfer*, 1995, **117**(11), p 843-850
20. N. Sahiti, F. Durst, and A. Dewan, Heat Transfer Enhancement by Pin Elements, *Int. J. Heat Mass Transf.*, 2005, **48**, p 4738-4747
21. N. Sahiti, F. Durst, and A. Dewan, Strategy for Selection of Elements for Heat Transfer Enhancement, *Int. J. Heat Mass Transf.*, 2006, **49**, p 3392-3400
22. V.K. Champagne, *The Cold Spray Materials Deposition Process: Fundamentals and Application*, 1st ed., CRC Press, Boca Raton, 2007
23. D. Seo, K. Ogawa, K. Sakaguchi, N. Miyamoto, and Y. Tsuzuki, Parameter Study Influencing Thermal Conductivity of Annealed Pure Copper Coatings Deposited by Selective Cold Spray, *Surf. Coat. Technol.*, 2012, **206**, p 2316-2324
24. S. Kikuchi, S. Yoshino, M. Yamada, M. Fukumoto, and K. Okamoto, Microstructures and Thermal Properties Cold-Sprayed Cu-Cr Composite Coatings, *J. Therm. Spray Technol.*, 2013, **22**(6), p 926-931
25. E. Litovsky, J.I. Kleiman, M. Shagalov, and R.B. Heimann, Measurement of the Thermal Conductivity of Cold Gas Dynamically Sprayed Alumina-reinforced Aluminum Coatings between  $-150^{\circ}\text{C}$  and  $+200^{\circ}\text{C}$ . New Test Method and Experimental Results, *Surf. Coat. Technol.*, 2014, **242**, p 141-145
26. L. Xiao-wei, M. Ji-an, and L. Zhi-xin, Roughness Enhanced Mechanism for Turbulent Convective Heat Transfer, *Int. J. Heat Mass Transf.*, 2011, **54**, p 1775-1781
27. G. Croce, P. D'agaro, and C. Nonino, Three-dimensional Roughness Effect on Microchannel Heat Transfer and Pressure Drop, *Int. J. Heat Mass Transf.*, 2007, **50**, p 5249-5259
28. L. Ventola, F. Robotti, M. Dialameh, F. Calignano, D. Manfredi, E. Chiavazzo, and P. Asinari, Rough Surfaces with Enhanced Heat Transfer for Electronics Cooling by Direct Metal Laser Sintering, *Int. J. Heat Mass Transf.*, 2014, **75**, p 58-74
29. S. Kandlikar, S. Joshi, and S. Tian, Effect of Surface Roughness on Heat Transfer and Fluid Flow Characteristics at Low Reynolds Numbers in Small Diameter Tubes, *Heat Transfer Eng.*, 2010, **24**(3), p 4-16

## 5.6 Paper #6: Failure Modes and Numerical Simulations

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The following paper focuses on the failure modes present for the cold sprayed pyramidal fin arrays. Experiments were conducted and different failure modes under shear conditions were identified. It was found that the different shear strengths measured seem to correlate with the equivalent plastic strain (PEEQ) at the fracture surface as determined with a finite element analysis model. Also, a single deposited layer on the substrate between the fins was observed for two types of powder. This observation was explained in this paper by the detrimental effect of angular momentum induced in the second layer of particles at the impact when they are deposited under the wire mask. This paper addresses the objective of the numerical simulation of the cold spray impact process presented in section 3.7.





# Finite Element Analysis and Failure Mode Characterization of Pyramidal Fin Arrays Produced by Masked Cold Gas Dynamic Spray

Yannick Cormier, Philippe Dupuis, Bertrand Jodoin, and Abbas Ghaei

(Submitted July 16, 2015; in revised form September 10, 2015)

This work evaluates the shear strength of pyramidal fin arrays made from various feedstock materials (cylindrical aluminum, spherical nickel, and cylindrical stainless steel 304 powders) deposited on an Al6061-T6 substrate. Higher shear strength was measured for the nickel fin array followed by the stainless steel 304 and the aluminum arrays. Different failure modes were observed by inspecting the fracture surfaces under Scanning Electron Microscope. Deposition between the cold sprayed nickel and stainless fins was detected whereas dimples were noticed on the substrate between the fins when aluminum is used as the feedstock material. A numerical simulation of normal and angled impacts using the high strain rate Preston-Tonks-Wallace model was carried out in order to have a better understanding of the experimental results. The equivalent plastic strain (PEEQ) obtained from the finite element analysis at normal impact correlates with the different shear strengths measured experimentally. Furthermore, even if a higher PEEQ was observed for angled impacts compared to its normal collision counterpart, it is suggested that the particles may not bond because of the rotational restitution momentum caused by the tangential friction generated during angled impacts. This rotational restitution momentum was not detected for particle impacts normal to the substrate surface.

**Keywords** additive manufacturing, cold spray, finite element analysis, pin fin array, shear strength

## 1. Introduction

Since its initial development in the late 1980s at the Institute of Theoretical and Applied Mechanics of the Siberian Branch of the Russian Academy of Sciences, Cold Gas Dynamic Spray, or Cold Spray, has been the subject of multiple studies aiming at elucidating the bonding mechanisms involved in this process. Many experimental studies have demonstrated that the bonding mechanisms in Cold Spray are the result of localized adiabatic shear instabilities creating intimate, oxide-free material contact which allows metallurgical bonding, combined with mechanical interlocking of particles (Ref 1-5). These studies have typically supported their findings with the observation of individual particle impact on the substrate, using various observation techniques. There are

many difficulties involved in the experimental characterization of the bonding mechanisms during the adhesion process itself, as the time resolution required for these measurements is short (on the order of a few nanoseconds). These difficulties are not encountered when using numerical models, however, which makes this type of investigation an attractive option to obtain insight into the physics involved in the bonding process. Assadi et al. (Ref 1) and Grujicic et al. (Ref 6) used such modeling techniques to demonstrate that the adiabatic shear instability and its resultant plastic flow localization contribute significantly to the deformation and adhesion of particles onto the substrate. Since these landmark publications, several authors have developed similar numerical models to further study the deformation and adhesion mechanisms in Cold Spray in more details (Ref 7-12), with varying degrees of success when it comes to the predictive capabilities of such models.

The success of a numerical model's predictions heavily lies with the choice of mechanical properties and deformation behavior model used for the materials considered. The mechanical properties of several materials for large deformation rates have been determined experimentally under various loading conditions, including for very high strain rate applications, where a sharp increase in the flow stress was noted for strain rates above  $10^5 \text{ s}^{-1}$  (Ref 13).

Most numerical studies dealing with Cold Spray have used the empirical Johnson-Cook (JC) model of material deformation, most likely due to its success in other high strain rate applications and since it is a rather simple

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model that is included in many commercially available finite element packages. It was noted by several authors (Ref 11, 12, 14-17) that this model fails to accurately predict material deformation at strain rates higher than  $10^5 \text{ s}^{-1}$ , due to the sharp increase in flow stress at strain rates typically found in Cold Spray. These are often greater than  $10^7 \text{ s}^{-1}$  (Ref 11, 14, 18), which may explain the limitations of the JC model in successfully describing the particle-substrate behavior and predicting the deformation and adhesion properties. It has been demonstrated by Rahmati and Ghaei (Ref 14) that the particle impacts are more accurately predicted when using the Preston-Tonks-Wallace (PTW) model than by using other models such as the Johnson-Cook, Modified Zerilli-Armstrong, Voyiadjis-Abad, Modified Khan-Huang-Liang, and Gao-Zhang model. The PTW description of material behavior is a complex constitutive model that was developed by Preston et al. (Ref 13) to predict the behavior of materials at very high strain rates through its description of the dislocation motion during plastic deformation. As such, this model takes into account the sharp increase of the flow stress at strain rates exceeding  $10^5 \text{ s}^{-1}$ , which may explain its success in obtaining realistic splat geometries and substrate deformations in operating conditions such as those found in the Cold Spray process.

Additive manufacturing using Cold Spray is a topic which is currently being studied by several groups (Ref 19-22), and has obtained a growing amount of attention. Masked Cold Gas Dynamic Spray (MCGDS) has been recently developed as a new additive manufacturing technique to build periodically repetitive structures such as pin fin arrays and has been proven to be a viable alternative in compact heat exchanger applications (Ref 19, 23, 24). The performance of pin fin arrays produced using this process has been characterized and found superior to those of conventional plain fin arrays of similar dimensions (Ref 19, 23). It was also demonstrated that using this technology, it is possible to create fin arrays of various materials, as well as hybrid arrays with fin materials tailored to the local fluid temperature encountered in the heat exchanger (Ref 26).

The focus of this work is to characterize the mechanical strength of aluminum, nickel, and stainless steel pyramidal pin fin arrays produced using MCGDS in shear loading conditions. The finite element analysis of particle-substrate and of particle-particle impacts during the deposition through a mask is also performed using a commercial finite element analysis software (ABAQUS) with the addition of PTW constitutive equation model for a more accurate description of the particle impact behavior at high strain rates found in Cold Spray. This was performed to help gain insight into the fundamental physics behind the observed deformation levels and failure mechanisms of the different fin arrays and to explain the trends obtained for the mechanical strengths of the different pin fin array samples. Furthermore, the numerical model is used to understand the deformation mechanics involved in impacts with both tangential and normal velocity components, to predict the final deformation state of the sub-

strate and to determine whether or not particle adhesion is likely to occur.

## 2. Fin Array Production

### 2.1 Feedstock Powders

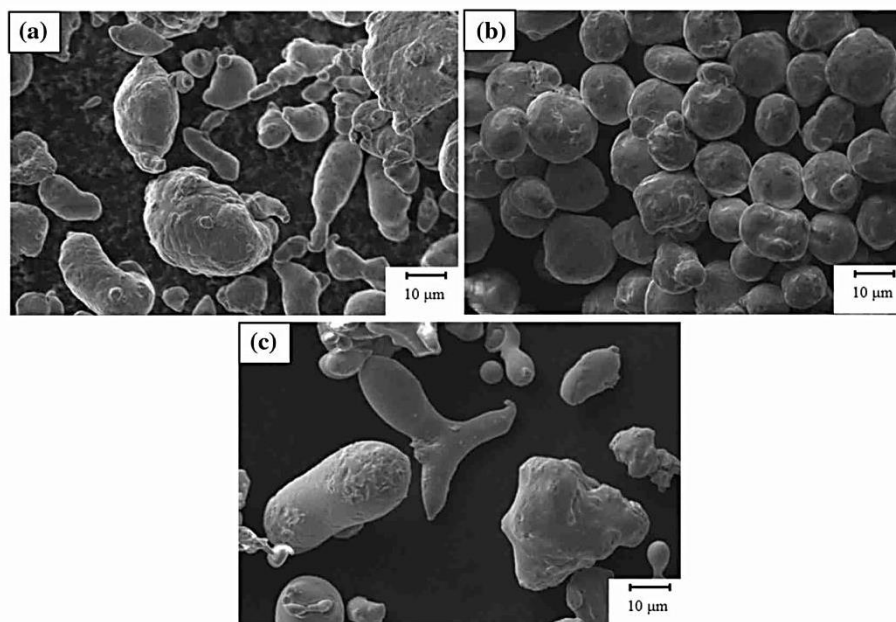
For this study, three types of powder are used: pure aluminum, pure nickel, and stainless steel 304. The pure aluminum (99.8% Al) (CenterLine (Windsor) Limited, Windsor, ON, Canada) presents an elongated geometry, as seen in Fig. 1(a). The powder size distribution was measured by laser diffraction analysis (Microtrac model S3500, Montgomeryville, PA, USA) resulting in a  $D_{10}$  value of 13.29 microns, a  $D_{50}$  value of 25.35  $\mu\text{m}$ , and a  $D_{90}$  value of 49.58  $\mu\text{m}$ . The commercially pure nickel powder (99.9%) (Atlantic Equipment Engineers, Upper Saddle River, NJ, USA) presents a spherical geometry, as seen in Fig. 1(b). The powder size distribution is characterized by a  $D_{10}$  value of 12.77  $\mu\text{m}$ , a  $D_{50}$  value of 22.34  $\mu\text{m}$ , and a  $D_{90}$  value of 40.44  $\mu\text{m}$ . The grade 304 stainless steel powder (Atlantic Equipment Engineers, Upper Saddle River, NJ, USA) presents an elongated geometry, as seen in Fig. 1(c). The powder size distribution is characterized by a  $D_{10}$  value of 13.50  $\mu\text{m}$ , a  $D_{50}$  value of 29.61  $\mu\text{m}$ , and a  $D_{90}$  value of 51.66  $\mu\text{m}$ . It should be noted that for the aluminum and stainless steel powders, many different morphologies and shapes of elongated particles were found during analysis, with a shape resembling a cylinder with rounded ends being most common. For simplicity, although the particle shapes are not all identical or geometrically perfect, the best descriptor for this type of shape was chosen to be "cylindrical."

### 2.2 MCGDS Fin Production Technique

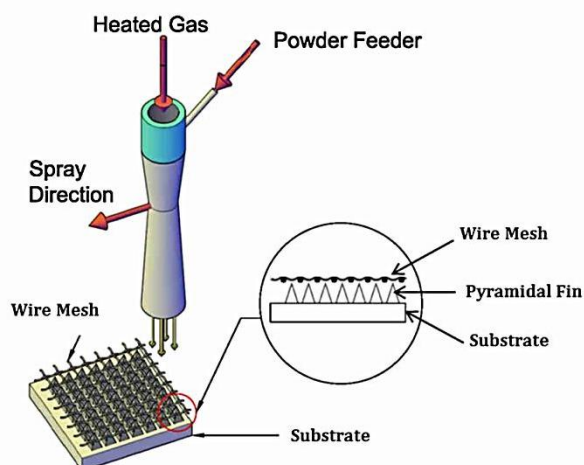
Square-based pyramidal pin fin arrays are manufactured using the MCGDS technique illustrated in Fig. 2. It involves masking of selected parts of the substrate by means of a plain woven, steel wire mesh mask (McMaster-Carr, Aurora, OH, USA). This manufacturing method allows building features of various shapes and dimensions depending on the chosen mesh characteristics used to mask the substrate. Features with reducing cross sections along the height such as pyramids were successfully created since the MCGDS process results in preferential build-up of material in the center of the mask openings (Ref 19). The deposition parameters are carefully chosen to ensure that the pyramids closely mimic the reversed geometry of the chosen wire mesh mask, with peaks aligning with the center of the mask openings. Further details on the manufacturing process used for the production of such features can be found elsewhere (Ref 19, 23). The mask selected for this work is characterized by a density of 12 openings per inch in both directions and a wire diameter of 0.035" (0.89 mm).

The commercial Cold Spray system used to produce the pyramidal fin arrays is the SST-EP (CenterLine (Windsor) Limited (Windsor, ON, Canada)). The pure aluminum





**Fig. 1** Scanning electron microscope (SEM) images of (a) As-received pure aluminum feedstock powder; (b) As-received pure nickel feedstock powder; (c) As-received stainless steel 304 feedstock powder



**Fig. 2** Schematic of the masking technique employed to produce pyramidal fin arrays (Ref 26)

powder was deposited using a polymer nozzle to prevent nozzle clogging typically experienced with aluminum powders, while a tungsten-carbide nozzle was used to deposit both the nickel and the stainless steel powders, as these powders would damage the polymer nozzle. Both nozzles have a 2.0-mm-throat diameter and a 6.4-mm-exit diameter, with a diverging section length of 126 mm. Table 1 presents the spray parameters used to manufacture the fins for each feedstock powder. The substrate material is aluminum alloy 6061-T6 with dimensions of 5.1- by 5.1-

by 1.3-cm thick. The substrates were cleaned and degreased using acetone prior to deposition.

### 3. Experimental Procedure

#### 3.1 Fin Characterization

Standard metallographic procedures were used to obtain cross sections of the sprayed fins, which were observed under a VHX-2000 depth-of-field microscope (Keyence, Mississauga, ON, Canada). Image analysis was performed using Clemex Vision Lite (Clemex Technologies Inc., Longueuil, QC, Canada) image analysis software. Furthermore, the cross sections were etched in order to highlight the particle's deformation. The aluminum and the stainless steel 304 fins were etched using Keller's and Methanolic Aqua Regia reagent, respectively, while the nickel fins were etched with a solution of 7.5 mL of hydrofluoric acid, 2.5 mL of nitric acid, and 200 mL of methanol. Finally, the fracture surface images were taken using a Scanning Electron Microscope (SEM) (Zeiss, Oberkochen, Germany) without any sample preparation after failure.

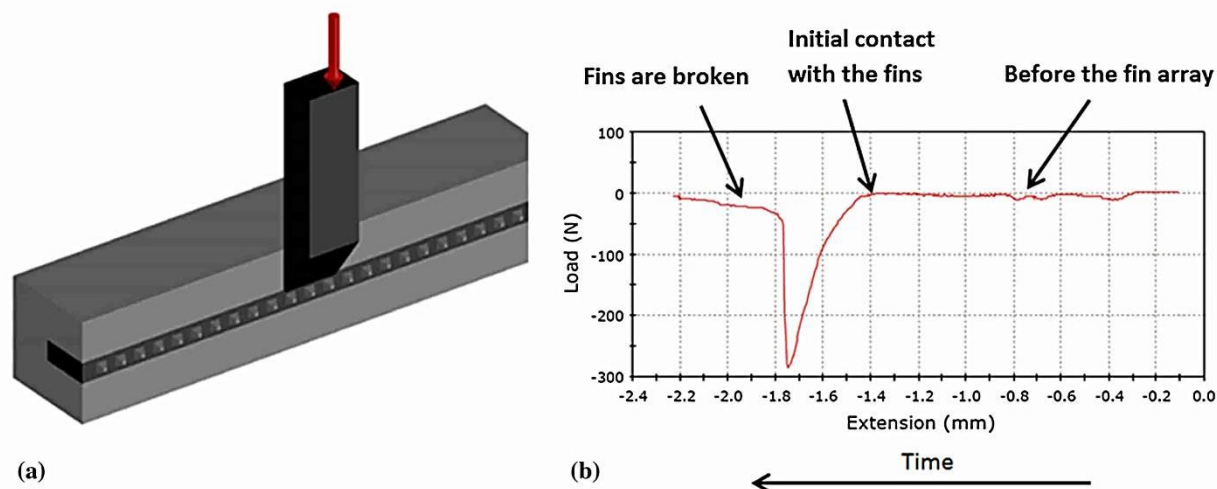
#### 3.2 Shear Strength Evaluation

The pyramidal fin arrays are designed to be implemented in high performance compact heat exchanger applications and will be subjected to shearing loads from the drag induced by fluid flow. Hence, shear tests are more appropriate than the purely tensile tests usually prescribed



**Table 1** Fin array spray parameters

| Powder type         | Nozzle inlet pressure, MPa | Nozzle inlet gas temperature, °C | Powder preheating temperature, °C | Standoff distance, mm |
|---------------------|----------------------------|----------------------------------|-----------------------------------|-----------------------|
| Pure aluminum       | 1.7                        | 350                              | N/A                               | 15                    |
| Pure nickel         | 2.8                        | 400                              | 300                               | 15                    |
| Stainless steel 304 | 3.1                        | 475                              | 300                               | 15                    |

**Fig. 3** (a) Diagram of the adhesion test apparatus; (b) typical load-displacement curve obtained (Ref 25)

for thermal spray coatings. The adhesion/cohesion strength of the pin fin array, in shear, was evaluated using the European standard EN15340-2007 (Ref 27). According to the procedure described in this standard, a tensile testing machine should apply a shearing load by means of a prescribed indenter until failure occurs. For this study, an Instron model 4482 (High Wycombe, Bucks, United Kingdom), equipped with a static load cell of 100 kN, was used. Single row samples of 24 fins (50.8 by 5.0 mm) were used to perform the shear strength evaluation. The shear distance was fixed to 50  $\mu\text{m}$  from the coating/substrate interface while the linear movement velocity was set to 50  $\mu\text{m/s}$ , as specified in EN15340-2007. The shear strength is calculated by dividing the measured force at failure by the pin fins' fracture area (quantified by optical microscopy). A representation of the apparatus used for this type of testing is shown in Fig. 3(a). Figure 3(b) shows the sharp changes in the load/displacement curve that occur when the indenter initially contacts the fins and when the fin failure occurs. The reported values are the average of 15 specimens for each group.

### 3.3 Particle Velocity Measurement

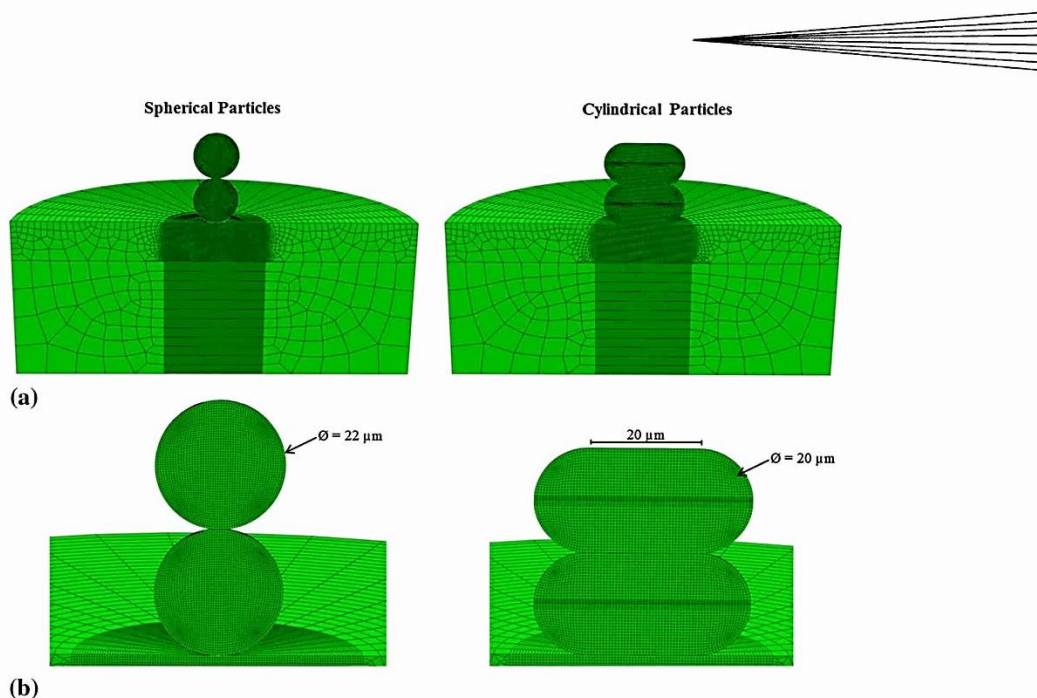
The in-flight particle velocity was measured using a Cold Spray Meter (CSM) (Tecnar Automation Ltd., Saint-Bruno, QC, Canada). The measures were taken in a free stream flow (without the substrate) at 15 mm from the nozzle exit at the centerline. Each particle velocity mea-

surement was taken over a 5 min period of time to ensure the collection of a sufficient number of particles. The average detection rate of the apparatus is approximately 150 particles/second. The apparatus is designed to accurately track particles with diameters ranging from 5 to 300  $\mu\text{m}$  traveling at velocities ranging from 10 to 1200 m/s.

The velocity is evaluated by measuring the time delay between two optical signals emitted by a particle when passing through a two-slits mask with a known distance between the slits. The system has a selection algorithm which matches the intensity of the first optical signal to that of the second to ensure that the same particle is measured from one slit to the next. In this manner, a large amount of data points are collected and the system performs a statistical analysis to determine the average and the standard deviation of the particle velocity.

### 3.4 Finite Element Analysis Model

**3.4.1 Simulation Layout.** A three-dimensional Finite Element Analysis (FEA) model using ABAQUS/Explicit was employed to simulate the collision behavior between cold-sprayed particles and substrate. In order to reduce the computational time, the three-dimensional models were simplified into half symmetric models. The mesh for both the substrate and the particles consists of hexahedral 8-node reduced integration thermally coupled elements (C3D8RT). The mesh size near the contact area was inputted as 1/50 of the particle diameter (Ref 28). The



**Fig. 4** Normal impact finite element model: (a) overall view; (b) zoomed on particles

**Table 2** General properties utilized in simulations (Ref 29)

| Properties                              | Pure aluminum | Pure nickel | Stainless steel 304 | Aluminum 6061-T6 |
|-----------------------------------------|---------------|-------------|---------------------|------------------|
| Density ( $\rho_0$ ), kg/m <sup>3</sup> | 2700          | 8890        | 7920                | 2700             |
| Specific heat ( $C_p$ ) (J/(kg K))      | 910           | 456         | 450                 | 900              |
| Thermal conductivity ( $k$ ), W/(m K)   | 237           | 91          | 15                  | 167              |

severe distortion of the mesh, occurring for particles with high velocity impact, was mitigated by setting the distortion control parameter to 0.4. The contact interaction at the particle/substrate and particle/particle interfaces was represented by a general contact condition with a coefficient of friction of 0.5.

The nickel particles were approximated by a half sphere with a diameter of 22  $\mu\text{m}$  whereas the aluminum and the stainless steel 304 particles were estimated by half cylinders with a diameter of 20  $\mu\text{m}$  and a height of 40  $\mu\text{m}$  to represent the average particles size and shape, as seen in Fig. 4. The simulations were done with two particles in order to be able to study the impact between the substrate and a particle as well as the impact between the particles. The multi-particle impacts were modeled without an initial gap between the particles. The general properties for these materials, such as density and thermal conductivity, as well as for the substrate (Al6061-T6) are given in Table 2. The initial temperatures for the substrate and the particles were set to 400 K (Ref 30). The particle velocities measured using the CSM for aluminum, nickel, and stainless steel 304 were, respectively,  $488 \pm 65$ ,  $459 \pm 61$ , and  $496 \pm 74$  m/s. For that reason, the particle initial velocity was set to 500 m/s in the model, for all three materials. Since the rotational velocity of the particles cannot be measured, the initial rotational velocity was

assumed to 0 m/s. All figures resulting from the numerical simulation shown in this paper are taken when the kinetic energy is minimal after the impact.

**3.4.2 Material Properties. Elastic Behavior:** The hydrodynamic stress (which causes volume changes) of the elastic response was modeled using the Mie-Grüneisen equation of state (EOS) in the linear  $U_s - U_p$  Hugoniot form given by (Ref 31):

$$p = \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} \left( 1 - \frac{\Gamma_0 \eta}{2} \right) + \Gamma_0 \rho_0 E_m, \quad (\text{Eq 1})$$

$$\eta = 1 - \frac{\rho_0}{\rho}, \quad (\text{Eq 2})$$

where  $p$  is the hydrodynamic stress,  $\rho_0$  is the initial density,  $\rho$  is the density,  $c_0$  is the material speed of sound,  $s$  is the Hugoniot slope coefficient,  $\Gamma_0$  is the Grüneisen's constant, and  $E_m$  is the internal energy per unit mass.

Conversely, the linear elastic model was assumed for the deviatoric stress (which induces shape changes) (Ref 31):

$$d = 2G_c \epsilon_s, \quad (\text{Eq 3})$$

where  $d$  is the deviatoric stress,  $G_c$  is the elastic shear modulus, while  $\epsilon_s$  is the elastic deviatoric strain.



**Table 3 Elastic behavior parameters (Ref 32)**

| Parameters                          | Pure aluminum | Pure nickel | Stainless steel 304 | Aluminum 6061-T6 |
|-------------------------------------|---------------|-------------|---------------------|------------------|
| Shear modulus ( $G$ ), GPa          | 26            | 76          | 86                  | 26               |
| Grüneisen's constant ( $\Gamma_0$ ) | 1.97          | 1.93        | 1.93                | 1.97             |
| Speed of sound ( $c_0$ ), m/s       | 5386          | 4650        | 4570                | 5240             |
| Hugoniot slope ( $s$ )              | 1.34          | 1.45        | 1.49                | 1.40             |

The elastic properties for the four materials utilized in the simulations are given in Table 3.

**Plastic Behavior:** The high strain rates model developed by Preston, Tonks, and Wallace (Ref 13) was used to describe the plastic response of the materials. The flow stress ( $\sigma$ ) is defined as

$$\sigma = 2 \left[ \hat{\tau}_s + \alpha \ln \left[ 1 - \varphi \exp \left( -\delta - \frac{\theta \varepsilon_p}{\alpha \varphi} \right) \right] \right] G_p, \quad (\text{Eq 4})$$

$$\alpha = \frac{s_0 - \hat{\tau}_y}{p}, \delta = \frac{\hat{\tau}_s - \hat{\tau}_y}{\alpha}, \varphi = \exp(\delta) - 1, \quad (\text{Eq 5})$$

where  $\hat{\tau}_s$  is the normalized work-hardening saturation stress,  $\hat{\tau}_y$  is the normalized yield stress,  $\theta$  is strain hardening rate,  $\varepsilon_p$  is the equivalent plastic strain,  $s_0$  is the saturation stress at 0 K, and  $p$  is the strain hardening constant. The plastic shear modulus ( $G_p$ ) is function of temperature and it is defined by this equation from the mechanical threshold stress (MTS) model (Ref 33):

$$G_p(T) = G_0 - \frac{D}{\exp\left(\frac{T_0}{T}\right) - 1}, \quad (\text{Eq 6})$$

where  $G_0$  is the shear modulus at 0 K,  $D$  is a material constant,  $T_0$  is a temperature material constant, and  $T$  is the node temperature.

The equations to determine the work-hardening saturation stress ( $\hat{\tau}_s$ ) and yield stress ( $\hat{\tau}_y$ ) are given below:

$$\hat{\tau}_s = \max \left\{ s_0 - (s_0 - s_\infty) \operatorname{erf} \left[ \kappa \hat{T} \ln \left( \frac{\gamma \dot{\varepsilon}_p}{\dot{\varepsilon}_p} \right) \right], s_0 \left( \frac{\dot{\varepsilon}_p}{\gamma \dot{\varepsilon}} \right)^\beta \right\}, \quad (\text{Eq 7})$$

$$\hat{\tau}_y = \max \left\{ y_0 - (y_0 - y_\infty) \operatorname{erf} \left[ \kappa \hat{T} \ln \left( \frac{\gamma \dot{\varepsilon}_p}{\dot{\varepsilon}_p} \right) \right], \min \left\{ y_1 \left( \frac{\dot{\varepsilon}_p}{\gamma \dot{\varepsilon}} \right)^{y_2}, s_0 \left( \frac{\dot{\varepsilon}_p}{\gamma \dot{\varepsilon}} \right)^\beta \right\} \right\}, \quad (\text{Eq 8})$$

where  $\hat{T} = T/T_m$ ,  $T_m$  is the melting temperature,  $s_\infty$  is the saturation stress near the melting temperature,  $\kappa$  is the temperature dependence constant,  $\gamma$  is the strain rate dependence constant,  $\dot{\varepsilon}_p$  is the plastic strain rate,  $\beta$  is the high strain rate exponent,  $y_0$  is the yield stress constant at 0 K,  $y_\infty$  is the yield stress constant near melting temperature,  $y_1$  is the medium strain rate constant,  $y_2$  is the medium strain rate exponent. The two definitions in the maximum function are used to model both low and high strain rate regimes, respectively. For the lower strain

rates (where stresses are governed by thermal activation) the first definition of the maximum function is used whereas the second definition is used for the high strain rate regime (which is governed by dislocation drag mechanisms).

The material parameter ( $\dot{\varepsilon}$ ) in Eq 7 and 8 is defined as

$$\dot{\varepsilon} = \frac{1}{2} \left( \frac{4\pi p}{3M} \right)^{1/3} \left( \frac{G_p(T)}{\rho} \right)^{1/2}, \quad (\text{Eq 9})$$

where  $M$  is the atomic mass.

The PTW and MTS parameters used in the simulations for each material are shown in Table 4. It should be noted that the PTW model and the MTS shear behavior are not included in ABAQUS. Consequently, these models were integrated in the finite element analysis by a user subroutine (VUHARD).

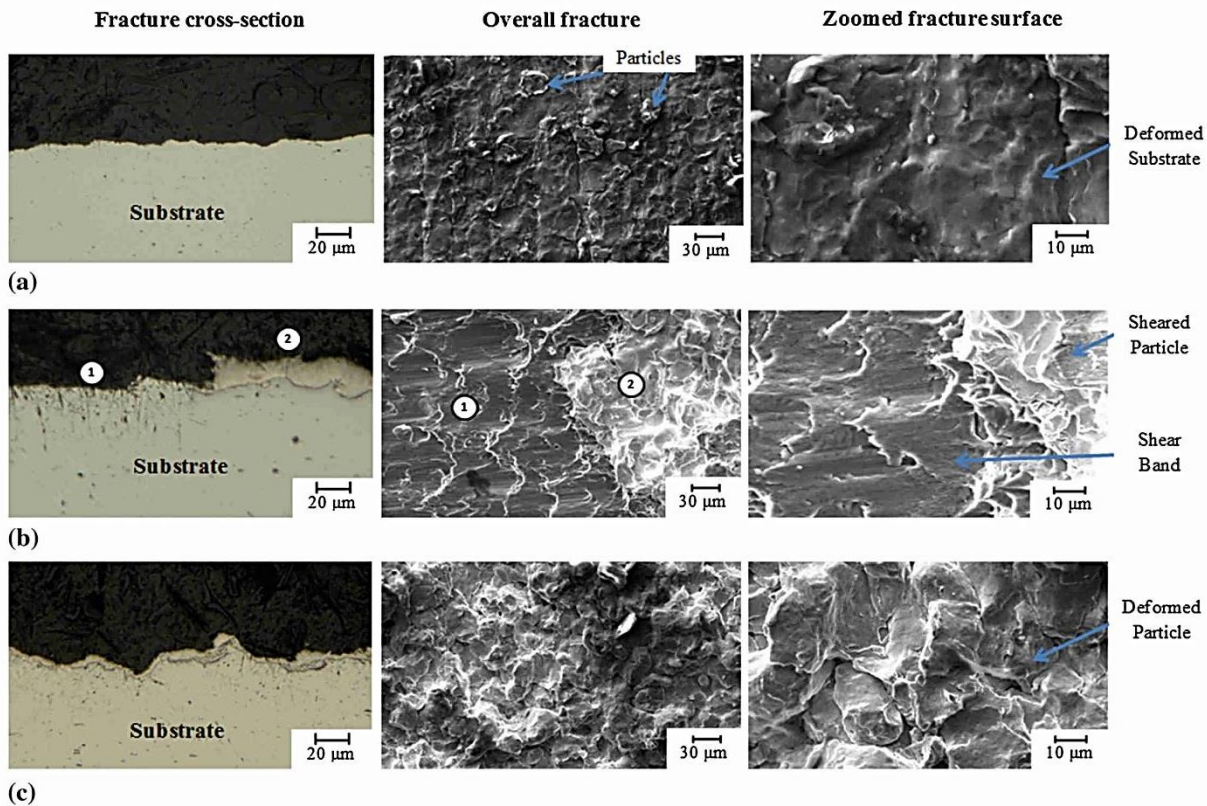
## 4. Results

### 4.1 Shear Strength

The shear strength of the different fin materials manufactured on Al6061 substrates were measured using the apparatus described in section 3.2. The results of these measurements are the following: the nickel fins exhibit the highest shear strength with a value of  $117 \pm 19$  MPa followed by the stainless steel fins at  $37 \pm 7$  MPa and finally the aluminum fins with a shear strength of  $22 \pm 4$  MPa. The aluminum fracture cross section (Fig. 5a, left) shows no presence of aluminum at the fracture location situated at the substrate coating interface, which indicates an adhesive failure. The nickel fracture cross section (Fig. 5b, left) presents a combination of adhesive and cohesive failure, with partial coating remaining on the substrate surface where the failure occurred. The results seem to indicate that the fracture began as an adhesive failure, and at some point defects in the coating microstructure promoted the change of failure mode from adhesive to cohesive failure. The stainless steel fins failed in a cohesive manner, as seen by the presence of a layer of deposited particles that remained attached to the substrate (Fig. 5c, left). Therefore, it is possible to identify three different failure modes: adhesive, mixed, and cohesive. Furthermore, the SEM images of the fracture surfaces, also presented in Fig. 5, support the failure mode identifications. The images of the aluminum fracture top surface (Fig. 5a, middle and right) show the deformed substrate which points out the debonding of the particles from the substrate. Only a few deformed particles could be seen on the

**Table 4 Plastic behavior parameters (Ref 34-41)**

| Parameters                                      | Pure aluminum          | Pure nickel            | Stainless steel 304    | Aluminum 6061-T6       |
|-------------------------------------------------|------------------------|------------------------|------------------------|------------------------|
| Strain rate dependence constant ( $\gamma$ )    | 0.0001                 | 0.00001                | 0.001                  | 0.0001                 |
| Strain hardening rate ( $\theta$ )              | 0.0529                 | 0.010                  | 0.020                  | 0.0529                 |
| Strain hardening constant ( $p$ )               | 3                      | 2                      | 10                     | 3                      |
| Yield stress constant at 0 K ( $y_0$ )          | 0.00942                | 0.0001                 | 0.0069                 | 0.00942                |
| Yield stress constant at melting ( $y_\infty$ ) | 0.00566                | 0.0001                 | 0.0015                 | 0.00566                |
| Medium strain rate constant ( $y_1$ )           | 0.00354                | 0.094                  | 0.050                  | 0.00354                |
| Medium strain rate exponent ( $y_2$ )           | 0.50                   | 0.575                  | 0.46                   | 0.50                   |
| High strain rate exponent ( $\beta$ )           | 0.23                   | 0.25                   | 0.25                   | 0.23                   |
| Saturation stress at 0 K ( $s_0$ )              | 0.032                  | 0.0045                 | 0.050                  | 0.032                  |
| Saturation stress at melting ( $s_\infty$ )     | 0.00791                | 0.00055                | 0.0075                 | 0.00791                |
| Temperature dependence constant ( $\kappa$ )    | 0.494                  | 0.110                  | 0.300                  | 0.494                  |
| Atomic mass ( $M$ ), kg/atom                    | $4.48 \times 10^{-26}$ | $9.24 \times 10^{-26}$ | $9.27 \times 10^{-26}$ | $4.48 \times 10^{-26}$ |
| Shear modulus at 0 K ( $G_0$ ), GPa             | 28.8                   | 85.1                   | 85.0                   | 28.8                   |
| Material constant ( $D$ ), GPa                  | 3.44                   | 9.13                   | 10.00                  | 3.44                   |
| Temperature material constant ( $T_0$ ), K      | 215                    | 269                    | 298                    | 215                    |

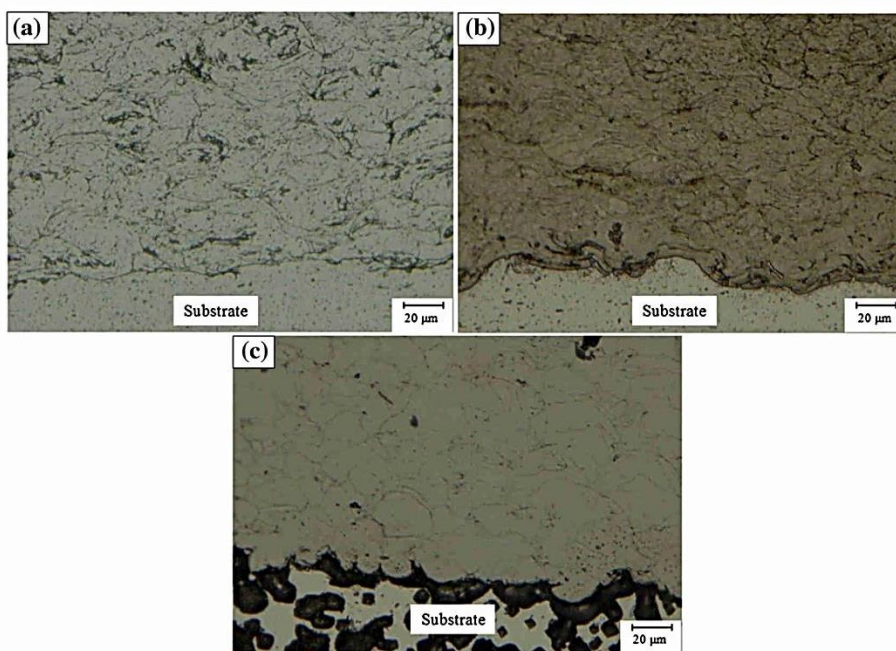

**Fig. 5** Different failure modes shown by fracture cross section, overall and zoomed fracture surface showing (a) aluminum with an adhesive failure; (b) nickel with a mixed (c) stainless steel 304 with a cohesive failure, where the fractures occurred from left to right

surface after the shear test is performed (Fig. 5a, middle). The nickel fracture surface (Fig. 5b, middle and right) presents shear bands on the substrate surface and after a certain distance, the transition to a trans-granular fracture of the nickel particles, which indicates a bond of similar strength at the interface and between nickel particles. The images of the stainless steel particles (Fig. 5c, middle and

right) reveal an inter-granular fracture in the fins which indicates a debonding between the stainless steel particles. This indicates that the bonding between the particles and the substrate is stronger than the cohesive strength of the stainless steel.

The high shear strength combined with the shear bands observed for the case of nickel is believed to indicate the





**Fig. 6** Etched cross section of fins made of (a) aluminum; (b) nickel; (c) stainless steel 304

presence of a metallurgical bond between the nickel particles and the substrate at the interface, while the lower strength value and the fact that there is no evidence of metallurgical bonding for the other two materials support the idea that the bonding mechanism is dominated by mechanical anchoring in these cases. The bonding between the stainless steel particles and the substrate cannot be evaluated due to the weaker cohesive strength of the particle-particle bond.

#### 4.2 Particles Deformation

Typically etched cross sections of the fins are shown in Fig. 6. For nickel, the particles are well deformed considering that the as-received powder is spherical whereas the deposited particles are splat shaped (Fig. 6b). The two cylindrical powders (aluminum and stainless steel) exhibit a less deformed geometry. They do not have splat shapes similar to what was observed for the nickel powder. It should be noted, by looking at the etched cross section for the aluminum or stainless steel, that it is not possible to determine if the cylindrical particles are more likely to hit the substrate in a preferred direction (horizontally, vertically, or at an angle). It is worth mentioning that the coatings are almost fully dense, with porosity levels lower than 3%.

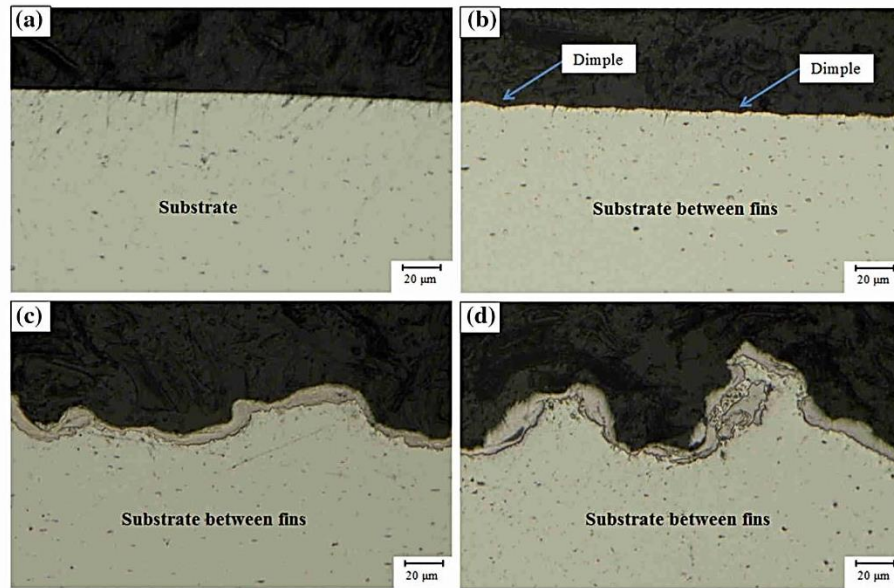
Looking at the substrate-coating interfaces in Fig. 6, it is possible to observe the substrate deformation caused by the deposition of the different powders. The soft Al6061-T6 substrate is more deformed/penetrated by the hard and heavy nickel and stainless steel powders (Fig. 6b and c)

than by the light and soft aluminum powder (Fig. 6a). Thus, mechanical anchoring to the substrate is better for nickel and stainless steel than for aluminum. This fact could partially explain the low adhesive shear strength of aluminum fins compared to the strong integrity of the stainless steel-substrate interface and the high shear strength of the nickel fins.

#### 4.3 Deposition Between the Fins

One observation that can be made is that deposition between the nickel and between the stainless steel fins occurs while no deposition has been found between the aluminum fins. This is illustrated in Fig. 7, which shows cross sections of the area between manufactured fins, with the as-received substrate shown for comparison purpose. During the manufacturing of aluminum fins, the aluminum particles lightly deform the substrate between the fins without bonding, as revealed by the presence of dimples on the substrate after the spray, while the hard nickel and stainless steel particles strongly deformed the aluminum substrate and a layer of these particles has bonded to the substrate between the fins, whereas the subsequently sprayed particles did not adhere to this initial layer. Since only a single layer of nickel or stainless steel is deposited between the fins, on the substrate, it shows that the critical velocity for the particle-substrate combination is not the same as the critical velocity for the particle-particle combination, otherwise more than a single layer would have adhered (Ref 42).





**Fig. 7** Typical cross section of the surface between the fins (a) As-received; (b) when aluminum fins are manufactured; (c) when nickel fins are manufactured; (d) when stainless steel 304 fins are manufactured

#### 4.4 Simulation Results

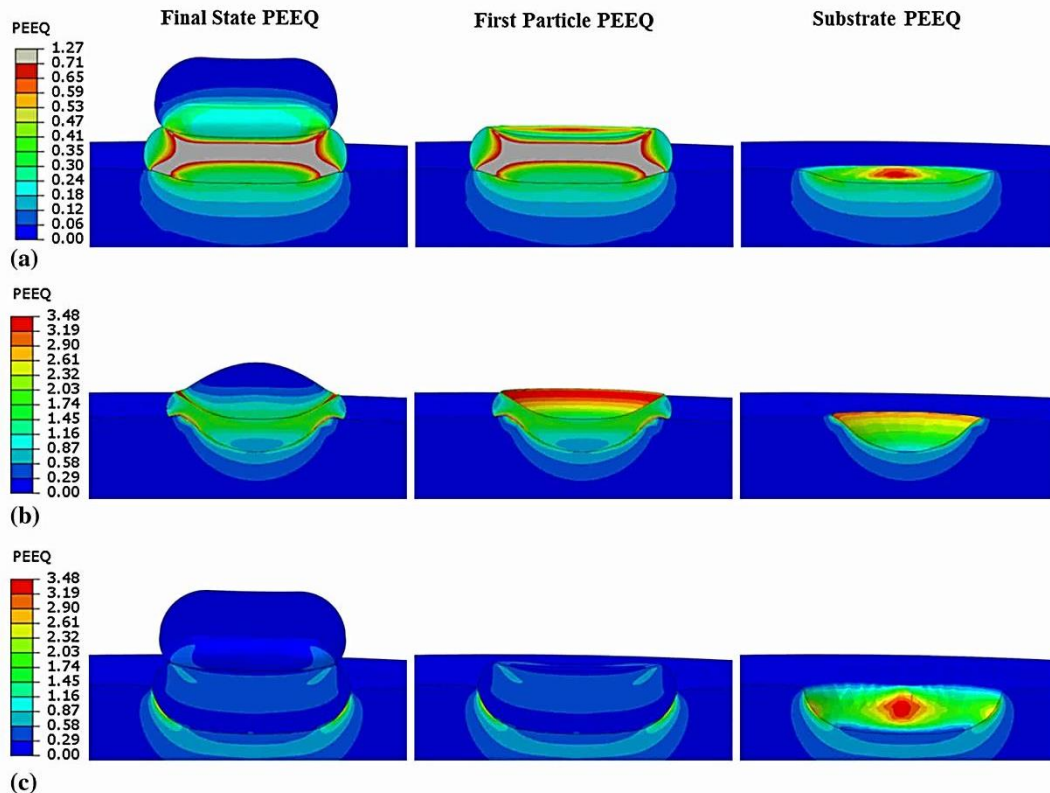
In order to have a metallurgical bond (cohesive forces), the oxide layer present on the material's surface must be removed and an intimate contact between the two metallic surfaces is also necessary, similar to the conditions required for cold welding. It was suggested by several authors that the PEEQ is a good indicator of the bonding quality (Ref 1, 6, 43), since a high PEEQ value can be associated to a better removal of the oxide layers that are present on the surface of the powder, leaving a clean metal to metal contact that is favorable for bonding. From the cold spray literature (Ref 44-46), it is possible to establish that the presence of surface oxides has a direct effect on the velocity required to achieve deposition in cold spray. Since the three materials involved in this study have very similar critical velocities and erosion velocities (thus spray windows) (Ref 47), it is assumed that the oxides present at the surface of the particles would tend to require similar plastic strain to be removed.

The equivalent plastic strain (PEEQ) values were generated from the numerical simulations and are presented in Fig. 8, which illustrates the final state of a two particle impact for the different materials under study as well as the first particle and the substrate PEEQ. The PEEQ values at the location where failure occurs seem to correlate qualitatively with the shear strengths measured experimentally (section 4.1). As seen in Fig. 8, the highest PEEQ is observed with the nickel particle's impacts at the particle-substrate and the particle-particle interface as well as with the stainless steel particle-substrate collision. However, stainless steel sample fails in a cohesive mode, which means the adhesion shear strength is not reflected in the measures reported in section 4.1. Therefore, the PEEQ values that

should be considered for the stainless steel particle impacts are the low values at the particle-particle interface (Fig. 8c, middle) to compare against the cohesive shear strength measured. Subsequently, since the high PEEQ on the stainless steel substrate is not taken into account when measuring the shear strength, nickel has the highest PEEQ values at the location where the failure occurs and also presents the highest shear strength, by far. In addition, the surface facing the substrate (under the particle that hit the substrate) has a lower PEEQ than the PEEQ measured on the substrate (Fig. 9) due to the low deformation of the stainless steel particle whereas the substrate deformed significantly. It should be noted that only one of the two interface surfaces are shown for all cases, except the stainless steel in the Al6061-T6 substrate, since showing one or both will lead to the same conclusion (high PEEQ on both surfaces or low PEEQ on both surfaces).

As mentioned above, for the case of stainless steel, the equivalent plastic strain is low at the particle-particle contact area whereas it is high on only one surface at the substrate-particle interface (Fig. 8 and 9). This suggests that the oxide layer will not be exposed to favorable conditions for its removal at the particle-particle contact zone, thus reducing the potential of metallurgical bonding between particles. Additionally, it will experience also non-favorable conditions for its removal for the impact of the particles with the substrate, since only the substrate will be clean of an oxide. The stainless steel part contacting the substrate is believed to still have its oxide due to the low PEEQ at that surface. The non-favorable condition of the particle-particle contact zone seems to be validated by the SEM surface fracture analysis of Fig. 5(c), showing that mechanical anchoring appears to





**Fig. 8** Final state, first particle, and substrate PEEQ at a normal impact for (a) aluminum; (b) Nickel; (c) stainless steel 304. It should be noted that the PEEQ scale for (a) is different from (b) and (c) to better observe the features of interfaces' deformation

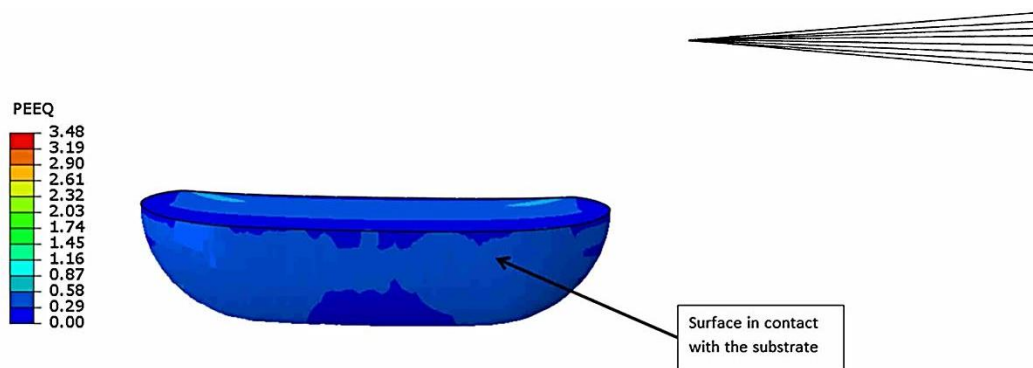
be the main adhesion mechanism between particles (deduced from the inter-granular fracture) while stronger bonds exist between the particles and substrate due the cohesive failure. It is also believed that the bond between the particle and the substrate is still mechanical anchoring due the low PEEQ on the stainless steel in contact to the substrate. The coating fails in cohesive mode because of the high anchoring of the first layer to the substrate as seen in Fig. 6(c) or with the deep crater of Fig. 8(c) compared to the low deformation of particle-particle contact.

For the aluminum particles, a low PEEQ value is observed for both impact types (particle-particle and particle-substrate). This can potentially explain the low shear strength measured for this case. There is little cohesive force between the substrate and the particles since the oxide layer is potentially not well removed, due to the low amount of plastic strain. The bonding is thus expected to be mostly based on mechanical anchoring, as seen in the SEM fracture images (Fig. 5a). Even if the PEEQ is comparable between the particle-particle and particle-substrate collision, all shear tests reported for this material ended in adhesive failure, which could be explained by the low amount of mechanical interlocking of the particles to the smooth substrate compared to stronger particle-particle mechanical interlocking, as suggested by Fig. 6(a).

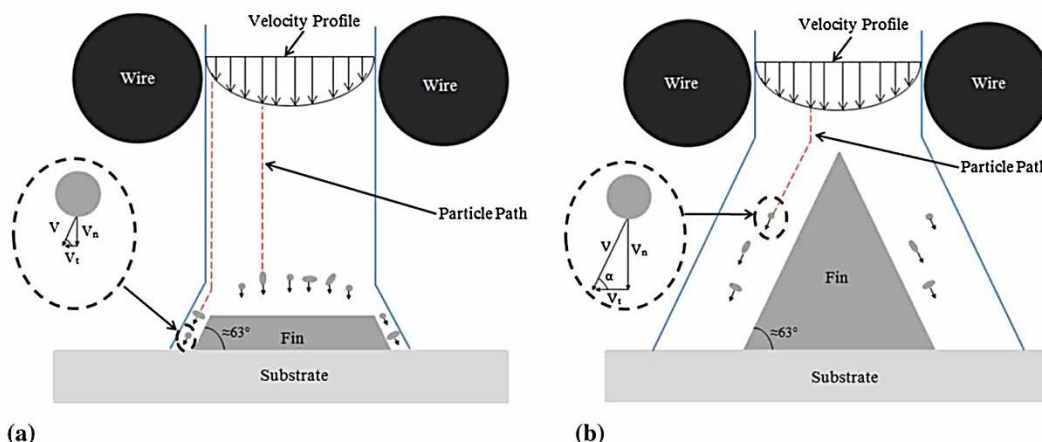
The high deformation of the substrate by the nickel and the stainless steel particles compared to the aluminum particles (as observed in Fig. 6) is also observed in the numerical simulation results of Fig. 8 which would imply stronger mechanical interlocking for these particles to the substrate and higher PEEQ values.

It should be noted that for the cylindrical particles, the results are shown with a horizontal impact. However, simulations have been done in three different orientations (horizontal, vertical, angled at 45° with the horizontal). The different impact orientations changed the absolute value of the PEEQ and the final deformation of the particles/substrate but do not change the conclusions, neither for stainless steel particles, nor for the aluminum particles.

Equivalent plastic strain has been linked to the particle and substrate oxide removal, one requirement to have local metallurgical bonding. Nevertheless, an intimate contact between the two surfaces is required to achieve bonding during the Cold Spray process. The contact pressure (CPRESS) resulting from the simulations is between 800 and 1000 MPa for all cases. In cold welding, the pressure required at ambient temperature for most non-ferrous material is in this order of magnitude (Ref 48). Therefore, a metallurgical bond for the nickel particles may occur since the two requirements (oxide removal/cleaning and proper contact pressure) for bond-



**Fig. 9** Plastic equivalent strain of the surface under the stainless steel particle in contact with the cleaned substrate



**Fig. 10** Schematic of the particles flowing around the fin feature (a) before the fin reaches a critical height; (b) after the fin has reached a critical height

ing are met, whereas even if the contact pressure is high enough, the oxide layers are unlikely to have been removed for the other particles at the fracture surface (inferred from the low PEEQ values). Consequently, the surfaces are not sufficiently close to form a bond (due to the oxide layer barrier) despite the large contact pressure.

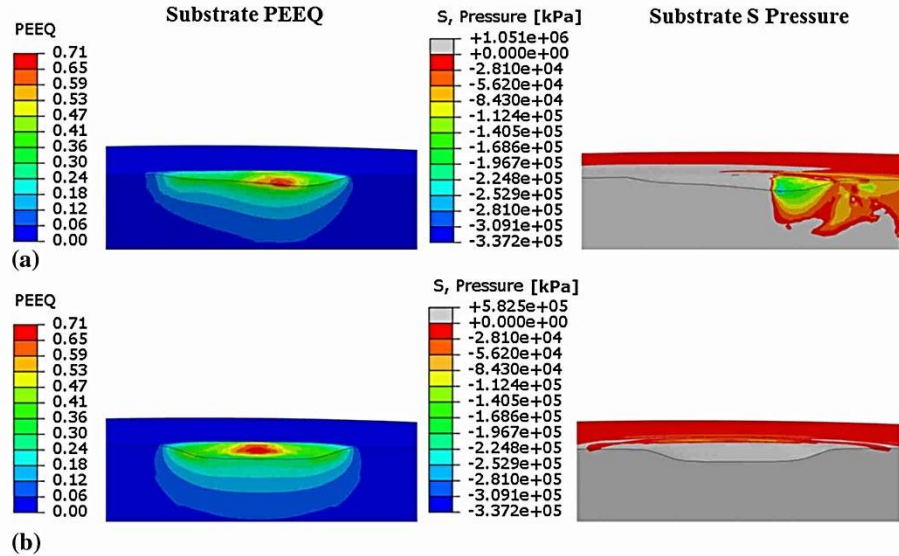
Modeling results indicated that the zones of high temperature are at the same locations than the zones of high PEEQ, as expected. However, the temperature values obtained for the various impacts modeled are lower than the melting point of the respective materials (less than 80% of the melting temperature, for all cases), which leads to the conclusion that local melting does not occur.

Another interesting fact that can be seen in Fig. 8 is the effect of the particle shape. For spherical particles such as nickel in this study, the zone of maximum PEEQ forms a circular ring whereas for the cylindrical particles (aluminum and stainless steel), this zone of maximal strain at the interface is concentrated at the middle of the impact crater on the substrate. This could result in a different surface oxide removal pattern and consequently change the available bonding area. Indeed, the spherical particles would potentially experience a larger area of oxide removal while the cylindrical particles will have a smaller

area where the oxides are removed. In order to confirm that the change of the location of the maximum PEEQ at the interface is not a material property but rather depends on particle shape for the same impact orientation, a simulation with spherical stainless steel particles was also performed. This numerical simulation resulted in a similar PEEQ profile as for the nickel particle. The PEEQ rings were also observed for impacts with cylindrical particles in the vertical orientation. This result is expected since when impacted vertically, the substrate is penetrated by the rounded end of the cylindrical particle.

In order to investigate the different deposition behaviors between the fins (underneath the wire mesh mask) for the three feedstock powders, the numerical model was used to simulate impacts occurring at an angle. As schematized in Fig. 10, the particles impact the substrate with a tangential and a normal velocity component, with their total velocity following the fin pyramid angle. After the fin is built to a certain height, it is observed that deposition occurs between the fins when spraying nickel or stainless steel. Before this critical height, the tangential component of particle velocity is negligible, as the portion of the flow that is deviated originates mostly from the slower section of the velocity profile (Fig. 10a). When the





**Fig. 11** Equivalent plastic strain (PEEQ) and hydrodynamic pressure (S pressure) for the cylindrical aluminum particles for (a) angular impact; (b) normal impact

pyramid is sufficiently tall, the flow redirection is substantial enough to obtain particles from center of the particle jet (where the velocity is higher than on the sides), which have a sufficiently large normal velocity component to create deposition under the mask wires (Fig. 10b).

The pyramid angle for the samples described in this work is measured to be on average  $63^\circ$  (Ref 26). For the numerical model, the particles were injected to flow parallel to the pyramid edge. The normal and tangential particle velocities with respect to the substrate top surface were determined using the following relations:

$$V_n = V \sin(\alpha), \quad (\text{Eq 10})$$

$$V_t = V \cos(\alpha), \quad (\text{Eq 11})$$

where  $V_n$  is the normal velocity,  $V_t$  is the tangential velocity,  $V$  is the measured particle velocity, and  $\alpha$  is the pyramid angle. Consequently, the values used for the normal and tangential velocity in these simulations are respectively 446 and 227 m/s.

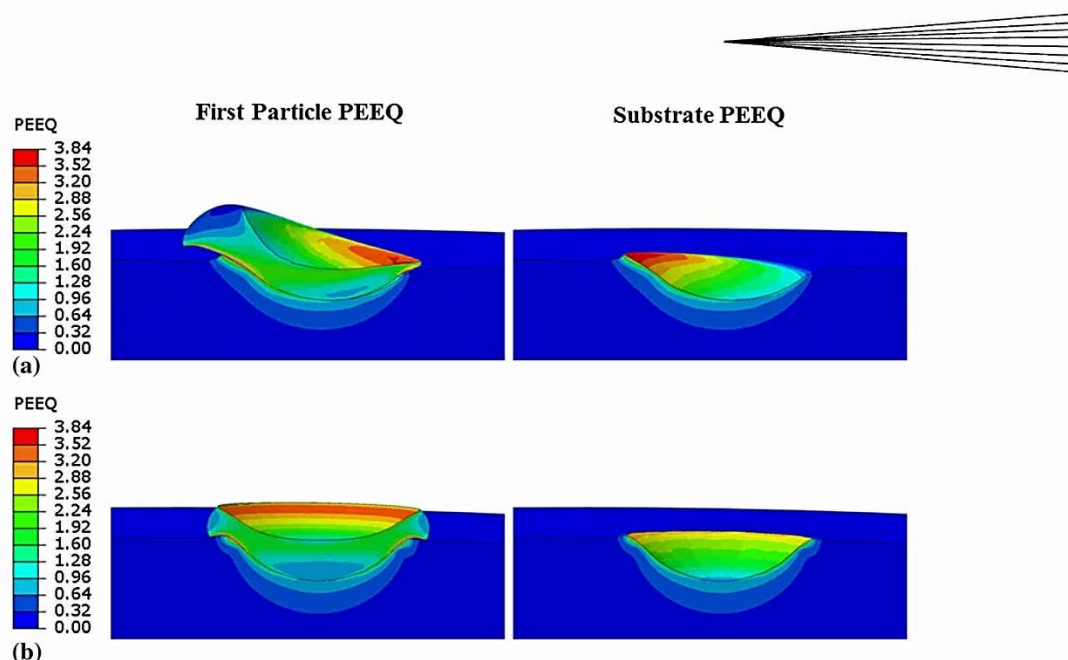
Figure 11 shows the comparison between a normal impact (such as the ones that lead to the build-up of fins) and an angular impact for aluminum particles, as potentially experienced by the particles that reach the substrate between the fins. For both impact types, the zone of highest equivalent plastic strain is located where the particle first hits the substrate. For the normal impact, this zone is located at the center of the crater, while for the angular impact this zone is shifted toward the point of first contact between the particle and the substrate. A notable difference between these two impacts is the presence of a zone of tensile stress for the angular impact due to the friction in the tangential direction that is not present in the normal impact. This tensile stress is detri-

mental to the particle adhesion as previously discussed in other works (Ref 1, 49). This difference in stress is believed to be a factor that explains why there is adhesion for normal contacts while for tangential collisions the particle will not adhere to the substrate but deform it and bounce off, leaving dimples without deposition between the fins. It should be noted that the hydrodynamic pressure (S Pressure) has a positive sign for compressive stress and a negative sign for tensile stress. On the S Pressure figures, the colored scale was set to show only tensile stresses in order to put emphasis on the tension zones. Therefore, all compressive stresses are shown in gray.

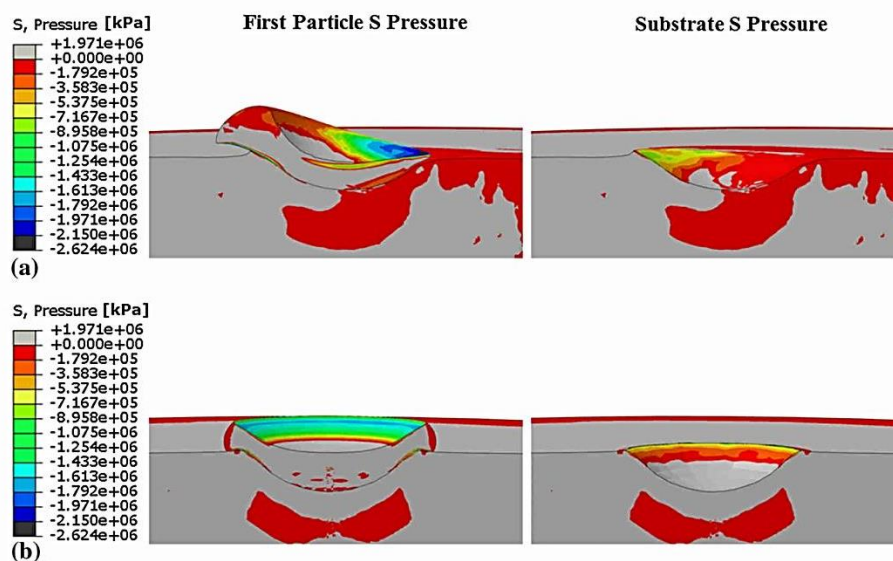
Figure 12 shows the equivalent plastic strain for the normal and angular impact of nickel particles. A higher PEEQ is observed for the angular collision for both the first particle and the substrate due to the smearing deformation. Similar to the aluminum numerical simulation, the S Pressure was computed for the nickel impacts (Fig. 13).

Although the total PEEQ is comparable for the inter-particle and the particle-substrate angular impact, the tensile stress is higher for the former. Then, one could believe that the higher tensile stress at the particle-particle impact explains why there is deposition on the substrate but not on the particle, as observed experimentally. However, for the normal impact, there is also a tensile stress zone, near the top of the impact point on the first particle, which would tend to prevent particle-particle bonding (as suggested by Assadi et al. (Ref 1) and Schmidt et al. (Ref 49)), as opposed to what has been observed experimentally (see Fig. 6b). This tension zone is unexpected for a normal impact since the particle does not have a tangential velocity component at impact. The presence of this tension zone is attributed to the material flow in the direction parallel to





**Fig. 12** Equivalent plastic strain (PEEQ) on the first particle and on the substrate for the nickel particles at (a) angular impact; (b) normal impact

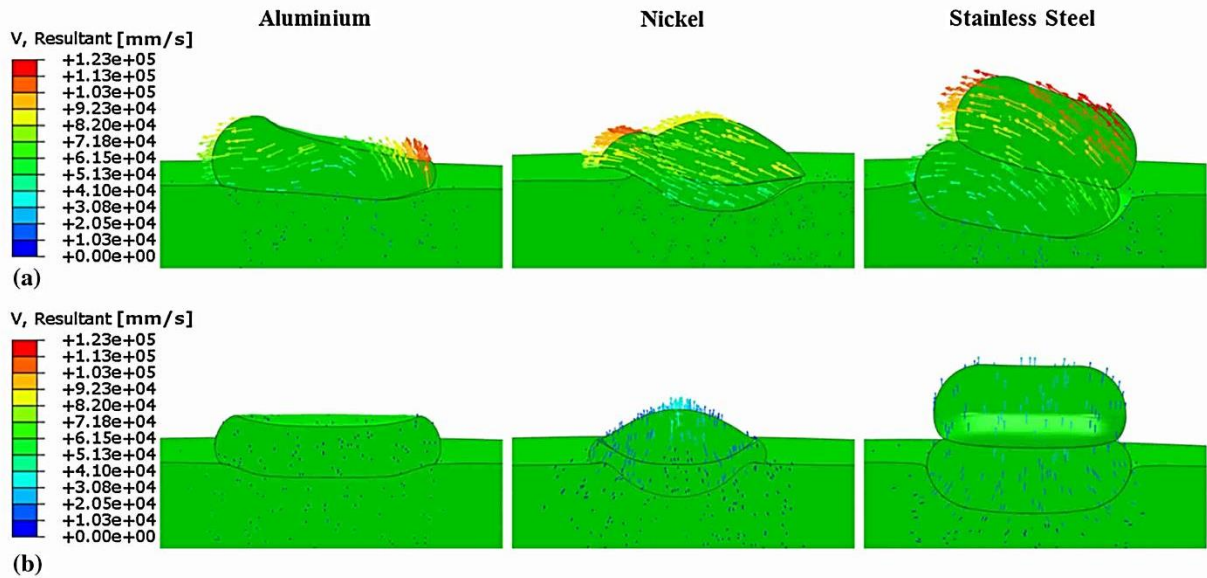


**Fig. 13** Hydrodynamic pressure (S pressure) on the first particle and on the substrate for the nickel particles at (a) angular impact; (b) normal impact

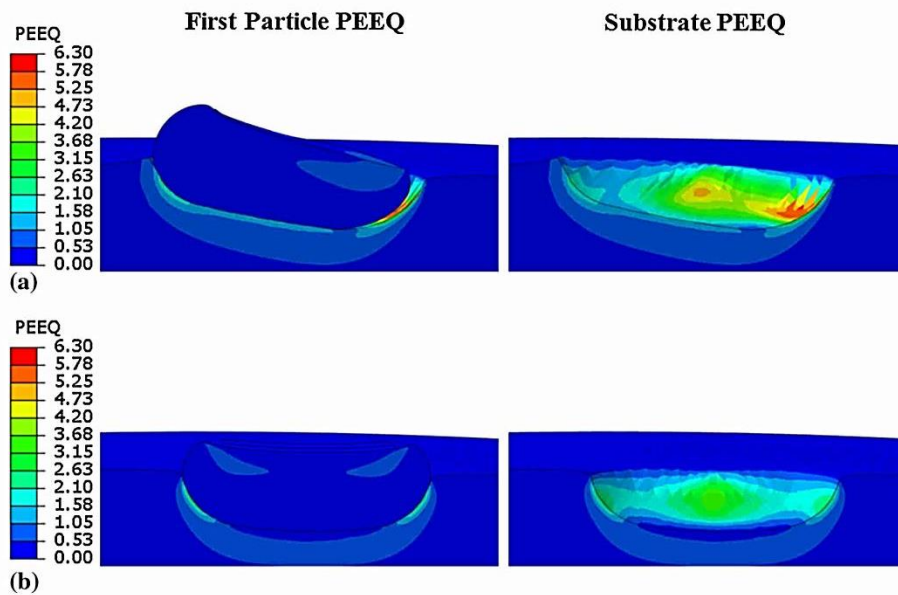
the surface that occurs during a normal impact. This material flow causes friction on both particles' surfaces, which in turn causes tensile stresses that overcome the compression stresses induced by compaction. At the bottom of the impact crater, there is less particle movement during the plastic deformation which causes less intense smearing and friction, which as a consequence leaves a zone of compressive stress. These observations made with the modeling results suggest that another mechanism than

tensile stress must contribute to prevent the bonding of nickel particles on nickel particles at an angle.

By computing the residual velocity, when the kinetic energy is minimum, it is possible to observe an angular momentum being created for angular collisions (Fig. 14). The shear force caused by the competition of the friction parallel to the surface and the particle's momentum induces a spin. It was shown by several authors (Ref 50, 51) working with angular inelastic collision with balls that



**Fig. 14** Resultant velocity after the collision for (a) angular impact; (b) normal impact

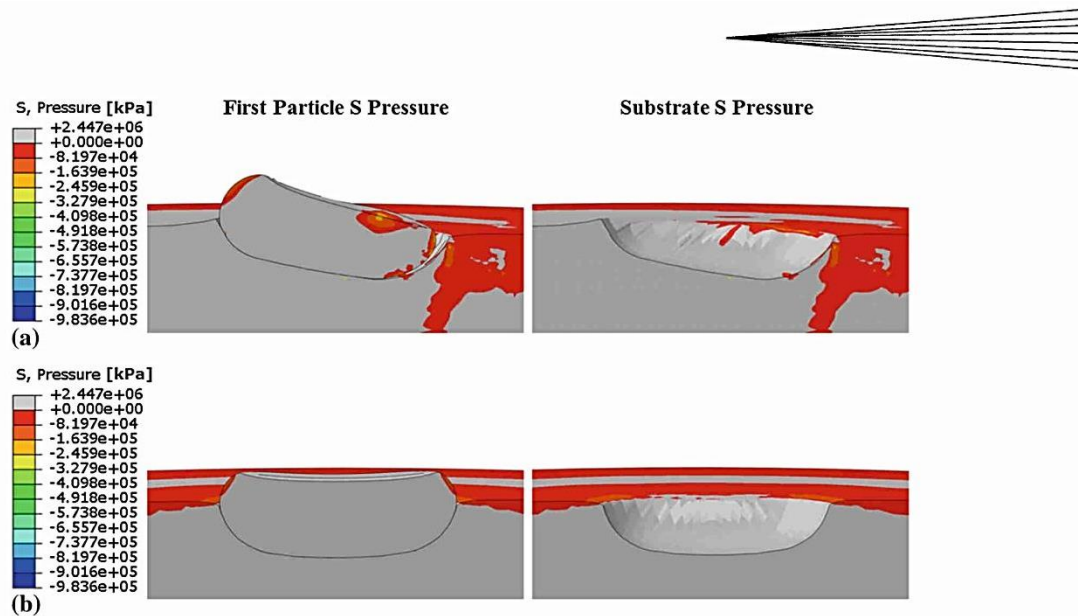


**Fig. 15** Equivalent plastic strain (PEEQ) on the first particle and on the substrate for the stainless steel particles at (a) angular impact; (b) normal impact

after the impact the ball leaves the surface with an angular momentum, while for a normal inelastic collision it leaves the surface without any rotational momentum. In terms of bonding strength, this result means that for particles in an angled impact, the bond has to resist an angular momentum while for normal impacts it only has to fight the restitution momentum (perpendicular to the substrate surface) (Fig. 14). In addition, the restitution velocity is

higher for the angular impact compared to the normal impact. This observation can be made for the three materials tested, as shown in Fig. 14. For the specific case of nickel deposited at an angle, the restitution velocity resulting from the particle-substrate impact is lower for the portion of the particle embedded in the substrate, while the unembedded portion is at higher velocity. It is believed that the tangential velocity of the portion





**Fig. 16** Hydrodynamic pressure (S pressure) on the first particle and on the substrate for the stainless steel particles at (a) angular impact; (b) normal impact

embedded in the substrate is slowed by the material that flowed from the substrate, whereas for the other portion there is no obstacle to slow the tangential velocity component. Furthermore, the higher average rotational velocity of the particle/particle impact debonds the particles whereas particles impacting the substrate experience a lower debonding force, as observed experimentally. These observations may explain why there is only one layer of material that stays attached to the substrate underneath the wire mesh mask.

For the stainless steel, higher PEEQ is encountered for the angular collision compared to the normal impact (Fig. 15). Figure 16 shows a zone of limited tension for the angular contact while none can be observed for the normal contact scenario, which supports the idea that the tensile stress zones do not dictate whether adhesion occurs and is sustained or not. Contrary to what was observed for the nickel particles' impact, a tensile stress zone is not present for the normal impact between stainless steel particles due to the low amount of tangential plastic strain (low smearing since the particles barely deform). Once again, in Fig. 14, it is possible to see that the rotational velocity after the angular impact is higher in magnitude than the perpendicular restitution velocity of the normal impact. It was observed experimentally between the fins that the first layer in contact for the substrate was successfully deposited whereas the particle-particle impact did not create a coating (Fig. 7d). Similar to the angled impact of nickel particles, the stainless steel particle's embedment into the substrate is believed to decrease the tangential velocity, which results in a lower restitution rotational momentum that could break the bond whereas the particle impacting on a particle results in a higher restitution velocity (Fig. 14a, right) and consequently a higher momentum that prevents bonding.

## 5. Summary and Conclusion

Different failure modes were identified using SEM analysis of fracture surfaces for pyramidal fin arrays of different materials under shear stress: adhesive mode for aluminum fins, mixed mode for nickel fins, and cohesive mode for the stainless steel fins. The shear strength of materials was also quantified, with the nickel fins (spherical powders) having significantly higher shear strength than the aluminum or stainless steel fins (cylindrical powders). A finite element analysis was performed using ABAQUS to explain the different failure modes and to gain insight into the physics of the impacts. The PTW constitutive equations were successfully used to model a two-particles impact on a substrate for cylindrical aluminum, spherical nickel and, cylindrical stainless steel particles. It was demonstrated with the model that there are two requirements to have metallurgical bonding; i.e., clean surfaces and intimate metal to metal contact, are met for both the substrate-particle and particle-particle contact for the nickel feedstock powder whereas for the two other materials tested the PEEQ value at the fracture surface is believed not to be high enough to clean the surfaces properly. It has been noticed that the PEEQ pattern is different for a spherical particle than for a cylindrical particle. The spherical particle (nickel) has a ring pattern whereas the cylindrical particles (aluminum and stainless steel) have a concentrated deformation zone at the middle of the substrate impact crater. The PEEQ values found numerically seem to correlate qualitatively with the shear strengths measured experimentally; lower PEEQ was observed for the aluminum impacts and the stainless steel impacts while the highest PEEQ was noticed for the nickel impacts. Furthermore, the numerical analysis of the impact at an angle has demonstrated that



even if the equivalent plastic strain is higher than the values obtained for normal impacts, the particles may not bond because of the angular momentum due to the friction in the direction parallel to the substrate. This explains the trends observed experimentally on whether deposition occurs or not between the pyramidal fin arrays, under the wire mesh mask. Moreover, the tensile stress zone that was observed numerically for angular impacts is not believed to explain the lack of deposition at an angle since in some cases of normal impact (nickel particles on the substrate, for example), a zone of tensile stress was created due to the friction of the material flow on the substrate surface, which is high enough to overcome the compressive stress attributed to compaction.

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## References

1. H. Assadi, F. Gärtner, T. Stoltenhoff, and H. Kreye, Bonding Mechanism in Cold Gas Spraying, *Acta Mater.*, 2003, **51**, p 4379-4394
2. M. Grujicic, J.R. Saylor, D.E. Beasley, W.S. DeRosset, and D. Helfrich, Computational Analysis of the Interfacial Bonding between Feed-Powder Particles and the Substrate in the Cold-Gas Dynamic-Spray Process, *Appl. Surf. Sci.*, 2003, **219**, p 211-227
3. W.-Y. Li, H. Liao, C.-J. Li, H.-S. Bang, and C. Coddet, Numerical Simulation of Deformation Behavior of Al Particles Impacting on Al Substrate and Effect of Surface Oxide Films on Interfacial Bonding in Cold Spraying, *Appl. Surf. Sci.*, 2006, **253**, p 5084-5091
4. T. Hussain, D.G. McCartney, P.H. Shipway, and D. Zhang, Bonding Mechanisms in Cold Spraying: The Contributions of Metallurgical and Mechanical Components, *J. Therm. Spray Technol.*, 2009, **18**(3), p 364-379
5. D. Zhang, P.H. Shipway, and D.G. McCartney, Cold Gas Dynamic Spraying of Aluminum: The Role of Substrate Characteristics in Deposit Formation, *J. Therm. Spray Technol.*, 2005, **14**(1), p 109-116
6. M. Grujicic, C.L. Zhao, W.S. DeRosset, and D. Helfrich, Adiabatic Shear Instability Based Mechanism for Particles/Substrate Bonding in the Cold-Gas Dynamic-Spray Process, *Mater. Des.*, 2004, **25**, p 681-688
7. G. Bae, Y. Xiong, S. Kumar, K. Kang, and C. Lee, General Aspects of Interface Bonding in Kinetic Spraying Coatings, *Acta Mater.*, 2008, **56**(17), p 4858-4868
8. T. Schmidt, F. Gärtner, H. Assadi, and H. Kreye, Development of a Generalized Parameter Window for Cold Spray Deposition, *Acta Mater.*, 2006, **54**, p 729-742
9. G. Bae, S. Kumar, S. Yoon, K. Kang, H. Na, H.-J. Kim, and C. Lee, Bonding Features and Associated Mechanisms in Kinetic Sprayed Titanium Coatings, *Acta Mater.*, 2009, **57**, p 5654-5666
10. P.C. King, G. Bae, S.H. Zahiri, M. Jahedi, and C. Lee, An Experimental and Finite Element Study of Cold Spray Copper Impact onto Two Aluminum Substrates, *J. Therm. Spray Technol.*, 2010, **19**(3), p 620-634
11. V. Lemiale, P.C. King, M. Rudman, M. Prakash, P.W. Cleary, M.Z. Jahedi, and S. Gulizia, Temperature and Strain Rate Effects in Cold Spray Investigated by Smoothed Particle Hydrodynamics, *Surf. Coat. Technol.*, 2014, **254**, p 121-130
12. D. Giraud, Study of the Mechanical and Metallurgical Contributions to Coating-substrate Bonding in Cold Spray for "Aluminum/Polyamide 66" and "Titanium/Ti-6Al-4 V", Ph.D. Thesis, Paris Institute of Technology, 2014
13. D.L. Preston, D.L. Tonks, and D.C. Wallace, Model of Plastic Deformation for Extreme Loading Conditions, *J. Appl. Phys.*, 2003, **93**(1), p 211-220
14. S. Rahmati and A. Ghaei, The Use of Particle/Substrate Material Models in Simulation of Cold-Gas Dynamic-Spray Process, *J. Therm. Spray Technol.*, 2014, **23**(3), p 530-540
15. C.Y. Gao and L.C. Zhang, Constitutive Modelling of Plasticity of fcc Metals Under Extremely High Strain Rates, *Int. J. Plast.*, 2012, **32**-33, p 121-133
16. J.-B. Kim and H. Shin, Comparison of Plasticity Models for Tantalum and a Modification of the PTW Model for Wide Ranges of Strain, Strain Rate, and Temperature, *Int. J. Impact Eng.*, 2009, **36**(5), p 746-753
17. R. Liang and A.S. Khan, A Critical Review of Experimental Results and Constitutive Models for BCC and FCC Metals Over a Wide Range of Strain Rates and Temperatures, *Int. J. Plast.*, 1999, **15**, p 963-980
18. P.C. King, S.H. Zahiri, and M. Jahedi, Microstructural Refinement within a Cold Sprayed Copper Particle, *Metall. Mater. Trans. A*, 2009, **40**(9), p 2115-2123
19. Y. Cormier, P. Dupuis, B. Jodoin, and A. Corbeil, Net Shape Fins for Compact Heat Exchanger Produced by Cold Spray, *J. Therm. Spray Technol.*, 2013, **22**(7), p 1210-1221
20. A. Sova, M. Doubenskaia, S. Grigoriev, A. Okunkova, and I. Smurov, Parameters of the Gas-Powder Supersonic Jet in Cold Spraying Using a Mask, *J. Therm. Spray Technol.*, 2013, **22**(4), p 551-556
21. V. Champagne, D. Helfrich, E. Wienhold, and J. DeHaven, The Development of Nickel-Aluminum Reactive Material by Cold Spray Process, Army Research Laboratory Technical Report ARL-TR-5189, 2010
22. D.Y. Kim, J.J. Park, J.G. Lee, D. Kim, S.J. Tark, S. Ahn, J.H. Yun, J. Gwak, K.H. Yoon, S. Chandra, and S.S. Yoon, Cold Spray Deposition of Copper Electrodes on Silicon and Glass Substrates, *J. Therm. Spray Technol.*, 2013, **22**(7), p 1092-1102
23. P. Dupuis, Y. Cormier, A. Farjam, B. Jodoin, and A. Corbeil, Performance Evaluation of Near-Net Pyramidal Shaped Fin Arrays Manufactured by Cold Spray, *Int. J. Heat Mass Transf.*, 2014, **69**, p 34-43
24. Y. Cormier, P. Dupuis, A. Farjam, B. Jodoin, and A. Corbeil, Additive Manufacturing of Pyramidal Pin Fins: Height and Fin Density Effects Under Forced Convection, *Int. J. Heat Mass Transf.*, 2014, **75**, p 235-244
25. Y. Cormier, P. Dupuis, B. Jodoin, and A. Corbeil, Mechanical Properties of Cold Gas Dynamic-Sprayed Near-Net-Shaped Fin Arrays, *J. Therm. Spray Technol.*, 2015, **24**(3), p 476-488
26. Y. Cormier, P. Dupuis, B. Jodoin, and A. Corbeil, Pyramidal Fin Arrays Performance Using Streamwise Anisotropic Materials by Cold Spray Additive Manufacturing, *J. Therm. Spray Technol.*, 2015, doi:10.1007/s11666-015-0267-6
27. EN 15340, European Standard Method-Determination of Shear Load Resistance of Thermally Sprayed Coatings, 2007, p. A91
28. W.-K. Li, H. Liao, C.-J. Li, G. Li, C. Coddet, and X. Wang, On High Velocity Impact of Micro-sized Metallic Particles in Cold Spray, *Appl. Surf. Sci.*, 2006, **253**, p 2852-2862
29. F.P. Incropera, D.P. DeWitt, T.L. Bergman, and A.S. Lavine, *Fundamentals of Heat and Mass Transfer*, 6th ed., Wiley, New York, 2006
30. P. Trahan, "Corrosion Protection of Friction Stir Welded Al 7075 Panel for use in Aerospace using Cold Gas Dynamic Spray," M.A.Sc. Thesis, University of Ottawa, 2013
31. Dassault Systemes Simulia, ABAQUS Analysis User's Manual, Abaqus 6.12, 2012
32. D.J. Steinberg, "Equation of State and Strength Properties of Selected Materials, Lawrence Livermore National Laboratory," 1996
33. B. Banerjee, "An Evaluation of Plastic Flow Stress Models for the Simulation of High-Temperature and High Strain-rate Deformation of Metals, Cornell University Library," 2005, p 1-43
34. M. Fugate, B. Williams, D. Higdon, K.M. Hanson, J. Gattiker, S. Chen, and C. Unal, Hierarchical Bayesian Analysis and the Preston-Tonks-Wallace Model, Los Alamos National Laboratory Technical Report LA-UR-05-3935, 2005
35. B. Banerjee and A.S. Bhawalkar, An Extended Mechanical Threshold Stress Plasticity Model: Modeling 6061-T6 Aluminum Alloy, *J. Mech. Mater. Struct.*, 2008, **3**(3), p 391-424





36. H.N. Jarmakani, "Quasi-Isentropic and Shock Compression of FCC and BCC Metals: Effects of Grain Size and Stacking-Fault Energy," Ph.D. Thesis, University of California, 2008
37. R.A. Austin and D.L. McDowell, Parameterization of a Rate-Dependent Model of Shock-Induced Plasticity for Copper, Nickel and Aluminum, *Int. J. Plast.*, 2012, **32-33**, p 134-154
38. P.S. Follansbee, *Fundamentals of Strength: Principles, Experiment, and Applications of an Internal State Variable Constitutive Formulation*, 1st ed., Wiley, New York, 2014
39. B. Banerjee, The Mechanical Threshold Stress Model for Various Tempered of AISI, 4340 Steel, *Int. J. Solids Struct.*, 2007, **44**, p 834-859
40. S.R. Chen, M.G. Stout, U.F. Kocks, S.R. MacEwen, and A.J. Beaudoin, Constitutive Modeling of a 5182 Aluminum as a Function of Strain Rate and Temperature, *Int. J. Solids Struct.*, 2007, **44**, p 834-859
41. M.L. Newman, B.J. Robinson, H. Schitoglu, and J.A. Dantzig, Deformation, Residual Stress, and Constitutive Relations for Quenched W319 Aluminum, *Metall. Mater. Trans. A*, 2003, **34(7)**, p 1483-1491
42. T.H. Van Steenkiste, J.R. Smith, and R.E. Teets, Aluminum Coatings via Kinetic Spray with Relatively Large Powder Particles, *Surf. Coat. Technol.*, 2002, **154**, p 237-252
43. F. Meng, H. Aydin, S. Yue, and J. Song, The Effects of Contact Conditions on the Onset of Shear Instability in Cold-Spray, *J. Therm. Spray Technol.*, 2015, **24(4)**, p 711-719
44. C.-J. Li, W.-Y. Li, and H. Liao, Examination of the Critical Velocity for Deposition of Particles in Cold Spraying, *J. Therm. Spray Technol.*, 2006, **15(2)**, p 212-222
45. C.J. Li, H.T. Wang, Q. Zhang, G.J. Yang, W.Y. Li, and H.L. Liao, Influence of Spray Materials and Their Surface Oxidation on the Critical Velocity in Cold Spraying, *J. Therm. Spray Technol.*, 2009, **19(1-2)**, p 95-101
46. K. Kang, S. Yoon, Y. Ji, and C. Lee, Oxidation Dependency of Critical Velocity for Aluminum Feedstock Deposition in Kinetic Spraying Process, *Mater. Sci. Eng. A*, 2008, **486(1-2)**, p 300-307
47. T. Klassen, F. Gärtner, T. Schmidt, J.O. Kliemann, K. Onizawa, K.R. Donner, H. Gutzmann, K. Binder, and H. Kreye, Basic Principles and Application Potentials of Cold Gas Spraying, *Mat.-wiss. u. Werkstofftech.*, 2010, **41(7)**, p 575-584
48. A.D. Kuritsyna, Relation Between Hardness and Resistance to Cold Welding [Antifriction Property] of Metals and Alloys, *Met. Sci. Heat Treat.*, 1959, **1(8)**, p 29-32
49. T. Schmidt, H. Assadi, F. Gärtner, H. Richter, T. Stoltenhoff, H. Kreye, and T. Klassen, From Particle Acceleration to Impact and Bonding in Cold Spraying, *J. Therm. Spray Technol.*, 2009, **18(5-6)**, p 794-808
50. R. Cross, Measurements of the Horizontal Coefficient of Restitution for a Superball and a Tennis Ball, *Am. J. Phys.*, 2002, **70(5)**, p 482-489
51. H. Brody, That's How the Ball Bounces, *Phys. Teach.*, 1984, **22**, p 494-497

## **Chapter 6 - Conclusions and Future Work**

### **6.1 Summary of Results and Main Contributions of this Thesis**

The general purpose of this research work was to improve the manufacturing method of fin arrays currently produced by the industrial partner and to study the potential of this new way to manufacture fin arrays. This was accomplished by establishing specific research objectives that were investigated by conducting several experimental and numerical trials.

The feasibility of depositing fin arrays by using the CGDS process was first verified. An alternative technique to the BEC's current manufacturing method to produce fin arrays was successfully developed by using the CGDS process as an additive manufacturing method by selectively masking the substrate by means of a commercially available steel wire mesh mask. The pyramidal fin arrays were established to be satisfactory for the industrial partner as cold spray parameters which give fin porosity less than 5% and clean passages were found. Indeed, in a point of view of production/feasibility, the industrial partner was satisfied but the performance and the viability of this type of fin arrays were still to be proven.

In order to verify the viability of such fin arrays to replace the continuous fins presently manufactured at BEC, an in-house test apparatus was developed to determine the hydrodynamic and thermal performance of the fin array samples. This setup was found to be accurate and precise. The accuracy was verified against data previously reported by Kays and London [15] for a non-finned surface and was found to agree closely. For the precision, the experiments conducted have a standard deviation less than 5% from the average. Therefore, several tests were performed with this setup to compare the performance of the fin arrays

produced by CGDS with continuous fins. It was found that the novel pyramidal fin arrays described in this work outperform the current fin array configuration used at BEC.

Using the in-house apparatus, the effects of several fin geometric parameters on the array performance were investigated, such as the fin base angle, the height and the fin density. It was found that the fin base angle has a minor influence on the performance whereas the fin density and height have a significant effect on the performance. The effects of varying the fin density and fin height were identified and it was found that there is a drastic increase in the volume based surface efficiency from 12 fins per inch to 16 fins per inch followed by a plateau. It was hypothesized that the increase is due to a change in flow structure while the samples on the plateau have potentially the same flow structure. The global performance by unit volume for the height study is largely similar over the entire range of flow conditions tested. The only exception is the sample of 1.0 mm which is lower in performance for the entire range. According to the experiments, the heat transfer coefficient is similar for height from 1.2 mm to 2.0 mm whereas the 1.0 mm and 2.2 mm samples are significantly different. Therefore, these differences were hypothesized by different flow structures for the 1.0 mm and 2.2 mm samples. Empirical relations linking the Nusselt number, the Reynolds number and the height were developed. Correlations that are connecting the Nusselt number, the Reynolds number and the fin density were also developed. Similarly to what was described in Žukauskas' work [19], flow regimes were identified, and were deduced by a change in slope on the thermal conductance graphs. Furthermore, the post-processing grinding of the fin array top surface (the process used to study the fin base angle) was established to be an efficient way to remove potential inconsistencies on the height which could cause a bypass effect. This

post-processing step does not significantly affect performance, compared to a fin array with similar height tolerances.

Pyramidal fin arrays were also successfully deposited on different bond coats to simulate the manufacturing method developed by BEC for the WMHE, since ultimately these fin arrays will be deposited on these types of bond coat. The fin arrays shear strength deposited on bond coats and on Al6061-T6 were evaluated. It was found that the fin height and the fin density as well as the surface roughness before deposition have no effect on the shear strength whereas the different bond coats materials have an influence. Nevertheless, all measurements for the shear strength were higher than the theoretical shear stress for the application range given by BEC which proves its viability in a real flow.

Near-net shaped pyramidal fin arrays of various materials (aluminum, nickel and stainless steel 304) as well as a streamwise anisotropic fin array sample were deposited using the CGDS process. Since, the cold sprayed pyramidal fin arrays are deposited by additive manufacturing instead of machining the fins from the bulk material, a variation of materials could be made when the high efficient service temperature metal is not needed and can be changed for a lower efficient service temperature material with an increased thermal conductivity to raise the heat exchanger's performance. A proof of concept was achieved for this idea and the performance of multi-material fin array was higher than the single high efficient service temperature fin array. While conducting these tests, an important result for the pyramidal fin arrays produced by CDGS were found; the roughness of the fin and the flow passage have been found to have a significant effect on the heat transfer and pressure loss of these types of fin. Therefore, it should be noted that not only the material is important for pin fin arrays but the roughness of the fin and the flow passage play a significant role as well.



Nevertheless, results obtained show a similar efficiency based on volume for the nickel, the aluminum and the multi-material samples while the stainless steel 304 (high efficient service temperature) is less efficient.

Different failure modes were identified using SEM analysis of fracture surfaces for pyramidal fin arrays of different materials under shear stress: adhesive mode for aluminum fins, mixed mode for nickel fins and cohesive mode for stainless steel fins. The shear strength of materials was also quantified, with the nickel fins having significantly higher shear strength than the aluminum or stainless steel fins. A numerical model to simulate the cold spray particle impact process was developed using finite element analysis to explain the different failure modes and to gain insight into the impact physics. The PTW constitutive equations were successfully used to model impact on a substrate for cylindrical aluminum, spherical nickel and cylindrical stainless steel particles. It was demonstrated with the model that the two requirements to have metallurgical bonding; i.e. clean surfaces and intimate metal to metal contact, are met for both the substrate-particle and particle-particle contact for the nickel feedstock powder whereas for the two other materials tested, the equivalent plastic strain (PEEQ) value at the fracture surface is believed to not be high enough to clean the surfaces properly. The metallurgical bonding for the nickel fins is also deduced from the shear band observed with the SEM while shear bands were not detected for the two others materials. The PEEQ values found numerically seem to correlate qualitatively with the shear strengths measured experimentally; lower PEEQ at the fracture surface was observed for the aluminum impacts and the stainless steel impacts while the highest PEEQ was noticed for the nickel impacts. Furthermore, the numerical analysis of the impact at an angle has demonstrated that even if the equivalent plastic strain is higher than the values obtained for normal impacts, the particles may not bond

because of the angular momentum due to the friction in the direction parallel to the substrate surface. This explains the trends observed experimentally on whether deposition occurs or not between the pyramidal fin arrays, under the wire mesh mask. Finally, it was supposed in the last paper that the deposition between the fins is a result from the particles that get around the fin and then deposit on the substrate but it is worth mentioning that the particles could hit the fin (loss some kinetic energy), bounce back in the stream around the fin and hit the substrate at an angle as well.

## **6.2 Future Work**

Using CGDS as a manufacturing technique for pin fin arrays is a novel method (a patent is pending based on the thesis work [117]) since it was first introduced in this work (Paper #1: Proof of Concept). Therefore, even if six peer-reviewed papers resulted from this thesis, only a portion of the experiments necessary to have the whole picture of this technology was covered in this research work. Consequently, several follow-up studies that could be of interest for further advancements on the topic presented in this thesis are listed below:

1. A few hypotheses in terms of flow structures were given in two papers and therefore, more experiments should be conducted in order to verify them. For example, studies using Particle Image Velocimetry (PIV) should be of interest in order to identify the flow structures encountered with this novel type of fin arrays.
2. Further investigation efforts should be put on the fin surface roughness. By varying the cold spray parameters, it could be interesting to see if the fin surface roughness will be modified. The motivation of this suggestion is if an application requires maximising the heat transfer while the pressure drop increase is not a crucial concern, the surface

roughness could be potentially optimized. Also, Computational Fluid Dynamics (CFD) modeling could be used to simulate the flow around a rough pyramid and therefore, quantify the effect of the roughness on the performances.

3. Several other experiments should be conducted in order to have a product that is ready to be implemented in the industry, especially for the recuperator applications. For the heat exchanger to be used in high temperature applications with the streamwise anisotropic materials, it needs more investigation. Indeed, the paper presented was only a proof of concept that varying the material could improve the heat exchange. The next step for this part of the project is to deposit the pyramidal fin arrays on the bond coat. Once the fins will be deposited, high temperature test should be performed on these fin arrays in order to investigate if there are changes in mechanical properties such as thermal conductivity and shear strength. Therefore, an adapted version of the shear test performed during this work should be designed in order to test these fin arrays at high temperature. On another note, since the recuperator will be ultimately used to recover heat from combustion product, corrosion tests should be performed before the implementation in the industry. These tests should be also conducted under a fatigue perspective since the recuperator will be used in a discontinuous manner. Also, an optimal length of each material to be deposited for given conditions should be investigated by a CFD model.
4. A hypothesis in the last paper was that the flow follows the pyramid edge under the wire mask and therefore, sometimes there is a layer of coating between the fins. Further investigation on that aspect should concentrate on modeling an impinging flow through the mesh with a pyramid as an obstacle to observe by numerical simulation the flow structures of the cold spray jet.

5. For the particle impact simulations, several other aspects could be covered. For example, to date, a method to model the adhesion on the substrate has not yet been developed. Further investigation, in this respect, will give a better representation of the cold spray process. Also, determining the particle temperature by using Fluent and inputting these results in Abaqus could lead to a more realistic model of the cold spray process.
6. The nickel particles were presumed to be metallurgically bonded to the substrate and between them. The presumption was deduced from the high shear strength measured, the shear bands observed and the high equivalent plastic strain (PEEQ) obtained by numerical simulation. However, this hypothesis could be potentially verified by using the technique developed by Price et al. [55] and Hussain et al. [56]. An intermetallic compound of Ni-Al will be formed at the interface where there is a clean metal-metal contact during an annealing process.
7. The difference between the thermal conductivity of a bulk material and of a deposited cold spray coating could be investigated under a Transmission Electron Microscope (TEM) by observing the dislocation density. On the same subject, further study could be performed with the laser flash diffusivity testing apparatus. Indeed, the effect of the cold spray parameters on the thermal conductivity could be performed, and by combining with a TEM analysis, a better understanding of the physical properties of coatings produced with the cold spray process could be acquired by this study. This analysis could also explain the difference that was observed between the two spray systems used for the production of the samples used in this thesis.



## References

- [1] U.S. Department Of Energy (DOE), “The Potential Benefits of Distributed Generation and Rate-Related Issues that may Impede Their Expansion”, 2007.
- [2] A. B. Lovins, *Small is Profitable: The Hidden Economic Benefits of Making Electrical Resources the Right Size*. Rocky Mountain Institute, 2002.
- [3] K. Takahashi, S. Barker, L. Kurdgelashvili, “Policy Options to Support Distributed Resources”, University of Delaware, 2005.
- [4] S. M. Kaplan, F. Sissine, *Smart Grid: Modernizing Electric Power Transmission and Distribution; Energy Independence, Storage and Security; Energy Independence and Security Act ... and Resiliency; Integra*. TheCapitol.Net, Inc., 2009.
- [5] D. Bohn, “Micro Gas Turbine and Fuel Cell - A Hybrid Energy Conversion System with High Potential”, North Atlantic Treaty Organization: Research and Technology Organization (NATO: RTO), 2005.
- [6] J. Tian, T. Kim, T.J. Lu, H.P. Hodson, D.T. Queheillalt, D.J. Sypeck, H.N.G. Wadley, “The Effects of Topology Upon Fluid-Flow and Heat-Transfer Within Cellular Copper Structures”, *Int. J. Heat Mass Transfer*, **47**(14-16), pp. 3171-3186, 2007.
- [7] S. Mahjoob, K. Vafai, “A Synthesis of Fluid and Thermal Transport Models for Metal Foam Heat Exchangers”, *Int. J. Heat Mass Transfer*, **51**(15-16), pp. 3701-3711, 2008.
- [8] Z. Anxionnaz, M. Cabassud, C. Gourdon, P. Tochon, “Heat Exchanger/Reactors (HEX reactors): Concepts, Technologies: State-of-the-Art”, *Chem. Eng. Process.*, **47**(12), pp. 2029-2050, 2008.
- [9] R.L. Shaner, “Heat Exchanger Having Metal Wire Screens, and Method of Making Stack of Screens Therefor”, U.S. Patent 4840228, Jun. 20, 1989.
- [10] J. Chisholm, “Method of Making a Crimped Wire Mesh Heat Exchanger/Sink”, U.S. Patent 4843693, Jul. 4, 1989.
- [11] A. Corbeil, “Study of Small Hydraulic Diameter Media for Improved Heat Exchanger”, M.A.Sc Thesis, University of Ottawa, 2009.
- [12] J. Assad, A. Corbeil, P.F. Richer, B. Jodoin, “Novel Stacked Wire Mesh Compact Heat Exchangers Produced Using Cold Spray”, *J. Therm. Spray Technol.*, **20**(6), pp. 1192-1200, 2011.
- [13] A.P. Alkhimov, A.N. Papyrin, V.F. Kosarev, N.I. Nesterovich, M.M. Shushpanov, “Gas-Dynamic Spraying Method for Applying a Coating”, U.S. Patent 5302414, Apr. 12, 1994.
- [14] F. P. Incropera, D. P. DeWitt, T. L. Bergman, A. S. Lavine, *Fundamentals of Heat and Mass Transfer*. 6th ed., Wiley, 2006
- [15] W. M. Kays, A. L. London, *Compact Heat Exchangers*. 3rd ed., McGraw-Hill, 1984
- [16] Y. Peles, A. Kosar, C. Mishra, C.J. Kuo, B. Schneider, “Forced convective heat transfer across a pin fin micro heat sink”, *Int. J. Heat Mass Transfer*, **48**, pp. 3615-3627, 2005.
- [17] B.A. Jasperson, Y. Jeon, K.T. Turner, F.E. Pfefferkorn, W. Qu, “Comparison of Micro-Pin-Fin and Microchannel Heat Sinks Considering Thermal-Hydraulic Performance and Manufacturability”, *IEEE Trans. Compon. Packag. Technol.*, **33**(1), pp. 148-160, 2001.

- [18] R.K. Shah, D.P. Sekulic, *Fundamentals of Heat Exchanger Design*. John Wiley & Sons, 2003.
- [19] A. Žukauskas, "Heat Transfer from Tubes in Crossflow", *Adv. Heat Transfer*, **8**, pp. 93-160, 1972.
- [20] N. Sahiti, A. Lemouedda, D. Stojkovic, F. Durst, E. Franz, "Performance comparison of pin fin in-duct flow arrays with various pin cross-sections", *Appl. Therm. Eng.*, **26**, pp. 1176-1192, 2006.
- [21] B.A. Brigham, G.J. Van Fossen, "Length to diameter ratio and row number effects in short pin fin heat transfer", *ASME J. Eng. Gas Turbines Power*, **104**, pp. 241-244, 1984.
- [22] E.M. Sparrow, J.W. Ramsey, C.A.C. Altemani, "Experiments on in-line pin fin arrays and performance comparison with staggered arrays", *ASME J. Heat Transfer*, **102**, pp. 44-50, 1980.
- [23] D.E. Metzger, C.S. Fan, S.W. Haley, "Effects of pin shape and array orientation on heat transfer and pressure loss in pin fin arrays", *ASME J. Eng. Gas Turbines Power*, **106**, pp.252-257, 1984.
- [24] C.L. Chapman, S. Lee, B.L. Schmidt, "Thermal Performance of an Elliptical Pin Fin Heat Sink", *ASME J. Eng. Gas Turbines Power*, **106**, pp. 24-31, 1984.
- [25] T.M. Jeng, S.C. Tzeng, "Pressure drop and heat transfer of square pin-fin arrays in in-line and staggered arrangements", *Int. J. Heat Mass Transfer*, **50**, pp. 2364-2375, 2007.
- [26] V.B. Grannis, E.M. Sparrow, "Numerical simulation of fluid flow through an array of diamond-shaped pin fins", *Numer. Heat Transfer, Part A*, **19**, pp. 381-403, 1991.
- [27] G. Tanda, "Heat transfer and pressure drop in a rectangular channel with diamond-shaped elements", *Int. J. Heat Mass Transfer*, **44**, pp. 3529-3541, 2001.
- [28] M.K. Chyu, Y.C. Hsing, V. Natarajan, "Convective Heat Transfer of Cubic Fin Arrays in a Narrow Channel", *J. Turbomach*, **120**, pp. 362-367, 1998.
- [29] Q. Li, Z. Chen, U. Flechtner, H.-J. Warnecke, "Heat transfer and pressure drop characteristics in rectangular channels with elliptic pin fins", *Int. J. Heat Fluid Flow*, **19**, pp. 245-250, 1998.
- [30] K.S. Yang, W.H. Chu, I.Y. Chen, C.C. Wang, "A comparative study of the airside performance of heat sinks having pin fin configurations", *Int. J. Heat Mass Transfer*, **50**, pp. 4661-4667, 2007.
- [31] Z. Chen, Q. Li, D. Meier, H.-J. Warnecke, "Convective heat transfer and pressure loss in rectangular ducts with drop-shaped pin fins", *Heat Mass Transfer*, **33**, pp. 219-224, 1997.
- [32] F. Zhou, I. Catton, "Numerical Evaluation of Flow and Heat Transfer in Plate-Pin Fin Heat Sinks with Various Pin Cross-Sections", *Numer. Heat Transfer, Part A*, **60**, pp. 107-128, 2011.
- [33] A.P. Colburn, "Heat transfer by natural and forced convection", *Eng. Bull. Purdue Univ. Res. Ser.*, **26**(84), pp. 47-50, 1942.
- [34] A.L. London, C.K. Ferguson, "Test results of high-performance heat-exchanger surfaces used in aircraft intercoolers and their significance for gas-turbine regenerator design", *Trans. ASME*, **71**, pp. 17-26, 1949.
- [35] P.G. LaHaye, F.J. Neugebauer, R.K. Sakkuja, "A generalized prediction of heat transfer surfaces", *Trans. ASME J. Heat Transfer*, **96**, pp. 511-517, 1974.

- [36] N. Sahiti, F. Durst, A. Dewan, "Strategy for selection of elements for heat transfer enhancement", *Int. J. Heat Mass Transfer*, **49**, pp. 3392-3400, 2006.
- [37] G.J. Soland, W.M. Mack Jr., W.M. Rohsenow, "Performance ranking of plate-fin heat exchanger surfaces", *Trans. ASME J. Heat Transfer*, **100**, pp. 514-519, 1978.
- [38] R.K. Shah, "Compact heat exchanger surface selection methods", *Proceedings of 6<sup>th</sup> Int. Heat transfer Conference*, Toronto, pp. 193-199, 1978.
- [39] S.W. Chang, T.L. Yang, C.C. Huang, K.F. Chiang, "Endwall heat transfer and pressure drop in rectangular channels with attached and detached circular pin-fin array", *Int. J. Heat Mass Transfer*, **51**, pp. 5247-5259, 2008.
- [40] N. Sahiti, F. Durst, A. Dewan, "Heat transfer enhancement by pin elements", *Int. J. Heat Mass Transfer*, **48**, pp. 4738-4747, 2005.
- [41] M. Iyengar, A. Bar-Cohen, "Design for Manufacturability of SISE Parallel Plate Forced Convection Heat Sinks", *IEEE Trans. Compon. Packag. Technol.*, **24**(2), pp. 150-158, 2001.
- [42] M. Geiger, M. Kleiner, R. Eckstein, N. Tiesler, U. Engel, "Microforming", *CIRP Ann. Manuf. Technol.*, **50**(2), pp. 445-462, 2001.
- [43] Y. Jeon, F.E. Pfefferkorn, "Effect of Laser Preheating Workpiece on Micro end Milling of Metals", *J. Manuf. Sci. Eng.*, **130**(1), pp. 1-9, 2008.
- [44] K.A. Moores, Y. Joshi, "Effect of Tip Clearance on the Thermal and Hydrodynamic Performance of a Shrouded Pin Fin Array", *J. Heat Transfer*, **125**, pp. 999-1006, 2003.
- [45] R. Rezaey, "High Temperature Gas to Liquid Metal Foam and Wire Mesh Heat Exchangers", M.A.Sc Thesis, University of Toronto, 2012.
- [46] P.F. Richer, "Development of Conventional and Nanocrystalline Bond Coats by Cold Gas Dynamic Spraying for Aerospace Thermal Barrier Coating", Ph.D. Thesis, University of Ottawa, 2010.
- [47] J.R. Davis, Ed., *Handbook of Thermal Spray Technology*. ASM International, 2004.
- [48] L. Pawlowski, *The Science and Engineering of Thermal Spray Coatings*. John Wiley & Sons, Chichester, England, 1995.
- [49] M.R. Dorfman, "Thermal Spray Processes", *Adv. Mater. Processes*, pp. 47-49, August 2002.
- [50] E. Irissou, J.-G. Legoux, A.N. Ryabinin, B. Jodoin, C. Moreau, "Review on Cold Spray Process and Technology: Part I-Intellectual Property", *J. Therm. Spray Technol.*, **17**(4), pp. 495-516, 2008.
- [51] P.F. Richer, M. Yandouzi, G. Berubé, M. Brochu, A. Zúñiga, B. Jodoin, "Development and Performance of Nanocrystalline Bond Coats Manufactured by Cold Gas Dynamic Spraying for Gas Turbine Engines", *Proceedings of the 2008 Canadian Society of Mechanical Engineering Forum (CSME 2008)*, Ottawa, Ontario, 2008.
- [52] H. Assadi, F. Gärtner, T. Stoltenhoff, H. Kreye, "Bonding Mechanism in Cold Gas Spraying", *Acta Mater.*, **51**, pp. 4379-4394, 2003.
- [53] M. Grujicic, C.L. Zhao, W.S. DeRosset, D. Helfrich, "Adiabatic shear instability based mechanism for particles/substrate bonding in the cold-gas dynamic-spray process", *Mater. Des.*, **25**, pp. 681-688, 2004.
- [54] W.-Y. Li, C.-J. Li, H. Liao, "Significant influence of particle surface oxidation on deposition efficiency, interface microstructure and adhesive strength of cold-sprayed copper coatings", *Appl. Surf. Sci.*, **256**, pp. 4953-4958, 2010.

- [55] T.S. Price, P.H. Shipway, D.G. McCartney, E. Calla, D. Zhang, "A Method for Characterizing the Degree of Inter-Particle Bond Formation in Cold Sprayed Coatings", *J. Therm. Spray Technol.*, **16**(4), pp. 566-570, 2007.
- [56] T. Hussain, D.G. McCartney, P.H. Shipway, D. Zhang, "Bonding Mechanisms in Cold Spraying: The Contributions of Metallurgical and Mechanical Components", *J. Therm. Spray Technol.*, **18**(3), pp. 364-379, 2009.
- [57] F.A. Calvo, A. Urena, J.M.G. Desalazar, F. Molleda, "Special Features of the Formation of the Diffusion Bonded Joints Between Copper and Aluminum", *J. Mater. Sci.*, **23**(6), pp. 2273-2280, 1988.
- [58] I. Manna, J.D. Majumdar, "Enhanced Kinetics of Diffusion Coating of Aluminum on Copper by Boundary Diffusion", *J. Mater. Sci. Lett.*, **12**(12), pp. 920-922, 1993.
- [59] Y. Funamizu, K. Watanabe, "Interdiffusion in Al-Cu System", *Trans. Jpn. Inst. Met.*, **12**(3), pp. 147-152, 1971.
- [60] S. Guetta, M. Berger, F. Borit, V. Guipont, M. Jeandin, M. Boustie, Y. Ichikawa, K. Sakaguchi, K. Ogawa, "Influence of Particle Velocity on Adhesion of Cold-Sprayed Splats", *J. Therm. Spray Technol.*, **18**(3), pp. 331-342, 2009.
- [61] Y. Xiong, K. Kang, G. Bae, S. Yoon, C. Lee, "Dynamic amorphization and recrystallization of metals in kinetic spray process" *Appl. Phys. Lett.*, **92**, pp. 1-3, 2008.
- [62] Y. Zou, W. Qin, E. Irissou, J.-G. Legoux, S. Yue, J. A. Szpunar, "Dynamic recrystallization in the particle/particle interfacial region of cold-sprayed nickel coating: Electron backscatter diffraction characterization" *Scr. Mater.*, **61**, pp. 899-902, 2009.
- [63] P.C. King, G. Bae, S. Zahiri, M. Jahedi, C. Lee, "An Experimental and Finite Element Study of Cold Spray Copper Impact onto Two Aluminum Substrates", *J. Therm. Spray Technol.*, **19**(3), pp. 620-634, 2010.
- [64] J. Villafuerte, *Modern Cold Spray: Materials, Process and Applications*. Springer International Publishing, 2015.
- [65] T. Samson, D. MacDonald, R. Fernández, B. Jodoin, "Effect of Pulsed Waterjet Surface Preparation on the Adhesion Strength of Cold Gas Dynamic Sprayed Aluminum Coatings", *J. Therm. Spray Technol.*, In press: DOI 10.1007/s11666-015-0261-z.
- [66] M. Perton, S. Costil, W. Wong, D. Poirier, E. Irissou, J.-G. Legoux, A. Blouin, S. Yue, "Effect of Pulsed Laser Ablation and Continuous Laser Heating on the Adhesion and Cohesion of Cold Sprayed Ti-6Al-4V Coatings", *J. Therm. Spray Technol.*, **21**(6), pp. 1322-1333, 2012.
- [67] G. Archambault, "Lay-up Moulding of a Carbon Fiber Reinforced Polymer Composite on a Cold Sprayed Metallic Layer", M.A.Sc. Thesis, University of Ottawa, 2015.
- [68] F. Gärtner, T. Stoltenhoff, T. Schmidt, H. Kreye, "The Cold Spray Process and its Potential for Industrial Applications", *J. Therm. Spray Technol.*, **15**(2), pp. 223-232, 2006.
- [69] T. Schmidt, F. Gärtner, H. Assadi, H. Kreye, "Development of a Generalized Parameter Window for Cold Spray Deposition", *Acta Mater.*, **54**, pp. 729-742, 2006.
- [70] T. Klassen, F. Gärtner, T. Schmidt, J.O. Kliemann, K. Onizawa, K.R. Donner, H. Gutzmann, K. Binder, H. Kreye, "Basic Principles and Application Potentials of Cold Gas Spraying", *Mat.-wiss. u. Werkstofftech.*, **41**(7), pp. 575-584, 2010.



- [71] G. Bae, Y. Xiong, S. Kumar, K. Kang, C. Lee, "General Aspects of Interface Bonding in Kinetic Spraying Coatings", *Acta Mater.*, **56**(17), pp. 4858-4868, 2008.
- [72] G. Bae, S. Kumar, S. Yoon, K. Kang, H. Na, H.-J. Kim, C. Lee, "Bonding Features and Associated Mechanisms in Kinetic Sprayed Titanium Coatings", *Acta Mater.*, **57**, pp. 5654-5666, 2009.
- [73] V. Lemiale, P.C. King, M. Rudman, M. Prakash, P.W. Cleary, M.Z. Jahedi, S. Gulizia, "Temperature and Strain Rate Effects in Cold Spray Investigated by Smoothed Particle Hydrodynamics", *Surf. Coat. Technol.*, **254**, pp. 121-130, 2014.
- [74] D. Giraud, "Study of the Mechanical and Metallurgical Contributions to Coating-substrate Bonding in Cold Spray for « Aluminium/Polyamide 66 » and « Titanium/Ti-6Al-4V »", Ph.D. Thesis, Paris Institute of Technology, 2014.
- [75] P.S. Follansbee, U.F. Kocks, "A Constitutive Description of the Deformation of Copper Based on the Use of the Mechanical Threshold Stress as an Internal State Variable", *Acta Mater.*, **36**, pp. 81-93, 1987.
- [76] W. Tong, R.J. Clifton, "Pressure-Shear Impact Investigation of Strain Rate History Effects in Oxygen-Free High-Conductivity Copper", *Mech. Phys. Solids*, **40**, pp. 1251-1294, 1991.
- [77] S. Huang, R.J. Clifton, "Macro and Micro-Mechanics of High Velocity Deformation and Fracture", *Proceedings of IUTAM Symposium on MMMHVDF*, Tokyo, Japan, 1985.
- [78] W.J. Murphy, A. Higginbotham, G. Kimminau, B. Barbrel, E.M. Bringa, J. Hawreliak, R. Kodama, M. Koenig, W. McBarron, M.A. Meyers, B. Nagler, N. Ozaki, N. Park, B. Remington, S. Rothman, S.M. Vinko, T. Whitcher, J.S. Wark "The Strength of Single Crystal Copper Under Uniaxial Shock Compression at 100 GPa", *J. Phys.*, **22**, pp. 1-6, 2010
- [79] E.M. Bringa, K. Rosolankova, R.E. Rudd, B.A. Remington, J.S. Wark, M. Duchaineau, D.H. Kalantar, J. Hawreliak, J. Belak, "Shock Deformation of Face-Centred-Cubic Metals on Subnanosecond Timescales", *Nat. Mater.*, **5**(10), pp. 805-809, 2006.
- [80] S. Rahmati, A. Ghaei, "The Use of Particle/Substrate Material Models in Simulation of Cold-Gas Dynamic-Spray Process", *J. Therm. Spray Technol.*, **23**(3), pp. 530-540, 2014.
- [81] C.Y. Gao, L.C. Zhang, "Constitutive Modelling of Plasticity of fcc Metals Under Extremely High Strain Rates", *Int. J. Plast.*, **32-33**, pp. 121-133, 2012.
- [82] J.-B. Kim, H. Shin, "Comparison of Plasticity Models for Tantalum and a Modification of the PTW Model for Wide Ranges of Strain, Strain Rate, and Temperature", *Int. J. Impact Eng.*, **36**(5), pp. 746-753, 2009.
- [83] R. Liang, A.S. Khan, "A Critical Review of Experimental Results and Constitutive Models for BCC and FCC Metals Over a Wide Range of Strain Rates and Temperatures", *Int. J. Plast.*, **15**, pp. 963-980, 1999.
- [84] P.C. King, S.H. Zahiri, M. Jahedi, "Microstructural Refinement within a Cold Sprayed Copper Particle", *Metall. Mater. Trans. A*, **40**(9), pp. 2115-2123, 2009.
- [85] D.L. Preston, D.L. Tonks, D.C. Wallace, "Model of Plastic Deformation for Extreme Loading Conditions", *J. Appl. Phys.*, **93**(1), pp. 211-220, 2003.

- [86] W.Y. Li, "Study on Effect of Particle Parameters on Deposition Behavior, Microstructure Evolution and Properties in Cold Spraying", Xi'an Jiotong University, China, 2005.
- [87] A. Papyrin, "Cold Spray Technology", *Adv. Mater. Processes*, **159**(9), pp. 49-51, 2001.
- [88] ASTM F2792-10e1, "Standard terminology for additive manufacturing", ASTM International, 2010.
- [89] G.N. Levy, R. Schindel, J.P. Kruth, "Rapid manufacturing and rapid tooling with layer manufacturing (LM) technologies: state of the art and future perspectives", *CIRP Ann. Manuf. Technol.*, **52**, pp. 589-609, 2003.
- [90] J.P. Kruth, M.C. Leu, T. Nakagawa, "Progress in additive manufacturing and rapid prototyping", *CIRP Ann. Manuf. Technol.*, **47**, pp. 525-540, 1998.
- [91] S.H. Huang, P. Liu, A. Mokasdar, L. Hou "Additive manufacturing and its societal impact: a literature review", *Int. J. Adv. Manuf. Technol.*, **67**, pp. 1191-1203, 2013.
- [92] J.P. Kruth, M.C. Leu, T. Nakagawa, "Material incress manufacturing by rapid prototyping techniques", *CIRP Ann. Manuf. Technol.*, **40**, pp. 603-614, 1991.
- [93] D. Ahn, J.-H. Kweon, S. Kwon, J. Song, S. Lee, "Representation of Surface Roughness in Fused Deposition Modeling", *J. Mater. Process. Technol.*, **209**, pp. 5593–5600, 2009.
- [94] L. Laperrière, G. Reinhart, *CIRP Encyclopedia of Production Engineering*, Springer Berlin Heidelberg, 2014.
- [95] S. Das, "Physical Aspects of Process Control in Selective Laser Sintering of Metals", *Adv. Eng. Mater.*, **5**(10), pp. 701–711, 2003.
- [96] L.E. Weiss, F. B. Prinz, "Method for Fabrication of Three Dimensional Articles", U.S. Patent 5301415, 1994
- [97] R. Merz, F.B. Prinz, K. Ramaswami, M. Terk, L.E. Weiss, Shape Deposition Manufacturing, *Proceedings of the Solid Freeform Fabrication Symposium*, 1994
- [98] D.Y. Kim, J.J. Park, J.G. Lee, D. Kim, S.J. Tark, S. Ahn, J.H. Yun, J. Gwak, K.H. Yoon, S. Chandra, S.S. Yoon, "Cold Spray Deposition of Copper Electrodes on Silicon and Glass Substrates", *J. Therm. Spray Technol.*, **22**(7), pp. 1092–1102, 2013.
- [99] O.N. Şara, "Performance Analysis of Rectangular Ducts with Staggered Square Pin Fins", *Energy Convers. Manage.*, **44**, pp. 1787–1803, 2003.
- [100] S.A. Lawson, A.A. Thrift, K.A. Thole, A. Kohli, "Heat Transfer from Multiple Row Arrays of Low Aspect Ratio Pin Fins", *Int. J. Heat Mass Transfer*, **54**, pp. 4099–4109, 2011.
- [101] H.W. Coleman, W.G. Steele, *Experimentation and Uncertainty Analysis for Engineers*. John Wiley & Sons, 1999.
- [102] EN 15340, "European Standard Method-Determination of Shear Load Resistance of Thermally Sprayed Coatings", p. A91, 2007.
- [103] M. Yandouzi, A.J. Böttger, R.W.A. Hendrikx, M. Brochu, P.F. Richer, A. Charest, B. Jodoin, "Microstructure and mechanical properties of B<sub>4</sub>C reinforced Al-based matrix composite coatings deposited by CGDS and PGDS processes", *Surf. Coat. Technol.*, **205**, pp. 2234-2246, 2010.
- [104] TA Instruments, "Thermophysical Properties", 2013.
- [105] K. Shinzato, T. Baba, "A Laser Flash Apparatus for Thermal Diffusivity and Specific Heat Capacity Measurements", *J. Therm. Anal. Calorim.*, **64**, pp. 413–422, 2001.

- [106] V.K. Champagne, *The cold spray materials deposition process: Fundamentals and application*, 1st ed., CRC Press, Boca Raton, Florida, 2007.
- [107] D. Seo, K. Ogawa, K. Sakaguchi, N. Miyamoto, Y. Tsuzuki, "Parameter Study Influencing Thermal Conductivity of Annealed Pure Copper Coatings Deposited by Selective Cold Spray", *Surf. Coat. Technol.*, **206**, pp. 2316-2324, 2012.
- [108] S. Kikuchi, S. Yoshino, M. Yamada, M. Fukumoto, K. Okamoto, "Microstructures and Thermal Properties Cold-Sprayed Cu-Cr Composite Coatings", *J. Therm. Spray Technol.*, **22**(6), pp. 926-931, 2013.
- [109] E. Litovsky, J.I. Kleiman, M. Shagalov, R.B. Heimann, "Measurement of the Thermal Conductivity of Cold Gas Dynamically Sprayed Alumina-reinforced Aluminum Coatings between  $-150^{\circ}\text{C}$  and  $+200^{\circ}\text{C}$ . New Test Method and Experimental Results", *Surf. Coat. Technol.*, **242**, pp. 141-145, 2014.
- [110] Dassault Systemes Simulia, ABAQUS Analysis User's Manual, Abaqus 6.12, 2012.
- [111] B. Banerjee, "An Evaluation of Plastic Flow Stress Models for the Simulation of High-Temperature and High Strain-rate Deformation of Metals", Cornell University Library, pp. 1-43, 2005.
- [112] The Welding Institute, <http://www.twi-global.com/technical-knowledge/faqs/structural-integrity-faqs/faq-what-is-reduced-integration-in-the-context-of-finite-element-analysis/>, consulted September 2015.
- [113] W.-K. Li, H. Liao, C.-J. Li, G. Li, C. Coddet, X. Wang, "On High Velocity Impact of Micro-sized Metallic Particles in Cold Spray", *Appl. Surf. Sci.*, **253**, pp. 2852-2862, 2006.
- [114] ABAQUS Inc., Adaptive Meshing and Distortion Control, ABAQUS/Explicit: Advanced Topics, 2005.
- [115] ABAQUS Inc., Contact Modeling, ABAQUS/Explicit: Advanced Topics, 2005.
- [116] P. Trahan, "Corrosion Protection of Friction Stir Welded Al 7075 Panel for use in Aerospace using Cold Gas Dynamic Spray", M.A.Sc. Thesis, University of Ottawa, 2013.
- [117] A. Corbeil, P. Richer, B. Jodoin, Y. Cormier, P. Dupuis, "Fabrication of Three-dimensional Heat Transfer Enhancing Features on a Substrate", U.S. Patent US20150197858, 2013.

# Appendices

The Hugoniot energy and pressure as well as the Mie-Grüneisen equation expressed in the linear equation of state form shown in section 4.6.1 are demonstrated in these appendices.

## A – Hugoniot Energy

In this section, the steps to have the Hugoniot energy starting from the energy conservation are presented.

Therefore, the energy conservation equation given in section 4.6.1 is defined as:

$$0 = \frac{\dot{M}}{A} (h_1 - h_0 + \frac{1}{2} (u_s - u_m)^2 - \frac{1}{2} u_s^2) \quad (\text{A.1})$$

The enthalpy (h) could be substituted by:

$$h = e + \frac{P}{\rho} \quad (\text{A.2})$$

which results to:

$$0 = \frac{\dot{M}}{A} (e_1 + \frac{P_1}{\rho_1} - e_0 - \frac{P_0}{\rho_0} + \frac{1}{2} (u_s - u_m)^2 - \frac{1}{2} u_s^2) \quad (\text{A.3})$$

From the mass conservation (equation 4.18) and developing the expression  $(u_s - u_m)^2$ , the equation could be rewritten as:

$$0 = \rho_1 (u_s - u_m) \frac{P_1}{\rho_1} - \rho_0 u_s \frac{P_0}{\rho_0} + \rho_0 u_s (e_1 - e_0 - u_s u_m + \frac{1}{2} u_m^2) \quad (\text{A.4})$$

$$P_0 u_s - P_1 (u_s - u_m) = \rho_0 u_s (e_1 - e_0 - u_s u_m + \frac{1}{2} u_m^2) \quad (\text{A.5})$$

By using the momentum conservation (equation 4.19) and the mass conservation (equation 4.18):

$$P_0 u_s + (\rho_0 u_s u_m + P_0) (u_m - u_s) = \rho_0 u_s (e_1 - e_0 - u_s u_m + \frac{1}{2} u_m^2) \quad (\text{A.6})$$



$$(\rho_0 u_s u_m^2 + P_0 u_m - \rho_0 u_s^2 u_m) = \rho_0 u_s (e_1 - e_0 - u_s u_m + \frac{1}{2} u_m^2) \quad (\text{A.7})$$

$$(P_0 u_m) = \rho_0 u_s (e_1 - e_0 - \frac{1}{2} u_m^2) \quad (\text{A.8})$$

By assuming, the value of  $P_0$  is negligible compared to  $P_1$ ,  $e_0$  is negligible compared to  $e_1$  and using mass conservation:

$$0 = \rho_0 u_s (e_1 - \frac{1}{2} (u_m)^2) \quad (\text{A.9})$$

$$0 = \rho_0 u_s e_1 - \frac{1}{2} \rho_0 u_s u_m^2 \quad (\text{A.10})$$

The expression  $\rho_0 u_s u_m$  is equal to  $P_1$  by the momentum equation, considering  $P_0$  is negligible, therefore:

$$0 = \rho_0 u_s e_1 - \frac{1}{2} P_1 u_m \quad (\text{A.11})$$

The mass equation could be rewritten as, by setting  $\eta = 1 - \rho_0/\rho_1$ :

$$u_m = \eta u_s \quad (\text{A.12})$$

This expression is used in A.11:

$$\rho_0 u_s e_1 = \frac{1}{2} P_1 \eta u_s \quad (\text{A.13})$$

$$\rho_0 e_1 = \frac{1}{2} P_1 \eta \quad (\text{A.14})$$

$$e_1 = \frac{P_1 \eta}{2 \rho_0} \quad (\text{A.15})$$

Finally, by setting the subscript 1 to  $H$ , the equation of Hugoniot energy (4.22) is demonstrated.

$$e_H = \frac{P_H \eta}{2 \rho_0} \quad (\text{A.16})$$

## B – Hugoniot Pressure

In this section, the steps to have the Hugoniot pressure starting from the momentum conservation are shown.

Therefore, the momentum conservation equation given in section 4.6.1 is defined as, if the value of  $P_0$  is assumed to be negligible compared to  $P_1$ :

$$P_1 = \rho_0 u_s u_m \quad (\text{B.1})$$

By using the relationship between  $\eta$ ,  $u_m$  and  $u_s$  (A.12), the equation above could be rewritten as:

$$P_1 = \rho_0 u_s^2 \eta \quad (\text{B.2})$$

The experimental relationship given in section 4.6.1 that expressed  $u_s$  as a linear function of  $u_m$  could be written in the following forms by using equation A.12:

$$u_s = c_0 + s\eta u_s \quad (\text{B.3})$$

$$u_s(1 - s\eta) = c_0 \quad (\text{B.4})$$

$$u_s = \frac{c_0}{(1 - s\eta)} \quad (\text{B.5})$$

Finally, by combining equation B.2 and B.5 as well as by setting the subscript 1 to  $H$ , the equation of Hugoniot pressure (4.23) is demonstrated.

$$P_H = \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} \quad (\text{B.6})$$

## C – Linear Equation of State Form of the Mie-Grüneisen

In this section, the steps required to express the Mie-Grüneisen model in a linear equation of state form are presented.

The goal of this appendix is to write the Mie-Grüneisen in the following form:

$$P = f(\rho_m) + g(\rho_m)e_m \quad (C.1)$$

As shown in section 4.6.1, the Mie-Grüneisen is expressed by the equation below:

$$P_m - P_H = \frac{\Gamma}{V} (E_m - E_H) \quad (C.2)$$

By using specific internal energy instead of internal energy, it could be expressed as:

$$P_m - P_H = \Gamma \rho_1 (e_m - e_H) \quad (C.3)$$

By using the equation demonstrated in Appendix A and B for the Hugoniot energy and pressure respectively as well as the relation from the Grüneisen model ( $\Gamma = \frac{\Gamma_0 \rho_0}{\rho_1}$ ), it is possible to express the Mie-Grüneisen as follows:

$$P_m - \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} = \Gamma_0 \rho_0 (e_m - \frac{P_H \eta}{2\rho_0}) \quad (C.4)$$

$$P_m - \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} = \Gamma_0 \rho_0 (e_m - \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} \frac{\eta}{2\rho_0}) \quad (C.5)$$

$$P_m = \Gamma_0 \rho_0 e_m - \Gamma_0 \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} \frac{\eta}{2} + \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} \quad (C.6)$$

$$P_m = \frac{\rho_0 c_0^2 \eta}{(1 - s\eta)^2} (1 - \frac{\Gamma_0 \eta}{2}) + \Gamma_0 \rho_0 e_m \quad (C.7)$$

The last equation is in the form that is implemented in Abaqus since the hydrostatic pressure is expressed as function of material parameters ( $\rho_0$ ,  $c_0$ ,  $\Gamma_0$  and  $s$ ).