AN EXPERIMENTAL INVESTIGATION OF THE HEAT TRANSFER AND FLUID MECHANICS OF METAL ADDITIVELY MANUFACTURED COMPACT HEAT EXCHANGER FINS

A Dissertation in
Mechanical Engineering
by
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Submitted in Partial Fulfillment of the Requirements for the Degree of
Doctor of Philosophy

May 2020
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ABSTRACT

Metal additive manufacturing (AM) has significant potential to improve heat exchanger performance through incorporation of novel geometries and materials, but there is limited understanding of AM heat exchanger functionality compared with traditional manufactured parts. For example, the selective laser melting (SLM) AM process can replicate the major features of a compact heat exchanger but typically results in large surface roughness that is absent in traditional manufacturing methods. Additional defects and deviations from the design intent can also be apparent in AM formed heat exchangers that result in large performance deviations. Yet, metal AM has design freedom and fast turn-around times that are not available with traditional manufacturing processes. Many industries are taking advantage of metal AM for prototyping and producing end-use parts.

This dissertation explored design considerations and consequences of using metal AM in compact heat exchanger applications. An initial study compared the performance of conventionally-built plate-fin air-liquid crossflow heat exchangers to additively manufactured heat exchanger of similar geometry. The full-size, as-built AM HX had significant surface roughness and some cracking in the air-side louvered fins but was able to replicate all the major geometry in the heat exchanger. Increased heat transfer and airside pressure loss was measured and caused by the additional surface roughness on the fin surfaces.

To gain a better understanding of the impact of surface roughness, single air-side heat exchanger surfaces call offset strip fins (OSF) were tested. Deviation between the additively manufactured geometries and previous correlations for smooth fins were found and further amplified as the surface roughness to fin spacing increased. At high Reynolds number, the AM surface roughness caused additional pressure losses that are not apparent in smooth fin geometries.
Earlier transition from laminar to turbulent-like flow was observed from pressure loss measurements and confirmed from Laser Doppler Velocimetry (LDV) measurements in a true-scale AM OSF array. The roughness on the fin surfaces resulted in instabilities and wake shedding from the upstream fin rows causing higher turbulence levels at lower Reynolds numbers compared with the smooth OSF. However, at very low and very high Reynolds numbers, the turbulence levels for the smooth and rough fins were similar. Therefore, turbulence levels alone are not the cause of increased pressure losses. The additional surface area and shear stresses from roughness elements on the fin surfaces are reasons for higher pressure losses in AM fins.

Altogether, these studies conclude that if AM OSF are used for compact heat exchanger applications, additional pressure loss will exist when compared to traditionally manufactured smooth OSF. To combat this, operating at lower Reynolds numbers helps avoid the high levels of increased pressure losses and creating conformally shaped heat exchangers can increase overall heat transfer surface area. To justify the use of AM for heat exchanger applications, other performance considerations could also be improved such as weight, size, and durability to offset the increased pressure drop.
Table of Contents

List of Figures .......................................................................................................................... vii
List of Tables ............................................................................................................................ xiii
Nomenclature ............................................................................................................................ xiv
General ..................................................................................................................................... xiv
Greek ....................................................................................................................................... xvi
Subscripts ................................................................................................................................. xvii
Accents ..................................................................................................................................... xvii
Acknowledgements .................................................................................................................. xviii

Chapter 1 Introduction ................................................................................................................ 1
  Fundamentals of Heat Exchangers ......................................................................................... 1
  Analysis of Conformal Heat Exchanger Benefit ................................................................. 7
  Additive Manufacturing ....................................................................................................... 10
  Outline of Dissertation ......................................................................................................... 14

Chapter 2 Review of Relevant Literature .................................................................................. 15
  Non-metal Additive Manufactured Heat Exchangers .......................................................... 15
  Metal additive Manufactured Heat Exchangers ................................................................. 17
  Surface Roughness and Metal Additive Manufacturing ..................................................... 20
  Offset Strip Fin Correlations and Fundamentals Physics ..................................................... 23
  Uniqueness of Current Research ......................................................................................... 31

Chapter 3 Design And Evaluation Of An Additively Manufactured Aircraft Heat Exchanger* ......................................................................................................................... 32
  Abstract ................................................................................................................................. 32
  Introduction ............................................................................................................................ 33
  Previous studies .................................................................................................................... 35
  Oil Cooler Design ................................................................................................................ 38
  AM Fabrication and Evaluation ....................................................................................... 43
  Performance Test Rig ........................................................................................................ 47
  Experimental Performance Results .................................................................................... 50
  Pressure Drop ....................................................................................................................... 50
  Heat Transfer ....................................................................................................................... 54
  Conclusions ........................................................................................................................... 55
  Acknowledgments ................................................................................................................ 56

Chapter 4 Additively Manufactured Overall Pressure Loss and Heat Transfer Performance of Offset Strip Fins used in Compact Heat Exchangers ......................................................... 57
Abstract .......................................................................................................................... 57
Introduction .................................................................................................................... 58
Previous Studies ........................................................................................................... 60
Fluid and Heat Transfer Parameter Definitions ......................................................... 62
OFS Geometries Studied ............................................................................................... 65
Experimental Rig, Baseline, and Measurement Uncertainty ...................................... 67
Surface Characterization of Fins .................................................................................. 70
Results ............................................................................................................................. 72
  Constant Spaced OSF ................................................................................................. 73
  Variable Spaced OSF ................................................................................................. 79
Conclusions .................................................................................................................... 87
Acknowledgements ........................................................................................................ 88

Chapter 5 Flow Field Measurements in a Metal Additively Manufactured Offset Strip Fin using Laser Doppler Velocimetry ................................................................. 89

Abstract ......................................................................................................................... 89
Introduction ..................................................................................................................... 90
Previous Studies ........................................................................................................... 92
Offset Strip Fin Designs and Characterization ............................................................. 94
Experimental Configuration ......................................................................................... 98
  LDV Measurements .................................................................................................. 98
  Friction Factor ......................................................................................................... 101
  Measurement Uncertainty ......................................................................................... 101
Results ............................................................................................................................. 102
  Flowfield Measurements ......................................................................................... 104
Conclusions .................................................................................................................... 114
Acknowledgements ........................................................................................................ 115

Chapter 6 Conclusions ................................................................................................. 116

Full Size Heat Exchanger Conversion .......................................................................... 116
Offset Strip Fin Heat Transfer, Pressure Loss, and Flow Field Measurements .......... 117
Recommendations for Future Work ............................................................................... 120

References ..................................................................................................................... 123

Appendix Comparison of Metal Additively Manufactured Louvered Fins from Identical
  CAD Geometry* .......................................................................................................... 131
List of Figures

Figure 1-1: Extruded heat sink of various sizes for electronics cooling [1] ....................... 2

Figure 1-2: Heat exchanger mounted on small aircraft that uses ambient airstream to cool engine oil [2] ........................................................................... 3

Figure 1-3: (a) Cross-sectioned computed tomography scan of oil cooler for small aircraft. (b) Close up view of air-side louvered fins and liquid-side turbulators in a cross-flow configuration. ................................................................. 5

Figure 1-4: Common extended heat transfer surfaces used in compact heat exchangers, typically made from sheet metal and brazed to tube wall [5]. ................................. 6

Figure 1-5: (a) Traditionally shaped square heat exchanger and (b) conformally shape heat exchanger in cross-flow configuration with air flow into page ........................................... 8

Figure 1-6: Conformally shaped heat exchanger with a reduction in thickness (transparent section) to remove the extra material added from the shape change ......................... 9

Figure 1-7: (a) Laser-based powder bed fusion process showing part half built surrounded in powder and powder hopper with re-coated blade. (b) Photograph through SLM machine window of current layer of powder being melted by laser. .......................... 11

Figure 1-8: (a) Additive manufacturing software adding support material, red, to shallow angled surfaces to the part, grey. (b) Material added to support upper feature. ............ 12

Figure 1-9: Traditionally manufactured OSF geometry compared with simplified offset strip fin geometry for additive manufacturing with geometric definitions and schematic based off Manglik and Bergles [9] .......................................................... 13

Figure 2-1: Printed LCM heat exchanger concepts with complex design [3] ....................... 16

Figure 2-2: Ti64 additively manufactured heat exchanger designed and tested by Aire et al. [8] ...................................................................................................................... 18
Figure 2-3: Simulation results comparing traditional tubes versus additively manufactured twisted tubes used in shell and tube heat exchanger [25]. .................................................................19

Figure 2-4: Bifurcating, symmetric flow distribution system (left), “flow distribution tree” (right) [33]......................................................................................................................................................20

Figure 2-5: Comparing as-built to designed Inconel 718 microchannels in vertical (a), horizontal (b), and diagonal (c) orientations [38] ...........................................................................................................22

Figure 2-6: Flow visualization in water tunnel in fully laminar region looking at fin rows 5-8 [56] (top image). Higher Reynolds number flow through entire fin bank [53]. .............26

Figure 2-7: Hotwire measurements of turbulence intensity for plain fin (left) and OSF [53] .............................................................................................................................................................................27

Figure 2-8: Normalized velocity contours of the OSF geometry at a Reynolds number of 2750 (left) Modified with different plate spacing (right) [57]. .............................................28

Figure 2-9: Contours of vorticity from 2D numerical simulation at Re ~ 1018 [63]..............29

Figure 2-10: Vorticity contours on mid plane of offset strip fin geometry with the spectral density distribution of temperature fluctuations [66].................................................................30

Figure 3-1: Laser-based powder bed fusion process simplified showing build volume with part and powder hopper with re-coater blade used to spread of next layer of material (left). View through machine window of heat exchanger building (right). .....................34

Figure 3-2: Interior of conventionally manufactured oil cooler, with liquid and air flow paths indicated .................................................................................................................................38

Figure 3-3: Comparison of the baseline oil cooler (left) and CAD model used for the AM build (right), indicating changes to the manifold walls.........................................................40

Figure 3-4: Close up view of the liquid-side turbulators and air-side louvered fins between the baseline oil cooler (left) and the CAD model used to build the AM HX (right)...........40
Figure 3-5: Comparison of manifold exterior shape between AM (left) and conventional HX (right).................................................................41

Figure 3-6: View of vortex generators in the enhanced second design of the AM oil cooler. .................................................................42

Figure 3-7: Sectioned cube of the interior of the baseline AM HX .........................42

Figure 3-8: The two AM HX designs, with external supports attached to the build plate......43

Figure 3-9: Examples of cracking at the outermost row of air-side fins as well as voids through fin thickness. .........................................................44

Figure 3-10: Close-up views of the liquid-side turbulators, air-side fins, and air-side vortex generators in the AM HX’s.................................................................45

Figure 3-11: Summary and naming convention of all four heat exchangers tested..............47

Figure 3-12: Heat exchanger performance rating test rig. .........................................................48

Figure 3-13: Pitot probe traversed across centerline of heat exchanger in horizontal and vertical directions to understand velocity profiles at heat exchanger inlet. .................49

Figure 3-14: Air-side pressure drop versus air mass flow rate for all oil coolers tested........51

Figure 3-15: Friction factor for the louvered fins on the air-side of the oil coolers tested ......52

Figure 3-16: Water-side pressure drop versus water mass flow rate for the baseline traditional and additive oil coolers.................................................................53

Figure 3-17: Comparing heat rejection for the traditional and additively manufactured oil coolers.........................................................................................55

Figure 4-1: Simplified offset strip fin geometry for additive manufacturing with geometric definitions and schematic based off Manglik and Bergles [9] .........................63
Figure 4-2: Three stainless steel offset strip fin geometries additively manufactured at a 45-degree build angle on build plate. The Variable spaced (VS) fin is showing the most tightly spaced fins from this view.

Figure 4-3: Top view of variable spaced (VS) fin geometry that has a step change in fin spacing every third of the way through the flow path.

Figure 4-4: Sectioned view of experimental rig used to test additively manufactured samples for heat transfer and pressure loss performance.

Figure 4-5: Geometry used to baseline the rig and compare to existing correlations, picture taken before seam was sealed shut.

Figure 4-6: Baseline geometry experimentally tested and compared with friction factor, $f$, correlations, [108]–[110], to verify correct rig set-up.

Figure 4-7: Baseline geometry experimentally tested and compared with Nusselt number correlations, [108], [110], [111], to verify correct rig set-up.

Figure 4-8: Profilometry measurements show different roughness types on upward and downward facing surfaces on the metal offset strip fins.

Figure 4-9: SEM images taken in the center of the downward and upward facing metal AM OSF fin surfaces at two different magnifications.

Figure 4-10: Friction factor of plastic (smooth) additively manufactured offset strip fins.

Figure 4-11: Friction factor of metal additively manufactured (rough) offset strip fins with power law regression.

Figure 4-12: Additional data points taken around Reynolds number of 1200.

Figure 4-13: Non-dimensional $j$ of metal additively manufactured (rough) offset strip fins with power law regression.
Figure 4-14: The overall ratio of nondimensional heat transfer to pressure drop for the two differently spaced OSF, compared with smooth fin correlations [104]. ...........................................79

Figure 4-15: Overall friction factor performance for the variable spaced fins (VS) with the results being nearly identical for both flow directions.................................................................80

Figure 4-16: The contribution of total pressure drop for each section of the VS fins. ...........81

Figure 4-17: Calculated $f$ for each subsection of the VS fin geometry with flow going from tight to wide spaced fins........................................................................................................82

Figure 4-18: Calculated $f$ for each subsection of the VS fin geometry with flow going from wide to tight spaced fins........................................................................................................82

Figure 4-19: Updated VS $f$ calculated from averaging Reynolds and $f$ in each fin section compared with each section’s respective f-factor .......................................................84

Figure 4-20: Updated VS $f$ calculated from averaging Reynolds and $f$ in each fin section compared with the constant spaced fin geometries...................................................84

Figure 4-21: Overall $j$ performance for the variable spaced fins (VS) with the results being nearly identical for both flow directions.................................................................86

Figure 4-22: Goodness factor for all three metal AM coupons .........................................86

Figure 5-1: Simplified offset strip fin with geometric definitions and schematic based off Manglik and Bergles [9] and linear laser path for measurements.................................94

Figure 5-2: Both the metal AM and plastic AM OSF samples were printed from the same geometric model with support material removed by machining........................................95

Figure 5-3: Profilometer measurements show different roughness types on upward and downward facing surfaces on the metal offset strip fins.................................................97
Figure 5-4: (upper) Experimental set-up of additively manufactured offset strip fins with LDV beams. (lower) Showing scale of measurement volume compared to scale of fin geometry. 99

Figure 5-5: A total sample count of 15,000 velocity measurements are required to be within 1% 100

Figure 5-6: Friction factor measured for the metal AM rough OSF and the smooth OSF compared with correlations from Mochizuki et al. [104] 103

Figure 5-7: The difference in measured $f$ between the smooth and rough OSF geometries. 103

Figure 5-8: Resin-based (smooth) AM OSF geometry tested showing non-dimensional wake distance 105

Figure 5-9: Wake profiles of time averaged axial velocity and turbulence intensity of the (a) smooth and (b) AM rough OSF geometries at Reynolds number = 300. 106

Figure 5-10: Wake profiles of time averaged axial velocity and turbulence intensity of the (a) smooth and (b) AM rough OSF geometries at Reynolds number = 1000. 108

Figure 5-11: Comparison of wake profiles at row 3 and 17 for the rough and smooth OSF at Reynolds number of 1000. 109

Figure 5-12: (a) The row-by-row velocity and (b) turbulence intensity for the smooth OSF, with (c) the velocity histograms at selected rows for a Reynolds number of 900. 111

Figure 5-13: The row-by-row velocity and turbulence intensity for the metal additively manufactured rough OSF. 113

Figure 0-1: Internal cross section of louvered heat exchanger fins from two different heat exchangers, bottom right fin of Build I was removed for inspection. 132

Figure 0-2: Upskin and downskin surfaces of the louvered fins from build II 132
List of Tables

Table 1-1: Change in normalized heat exchanger performance when frontal area is increased by 50% with conformal shaping. ................................................................. 9

Table 3-1: Size, weight, and design information for the heat exchangers................................. 46

Table 4-1: Measured surface roughness of additively manufactured fins ................................. 71

Table 5-1: Fin dimension measurements of manufactured OSF geometries ............................ 96

Table 5-2: Measured surface roughness of additively manufactured fins ............................... 98
Nomenclature

General

A  Surface area
AM  additively manufactured or additive manufacturing
BAM  baseline additively manufactured
BTM  baseline traditional manufactured
°C  degree Celsius
CAD  computer aided design
CFD  computation fluid dynamics
$C_p$  specific heat
CT  computed tomography
d_B  beam diameter of the LDV measurement volume
DEHS  Di-Ethyl-Hexyl Sebecat
$D_h$  hydraulic diameter
DMLS  direct metal laser sintering
EAM  enhanced additively manufactured
EDM  electric discharge machining
ETM  enhanced traditional manufactured

\[ f = \frac{2\Delta P D_h}{\rho L u_m^2} \]  

DMLS  direct metal laser sintering

$\Delta h$  change in fluid enthalpy
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$j$</td>
<td>Colburn heat transfer factor, $j = StPr^{\frac{2}{3}} = \frac{Nu}{RePr^{\frac{1}{3}}}$</td>
</tr>
<tr>
<td>K</td>
<td>kelvin</td>
</tr>
<tr>
<td>$ke$</td>
<td>kinetic energy</td>
</tr>
<tr>
<td>kg</td>
<td>kilogram</td>
</tr>
<tr>
<td>kPa</td>
<td>kilopascal</td>
</tr>
<tr>
<td>$l$</td>
<td>fin length</td>
</tr>
<tr>
<td>L</td>
<td>overall length of fin array</td>
</tr>
<tr>
<td>$l_B$</td>
<td>length of the beam (length of the LDV measurement volume)</td>
</tr>
<tr>
<td>LES</td>
<td>large eddy simulation</td>
</tr>
<tr>
<td>LDV</td>
<td>laser doppler velocimetry</td>
</tr>
<tr>
<td>m</td>
<td>meter</td>
</tr>
<tr>
<td>mm</td>
<td>millimeter</td>
</tr>
<tr>
<td>N</td>
<td>total velocity counts</td>
</tr>
<tr>
<td>nm</td>
<td>nanometer</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt Number</td>
</tr>
<tr>
<td>OSF</td>
<td>offset strip fin(s)</td>
</tr>
<tr>
<td>P</td>
<td>static pressure [Pascal]</td>
</tr>
<tr>
<td>PBF</td>
<td>powder bed fusion</td>
</tr>
<tr>
<td>$pe$</td>
<td>potential energy</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl Number</td>
</tr>
<tr>
<td>$R_a$</td>
<td>line-based average surface roughness</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number, $Re = \frac{\rho u_m D_h}{\mu}$</td>
</tr>
<tr>
<td>$s$</td>
<td>fin spacing</td>
</tr>
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### Symbols and Units

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$S_a$</td>
<td>area-based average surface roughness</td>
</tr>
<tr>
<td>$S_q$</td>
<td>area-based RMS surface roughness</td>
</tr>
<tr>
<td>$S_z$</td>
<td>area-based peak-valley surface roughness</td>
</tr>
<tr>
<td>SLM</td>
<td>selective laser melting</td>
</tr>
<tr>
<td>St</td>
<td>Stanton Number</td>
</tr>
<tr>
<td>$t$</td>
<td>fin thickness</td>
</tr>
<tr>
<td>T</td>
<td>temperature (°C)</td>
</tr>
<tr>
<td>TI</td>
<td>turbulence intensity</td>
</tr>
<tr>
<td>q</td>
<td>heat flux</td>
</tr>
<tr>
<td>U</td>
<td>overall heat transfer coefficient</td>
</tr>
<tr>
<td>u</td>
<td>velocity [m/s]</td>
</tr>
<tr>
<td>$u'$</td>
<td>fluctuation component of velocity</td>
</tr>
<tr>
<td>VG</td>
<td>vortex generator</td>
</tr>
<tr>
<td>VS</td>
<td>variable spaced fin array</td>
</tr>
<tr>
<td>w</td>
<td>work</td>
</tr>
<tr>
<td>W</td>
<td>watt</td>
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### Greek Symbols

<table>
<thead>
<tr>
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<tbody>
<tr>
<td>$\alpha$</td>
<td>$s/h$</td>
</tr>
<tr>
<td>$\delta$</td>
<td>$t/l$</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>$t/s$</td>
</tr>
<tr>
<td>$\rho$</td>
<td>fluid density</td>
</tr>
<tr>
<td>$\mu$m</td>
<td>-micrometer</td>
</tr>
<tr>
<td>$\mu$</td>
<td>dynamic viscosity</td>
</tr>
</tbody>
</table>
\[\nu\] kinematic viscosity

\[\eta\] efficiency

\[\Delta X\] change in property X

**Subscripts**

avg average value

f fin

i discrete point

in inlet fluid property

m mean value

net total change

out exit fluid property

T total

**Accents**

\[\dot{X}\] rate of change of property X

\[\bar{X}\] averaged property X
Acknowledgements

There are numerous people who have helped me through my graduate school career to whom I owe enormous thanks. I must start by thanking my advisor, Dr. Stephen Lynch, at a professional level who has guided and shaped my career through his mentoring and technical research experience to which I would be much less without. I must also thank Dr. Lynch at a personal level for his kindness, patience, and humor which made the difficult times enjoyable. I am also thankful for the advice from my committee members Dr. Edward Reuzel, Dr. Timothy Simpson, and Dr. Karen Thole. I have learned so much from working with my committee directly on projects and indirectly with their respective students. I hope that I cross paths with my advisor and committee members in future research projects.

I must also thank all the past and present graduate students whose efforts and investments into the ExCCL lab have allowed me to have a more successful and enjoyable time in graduate school. Additionally, the START lab crew was very kind in letting me use their tools and tolerant of the resulting loudness and heat from the research test rig I developed.

There are a few other specific people that I must acknowledge who have helped me technically and personally throughout the years during graduate school. First, I thank Corey Dickman who knows more about the details of metal additive manufacturing than anyone else I have ever met; I learned something new every time we spoke. Second, I thank Phil Irwin, Bill Genet, and Rob McAllister for teaching me their expertise of metalworking and machining. They have all helped me directly and indirectly on research projects and I truly appreciate their patience and open-mindedness on the creative designs I bring them. Third, I would like to thank Professor Liming Chang and Professor Jing Du who were some of the first faculty I interacted with at Penn State. I must also thank Mike Bichnevicius who is a great friend and research collaborator that I mentored and learned from; I wish him the best in his future career. Last, I appreciate the humor
and hands-on knowledge of Jeremiah Bunch who was a great reference for practical engineering design and fun to work with.

Some of my research, unrelated to the work in this document, was funded by Pratt and Whitney which I am appreciative of their guidance and involvement. As a graduate student, I feel very lucky for the information and experiences that Pratt and Whitney have let me be a part of.

“A true teacher would never tell you what to do. But he would give you the knowledge with which you could decide what would be best for you to do.”

— Christopher Pike
Chapter 1

Introduction

The use of heat exchangers for thermal management of high performance machinery and power consuming devices is widespread and critical to keep components within the designed operating temperatures. Like most engineering applications, these devices must be compact, durable, high performing, and as inexpensive as possible. However, these constraints are largely application based, as very easily seen when comparing an automobile and aircraft heat exchanger. For standard automobiles, space and weight requirements are much less strict when compared to aircraft requirements. Additionally, military and satellite applications push the limits of heat exchanger design even further. For this reason, understanding the fundamental physics of heat exchanger operation is critical in order to create smaller, more efficient heat exchangers.

Fundamentals of Heat Exchangers

The simplest type of heat exchanger, known as a heat sink, is commonly found on circuit boards to spread and dissipate the heat generated by electrical components. To further augment heat transfer, these heat sinks can be used in conjunction with fans to dissipate larger quantities of heat. As shown in Figure 1-1, heat sinks can be rather simple or optimized for the intended application.
For many other applications, a fluid used in a thermodynamic cycle will need to be cooled or heated. A common example is in a refrigeration cycle where the refrigerant fluid must reject thermal energy to the surroundings. A heat exchanger in this application would have two working fluids; the air in the surroundings (forced by a fan) and the refrigerant. Similarly, on aircraft, heat exchangers known as oil coolers transfer excess heat from the engine to the surrounding air using lubrication oil as a working fluid. This process also keeps the temperature of the oil low enough so that the oil viscosity is correct for sufficient lubrication.

Two-fluid heat exchangers are commonly found on high performance machines because of the large heat rejection required. Figure 1-2 shows an example of a two-fluid heat exchanger used for small civilian aircraft to remove heat from lubricating oil for the engine, where the heat is transferred to the cooler air that is exhausted to ambient.

Figure 1-1: Extruded heat sink of various sizes for electronics cooling [1]
For each working fluid in the heat exchanger, the first law of thermodynamics, Equation 1, can be used in conjunction with a control volume around the fluid.

\[ \dot{w}_{net} + \dot{q}_{net} = \dot{m}(\Delta ke + \Delta pe + \Delta h) \]  

(1)

However, since the objective of a heat exchanger is to transfer thermal energy from one fluid to another, it is common to neglect changes in potential energy, kinetic energy, and the production or consumption of work, resulting in Equation 2:

\[ \dot{q}_{in} = \dot{m}\Delta h \]  

(2)

This form of the first law of thermodynamics can be used for fluids that change properties or undergo phase change inside the heat exchanger. However, if the fluid does not change phase and the first calorific equation of state is employed, Equation 2 can be simplified further. Equation 3 uses the specific heat of the fluid which can be a function of temperature (see Equation 3) or held constant (see Equation 4):

\[ \dot{q}_{in} = \dot{m} \int_{T_{in}}^{T_{out}} C_p dT \]  

(3)
\[ q_{in} = \dot{m}C_p(T_{out} - T_{in}) \] (4)

Lastly, it is common to assume that there is negligible heat loss to the surroundings, therefore, the change in thermal energy of one fluid must match the change in thermal energy for the second fluid shown in Equation 5.

\[ \dot{m}_2C_{p,2}(T_{2,in} - T_{2,out}) = \dot{m}_1C_{p,1}(T_{1,out} - T_{1,in}) \] (5)

The first law of thermodynamics explains the conversation of energy for any heat exchanger but does not describe the method that heat is transferred. Temperature differences must exist between the fluids for heat transfer to occur which can be calculated from Equation 6.

\[ q = UA \Delta T_{lm} \] (6)

Since the temperature of the fluids change as they pass through the heat exchanger, the log-mean temperature difference, \( \Delta T_{lm} \), is used to calculate an accurate average temperature. The overall heat transfer coefficient, \( U \), depends on the heat exchanger geometry, material, and the fluid motion over the heat exchanger surfaces that form the heat transfer area, \( A \).

The thermodynamic and heat transfer analysis show that a heat exchanger will have highest performance when the temperature difference, specific heat, and flow rate of the two fluids are maximum as well as a maximized surface area and overall heat transfer coefficient. However, the specific heat of a fluid is an intensive property which cannot be changed and high mass flow rates can cause larger pumping power requirements that might be unattainable. From an engineering perspective, the heat exchanger geometry must be designed to maximize heat transfer for the working fluids and their respective temperatures and flow rates. Other factors such as durability, material selection, and cost-effective manufacturing should be taken into account but are not needed for a heat transfer and fluid dynamic analysis.

For two-fluid heat exchangers, there are three major classes which are defined based on the direction of each fluid’s flow relative to the other, known as parallel (co-current), counter (counter-current), and cross-flow heat exchangers. Each class can be subdivided into more specific types.
such as shell and tube, plate fin, spiral, etc., but each type of heat exchanger has its advantages, and the cross-flow heat exchanger is commonly used for high performance applications because it can be compact in size but still has large heat transfer.

The cross-flow heat exchanger geometry is also a good choice when one of the fluids is a liquid while the other is a gas. Typically, gasses, especially air, have a low thermal conductivity and low specific heat compared to liquids. Consequently, the gas side heat transfer surfaces will need augmentation so the gas side does not limit the heat transfer. A means to increase augmentation is shown in Figure 1-3 when comparing the liquid-side geometry to the air-side for a typical cross-flow plate-fin heat exchanger. The liquid flow path does have some turbulators to mix the flow, but the air (gas) side of the heat exchanger has over three times more surface area for heat transfer. This is evidence that there is a large focus to add air-side enhancement for cross-flow heat exchangers which is observed in literature and application.

![Cross-sectioned computed tomography scan of oil cooler for small air craft.](image1)

![Close up view of air-side louvered fins and liquid-side turbulators in a cross-flow configuration.](image2)

**Figure 1-3:** (a) Cross-sectioned computed tomography scan of oil cooler for small air craft. (b) Close up view of air-side louvered fins and liquid-side turbulators in a cross-flow configuration.

Many different air-side geometries have been explored to enhance convective heat transfer with constraints being cost, the ability to manufacture, and the application. Figure 1-4 shows some
of the most commonly used enhanced extended air-side features, although different iterations on these designs have been created. The air-side extended features, known as fins, usually start off as sheets of thin metal and then are bent and formed into the repeated, desired shape. Consequently, the stamping process creates a smooth fin surface with the only roughness coming from burred fin edges from the mold form having wear or surface fouling during operation [3], [4].

To compare the fluid dynamic and heat transfer performance of different fin types, a few non-dimensional parameters are commonly used. These include Reynolds number (see Equation 6), Fanning friction factor (see Equation 7), and the Colburn factor (see Equation 8). The Reynolds number defines the flow regime using a ratio of the inertial to viscous forces. For an offset strip fin (OSF, investigated in this work) a Reynolds number below 1000 is considered laminar, a Reynolds

![Common extended heat transfer surfaces](image)

Figure 1-4: Common extended heat transfer surfaces used in compact heat exchangers, typically made from sheet metal and brazed to tube wall [5].

To compare the fluid dynamic and heat transfer performance of different fin types, a few non-dimensional parameters are commonly used. These include Reynolds number (see Equation 6), Fanning friction factor (see Equation 7), and the Colburn factor (see Equation 8). The Reynolds number defines the flow regime using a ratio of the inertial to viscous forces. For an offset strip fin (OSF, investigated in this work) a Reynolds number below 1000 is considered laminar, a Reynolds
number above ~2000 is considered turbulent, and the flow regime between is transitional. The Fanning friction factor and Colburn factor are a non-dimensional pressure drop and heat transfer performance parameters, respectively.

\[ Re = \frac{\rho u_m D_h}{\mu} \] (7)

\[ f = \frac{2\Delta P D_h}{\rho L u_m^2} \] (8)

\[ j = St Pr^2 = \frac{Nu}{Re Pr^3} \] (9)

**Analysis of Conformal Heat Exchanger Benefit.**

A method to produce a higher performing heat exchanger could involve traditionally manufactured fin types in a conformally shaped heat exchanger to increase the overall heat transfer area. Figure 1-5 presents an example showing how four constraints from other components limit the size of a traditionally shaped heat exchanger in a cross-flow configuration. When the conformally shaped heat exchanger is used to fill the entire design space, the frontal area of the heat exchanger is increased by 50%, resulting in more air-side finned surface area. For this example, with common heat transfer augmentation surfaces, the performance of the traditionally shaped and conformally shaped heat exchangers can be estimated from correlations developed by previous researchers, such as Webb et al. [6], shown in Table 1-1.
The values with green boxes represent an increased performance value for the conformal heat exchanger relative to the traditional design, while the red boxes represent a performance decrease. The grey boxes represent an identical performance to the traditionally shaped heat exchanger, and each column in the table for the conformally shaped heat exchanger has at least one or two variables that are held constant with respect to the traditional heat exchanger. Note that all values are normalized to the traditional heat exchanger, except for Reynolds number to describe the flow regime. With the increased frontal area, heat exchanger parameters will change depending on what variables are kept constant. For example, if the Reynolds number is kept constant, then the overall flow rate, mass, and heat transfer increase proportionally.

If the heat transfer is to be kept constant between the two heat exchangers, then the pressure drop and flow rate are reduced. If the air-side cooling fluid flow rate through the fins must be held constant for system operation, then the Reynolds number and pressure drop are reduced while the heat transfer is increased. The conformally shaped heat exchanger can have a reduced axial air-side...
flow length, Figure 1-6, to match the mass of the traditionally shaped heat exchanger, with the performance shown in Table 1-1.

Table 1-1: Change in normalized heat exchanger performance when frontal area is increased by 50% with conformal shaping.

<table>
<thead>
<tr>
<th></th>
<th>Traditional</th>
<th>Conformal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Re</td>
<td>900</td>
<td>900</td>
</tr>
<tr>
<td></td>
<td>550</td>
<td>600</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td></td>
<td>0.48</td>
<td>0.56</td>
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<tr>
<td></td>
<td>0.67</td>
<td>0.38</td>
</tr>
<tr>
<td></td>
<td>0.48</td>
<td>1.00</td>
</tr>
<tr>
<td>Heat transfer</td>
<td>1.00</td>
<td>1.50</td>
</tr>
<tr>
<td></td>
<td>1.00</td>
<td>1.10</td>
</tr>
<tr>
<td></td>
<td>1.18</td>
<td>0.89</td>
</tr>
<tr>
<td></td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>Flow rate</td>
<td>1.00</td>
<td>1.50</td>
</tr>
<tr>
<td></td>
<td>0.90</td>
<td>1.00</td>
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<tr>
<td></td>
<td>1.50</td>
<td>1.50</td>
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<tr>
<td></td>
<td>1.00</td>
<td>1.20</td>
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<tr>
<td></td>
<td>1.50</td>
<td>1.95</td>
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<td>Mass</td>
<td>1.00</td>
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</tbody>
</table>

With the decreased axial flow length, lower pressure drop can be achieved without a large decrease in heat transfer. Additionally, with nearly twice the air-side flow rate, the heat transfer can be increased without any change in mass or pressure drop. This example highlights the significant of using conformally shaped heat exchangers to obtain improved heat exchanger performance. However, manufacturing a conformally shaped heat exchanger for a particular application is expensive and often not possible using traditional manufacturing processes. On the contrary, metal
additive manufacturing can create complex and optimized shapes for a particular application. However, if metal additive manufacturing is used, the air-side extended surface features must be able to manufacture without defect and perform equal or better than the traditional heat exchanger. This calls for the limitations, opportunities, and consequences of metal additive manufacturing for heat exchanger applications to be explored.

**Additive Manufacturing**

In order for to use additive manufacturing process to benefit heat transfer applications, it must be able to build the heat exchanger out of metal due to its high thermal conductivity and strength. Four main metal additive manufacturing processes exist: powder bed fusion (PBF), directed energy deposition, binder jetting, and sheet lamination. Each AM process has its benefits, but the SLM process typically produces the highest feature resolution with compatible metallic alloys which include aluminum alloys, nickel alloys, titanium alloys, cobalt chrome, and stainless steels [7]. Consequently, the majority of AM heat exchangers studies in the current literature are manufactured from the SLM process.

Figure 1-7 depicts a simplified SLM process, also known as laser-based powder bed fusion (PBF), and shows how a laser melts powdered metal together in the locations based on a 2-dimensional cross-section of a three-dimensional model. Consequently, because the final geometry is based off a 3-dimensional computer aided design (CAD) model, the geometry can be very complex and custom. This could be beneficial to optimize the air-side features for heat transfer and controlling the fluid dynamics.
However, the PBF process does have some feature limitations as shown in Figure 1-8 with unsupported features and surfaces. This occurs when a section of solid material is needed with no previous solid layer directly below. The powdered metal itself cannot support many solid layers of material above it so support material between the build plate and the part, must be added. Furthermore, support material prevents the current layer being built from distorting caused by thermal stresses during the PBF process. For internal heat exchanger passages, support material cannot often be removed and therefore, becomes a flow blockage. Additionally, the angle of a surface that requires supporting material beneath it is also a function of the material that is being used and the size of the feature. This means that designing with the SLM process limitations taken into account for heat exchanger applications is critical for a successful build.
Once the additive process is complete, the most common post processing steps are heat treat to remove residual thermal stress, part removal from the build plate, and sometimes surface finishing or machining. The only required post process step is the removal from the build plate but heat treat can also increase the thermal conductivity of the material as described in [8] which is important for heat transfer applications.

Based on the limitations of the SLM process and previous heat transfer and fluid dynamic studies, the OSF geometry is ideal for consideration in additively manufactured heat exchangers for four main reasons:

- Previous studies and correlations for OSF can be used to compare additive geometries to those made with traditional techniques
- The OSF is relatively simple when compared to other fin geometries so the geometric definition for non-dimensional parameters is well defined.
- The OSF can be additively manufactured with no internal support material in a larger range of build orientations when compared to other fin types.
- Characterization of the geometry is simple, only needing the two sides of the fin surface

The offset strip fin geometry can be further simplified when using the SLM process by removing the additional material on the roof and floor of the flow channel. This additional material, which is
equal to the thickness of the fins, solely exists from the traditional manufacturing process of folding the fins from sheet metal.

Figure 1-9: Traditionally manufactured OSF geometry compared with simplified offset strip fin geometry for additive manufacturing with geometric definitions and schematic based off Manglik and Bergles [9].

The main hesitation of using additive manufacturing for heat exchanger applications is the unknown performance characteristics associated with the manufacturing process. AM surfaces are typically very rough and sometimes have distortion or other defects which impact heat exchanger performance. Existing correlations to predict heat exchanger fin performance are accurate at capturing the complex flow physics of the traditionally manufactured fins, but the correlations fail to predict the performance for AM fins. Understanding the design considerations and consequences of using the selective laser melting additive manufacturing process for compact heat exchanger fins is the focus of this work.
Outline of Dissertation

The dissertation is organized in manuscript format. A review of current and relevant literature is discussed in Chapter 2. Chapter 3, published in the Journal of Applied Thermal Engineering, discusses the process and impact of reproducing a traditionally built heat exchanger with metal additive manufacturing. Chapter 4 examines the pressure loss and heat transfer associated with metal additively manufactured offset strip fins. Chapter 5 investigates and compares the velocity and turbulence levels in a true scale smooth and additively manufactured rough offset strip fin geometry, which is analogous to the traditionally manufactured smooth and additively manufactured rough offset strip fins. Velocity and turbulence measurements in a true-scale additively manufactured offset strip fin array has not been studied previously. Chapter 6 summarizes the major findings and recommends future work.
Chapter 2
Review of Relevant Literature

To current date, the number of studies that combine the use of additive manufacturing for heat exchanger applications is very small when compared to the number of studies focused on heat exchangers or additive manufacturing as their respective fields. Because of this, both additive and heat exchanger studies need to be reviewed to build a foundation to explore the use of additive manufactured heat exchangers. Some unique designs have been created using AM in plastics or metals but in many cases the designs go untested, or cannot be compared to any baseline designs therefore, performance augmentation is unknown.

This section of the document starts by listing the more applied research on full additive heat exchangers and transitions to more fundamental research. The end of this section summarizes the offset strip fin geometry and how performance could change if produced with metal additive manufacturing.

Non-metal Additive Manufactured Heat Exchangers

There are a number of studies that have additively manufactured heat exchangers using polymers or other non-metals, which are relevant because many of the design challenges and processes are analogous to additively manufactured heat exchangers using metal.

Arie et al. [10] laser welded thin (150 μm) polyethylene sheets layer by layer to create a multi-pass air to water heat exchanger which was experimentally tested. Consequently, because the polymer sheets were so thin, the thermal resistance of the polymer only accounted for 3% of the total thermal resistance, making plastics a viable material. Another way to decrease the thermal
resistance impact of the plastic was demonstrated by Hymas et al. [11] by embedding metal fibers into the polymer matrix for the FDM process. This allows for low cost heat exchangers with higher thermal conductivity than a pure plastic part.

Ceramic heat exchangers have also been made through a lithography-based ceramic manufacturing (LCM) technique shown by Scheithauer et al. [12]. Although these geometries were not tested, a fairly complete process from CAD to build to post-processing was described. Figure 2-1 shows two of the concept heat exchanger designs that have large surface area relative to their volume envelope.

![Figure 2-1: Printed LCM heat exchanger concepts with complex design [3]](image)

The most recent review on polymer and polymer composite for heat exchangers by Deisenroth et al. [13] in 2017 concludes that the future work and challenges include the characterization of surface roughness, thermal conductivity improvement, reducing part build time, and the understanding of porosity. Importantly, all of these topics are related to metal additive manufacturing for heat exchanger applications as well.
Metal additive Manufactured Heat Exchangers

Heat exchangers for high performance applications are usually metal because of the high fluid pressures and temperature that exist during operation. For example, an aluminum Cessna aircraft oil cooler operates at an inlet oil temperature and pressure around 107°C and 690 kPa [14] which most additive manufactured plastics could not withstand. In fact, a unique aspect of AM is the ability to create the exact same geometry out of different metals which is explored by Gerstler et al. [15] when the same heat exchanger geometry was made from Aluminum, Cobalt Chrome, Inconel 718, and Titanium 6-4.

Arie et al. [16] has done extensive research in the field of additive manufacturing heat exchangers made from Titanium 6-4 powder. First, optimization using numerical methods showed micro-channel geometries created with AM could increase the gravimetric heat transfer density (Q/MΔT [W/kg-K]) by 60%. Then Arie et al. designed and manufactured a small liquid-to-air heat exchanger that had a 15% to 50% increase in heat transfer coefficient (dependent on flow rate) for the same pressure drop compared to traditional wavy fins [17]. This analysis was performed using a Wilson plot technique which allows for the calculation of the heat transfer coefficient described in [18], [19]. Interestingly, the design was based on a scaled down version of a 12.2 MW heat exchanger for a power plant with modifications for SLM. Then, a larger, optimized heat exchanger manufactured and tested by Arie et al. showed even greater performance augmentation compared to traditional geometries [20]. One conclusion of their work is that further increases in performance could be reached if the metal additive process could produce thinner air-side and manifold features without the consequence of feature failures.

The only part of the heat exchangers from the Arie et al. [16], [17], [20] studies that were not additively manufactured from metal were the water-side headers as shown in Figure 2-2. This is rational since the objective of the study is to understand the air-side enhancements, but
considering the build direction required to have no internal supports for AM, could limit the air-side features when building the entire heat exchanger core with the water-side headers.

Hathaway et al. [21] and Garde et al. [22] improved on an existing oil cooler design used for a construction vehicle by using easy additively manufactured shapes like lenticular tubes to decrease pressure drop. Many test prints were created to understand the feasibility and characterize the outcome of different geometry. Unfortunately, the heat transfer was lower, and the pressure drop was higher than the existing oil cooler. It was concluded that this was due to trapped metal powder in the oil tubes which greatly reduced the effective heat transfer area.

Numerical simulations to optimize heat transfer surfaces specifically for additive manufacturing have also been performed. Elongated, dendrite-like structures from the design tool created by Harertel et al. [23] showed increased performance compared to traditional slotted channels. However some fluid assumptions were made, such as fully developed internal flow, and no modeling of roughness or build lines were taken into account. Similarly, Dede et al. [24] used topology optimization on the extended features for a impingement heat sink application. The optimized geometry performed experimentally similar or slightly better than the baseline but the as-built surface roughness and porosity, which was not modeled, might be a cause of the lower than expected experimental performance. Bernardin et al. [25] ran CFD for a shell and tube heat
exchanger and replaced the straight tubes with a twisted design, shown in Figure 2-3. By doing this, the heat transfer rate increased by 59% and the overall heat transfer coefficient increased by 40%. An experimental test of the manufactured metal heat exchanger is planned, but as-built roughness was not taken into account during the simulations so, experimental results might differ.

Korinko et al. [26] looked in depth at a heat exchanger process tube for a chromatographic process that cycles between -40 and 140°C while under pressure. Because of this, the strength of the tube must be significant so that material pores and defects were examined parallel and perpendicular to the build direction. Machine parameters were studied find the optimal material strength based on laser power, scan pattern, point distance, point exposure, and hatch spacing to reduce the amount of porosity. Lastly, much of the coiled tubing needed support material which was a “labor intensive” process to remove and non-ideal for an additive process.

There have also been studies of metal additive heat sinks with the main focus for electronics cooling. Wong et al. [27]–[29] experimentally tested forced convection heat sinks with diamond, oval, lattice-like, pin, and others geometries. The diamond geometry which allowed for high packing density and constant boundary layer disruption, performed the best when evaluated in heat transfer and pressure drop compared to the other geometries tested.
Surface Roughness and Metal Additive Manufacturing

Since all the surfaces of metal additively manufactured powder bed fusion parts have roughness as a consequence of the manufacturing process, understanding this impact on heat transfer and fluid dynamics is relevant for heat exchangers. This is especially important for mini and microchannels because of the large ratio of roughness to channel diameter. Much of the research and correlations for traditional microchannels cannot be applied without modification to account for rough additively manufactured ones.

Kirsch et al. [30] performed a complete study on additively manufactured micro-channels with the addition of different pin fin configurations and other geometries. An interesting finding was the dependency of pin density on surface roughness; the more densely packed the pins, the larger the surface roughness [31]. This concept could play an important role in geometries outlined in [32]–[34] where the geometry is bifurcating as shown in Figure 2-4. The smallest channels could have larger than expected surface roughness and friction factors if the findings in Kirsch et al. apply.

![Figure 2-4: Bifurcating, symmetric flow distribution system (left), "flow distribution tree" (right) [33].](image-url)
Stimpson et al. [35] found a similar result with rectangular microchannels created using the PBF process. Both the friction factor and Nusselt number increased in these microchannels, however, the friction factor had a larger increase than the increase in Nusselt number. Stimpson et al. also developed correlations to relate the roughness in these channels to friction factor and Nusselt number. They found that a direct relationship between arithmetic mean surface roughness ($R_a$) and sand grain roughness ($k_s$) as shown in Equation 10 [36], provided the best fit to the data.

\[
\frac{k_s}{D_h} = 18 \cdot \frac{R_a}{D_h} - 0.05 \tag{10}
\]

The estimated sand grain roughness is then used in the traditional Colebrook correlation given by Equation 11 [37].

\[
\frac{1}{\sqrt{f_d}} = -2.0 \log \left( \frac{k_s/D_h}{3.7} + \frac{2.51}{Re\sqrt{f_d}} \right) \tag{11}
\]

Another important variable on the outcome of roughness is the part build orientation relative to the build plate. Snyder et al. [38] found that vertically built micro-tubes matched most closely to the design intent and resulted in the lowest surface roughness when compared with the two other build directions, as shown in Figure 2-5. A similar set of circular micro-tubes were also then tested for heat transfer and pressure drop performance. It was found that the Darcy friction factor ($f_d$) was dependent on build orientation while heat transfer, in terms of overall Nusselt number (Nu), was not [39]. Other channel cross-section shapes using diamond and tear drop were created to achieve a more circular final feature and avoid the caved-in top wall of the circular channels. Furthermore, the part’s final geometry and performance outcome was shown to be dependent on material and to vary by 10-20% between builds of the same material [40].
Pakkanen et al. [41] performed a similar study with 10 millimeter circular and tear-drop shaped tubes to understand correlations between build orientation, build defects, and surface roughness. It was found that some channels became slightly elliptic and deviated the most from design when the build angle was below 45 degrees with the most vertically-built channels closely matching the design intent. The channels made with AlSi10Mg had fairly consistent surface roughness ($R_a$) values of about 20 μm with the lowest roughness values on the bottom surface in the horizontal build orientation.

Kirsch and Thole [42] studied 3 different wave-length, PBF wavy fin microchannels with hydraulic diameters around 0.55 millimeters. As found previously the PBF process added large amounts of roughness to the inner channel walls and the as-built hydraulic diameter was about 10% larger than the design. The larger wave-length, wavy fins led to increased friction factors but were also accompanied by Nusselt number augmentation while the short wavelength channels had similar Nusselt number augmentation but greatly increased friction factor. Additional CFD studies concluded that for short wavelength channels, the increase in friction factor is due to the flow separation from the side walls as it accelerates through the peaks and valleys of the channel. The greatest performance increase for these wavy channels was found at Reynolds numbers below about 4000. These wavy channels were then numerically optimized using CFD with three different
objective functions; “minimize the pressure drop between channel inlet and outlet, maximize the heat transfer on the channel walls, and maximize the ratio between heat transfer and pressure drop” [43]. The pressure drop minimization optimization was not accomplished experimentally because the CFD could not predict the flow-field of larger roughness features created during the PBF process. However, the CFD Nusselt number augmentation matched well experimentally because the PBF process could replicate the majority of the modified wall features [43].

**Offset Strip Fin Correlations and Fundamentals Physics**

Since 1980, many review books about compact heat exchangers have detailed performance results for various fin types including the OSF [6], [44]–[47], with Webb [6] having the most detail. The use of offset strip fins (OSF) are common on compact heat exchangers because they have the highest heat transfer and are typically operate at a Reynolds number between 500 to 10,000 [45], although Reynolds numbers up to 100,000 have been experimentally tested [48]. However, because of the high augmentation of heat transfer, the OSF also has high pressure drop.

Performance correlations for OSF geometries have been created and modified as the database of experimental studies and geometries has expanded, with the most recent and widely used correlation by Manglik et al. [9] in 1995. Other studies such as Dong et al. [49] in 2007, Kim et al. [50] in 2011, and Song et al. [51] in 2017 have added additional terms or used numerical studies to increase the accuracy of the correlations.

Manglik et al. [9] combines the regression analysis for both the laminar and turbulent flow regimes to create a continuous function, shown in Equation 12 and Equation 13.

$$j = [0.6522 Re^{-0.5403} \alpha^{-0.1541} \delta^{0.1499} \gamma^{-0.0678}] \times [1 + 5.269 e^{-5} Re^{1.34} \alpha^{0.504} \delta^{0.456} \gamma^{-1.055}]^{0.1} \quad (12)$$
\[
f = [9.6243 Re^{-0.7422} \alpha^{-0.1856} \delta^{0.3053} \gamma^{-2.659}] \times \\
[1 + 7.66e^{-8} Re^{4.429} \alpha^{0.920} \delta^{3.767} \gamma^{0.236}]^{0.1} \quad (13)
\]

Dong et al. [49] uses a regression analysis as well and add an additional term that compares the ratio of the overall fin array length to that of a single fin. This takes into account the effect of the number of OSF rows, shown in Equation 14 and Equation 15.

\[
j = 2.092 Re^{-0.281} \alpha^{-0.739} \delta^{0.972} \gamma^{-0.78} \frac{L}{l}^{-0.497} \quad (14)
\]

\[
f = 2.092 Re^{-0.281} \alpha^{-0.739} \delta^{0.972} \gamma^{-0.78} \frac{L}{l}^{-0.497} \quad (15)
\]

The only study found that compares a metal additive manufactured (SLM) offset fin geometry to the traditional correlations by Manglik is Wong et al. [29]. It was found that the additive OSF compared well with the Manglik correlations for f-factor but was below the correlation for the j-factor, where the lower j-factor was attributed to having an insulated fin tip boundary condition instead of having both bounding walls heated evenly. It should also be noted that the bounding walls were smooth, and the ratio of the roughness to hydraulic diameter was about \( R_a/D_h = 0.007 \). These results would suggest that the roughness on the fins does not have a large impact on the fluid dynamics between the tested Reynolds numbers between 200-2000 because the f-factor matched previous smooth-fin correlations. However, many of the OSF geometries that Kays and London [47] used for correlations had hydraulic diameters that were about one-third of the hydraulic diameter in Wong et al. [29]. This would result in an \( R_a/D_h \) of about 0.02, which may result in larger sensitivity of the flow to roughness.

Related to this, Li et al. [52] studied round microchannels made from glass, silicon, and stainless steel. The glass and silicon tubes were found to be smooth and followed the conventional laminar correlation, \( f \cdot Re = 64 \), with a relative roughness, \( R_a/D_h \), below 1%. However, the stainless steel tubes with relative roughness of up to 4% had \( f \cdot Re \) as high as 87; a 37% increase.
Assuming a typical SLM-process surface roughness, $R_a$, of 20 $\mu$m, a OSF listed in [53] would have a comparable $R_a/D_h$ to the stainless tubes from Li et al. [52]. Webb et al. [54] discussed the effects of burred edges and roughened bounding walls in OSF geometries and concluded that this could increase the friction factor as much as 14% in the turbulent flow regime, but less than 10% for Reynold numbers under 500. To account for this, a new friction factor correlation was proposed for fully developed turbulent flow for an OSF geometry, shown in Equation 16 and Equation 17.

$$f_p = 3.78\text{Re}^{-0.62} \left( \frac{1}{D_h} \right)^{-0.22} \quad (16)$$

where $D_h$ is defined as:

$$D_h = \frac{2(s - t)h}{(s + h) + \frac{ht}{T}} \quad (17)$$

Further fundamental analysis of the OSF has been studied through qualitative flow visualization performed by Mochizuki et al. [53], [55] and Dejong et al. [56]. Both of these studies used a water tunnel with ink injection for visualization, but also performed experiments in a wind tunnel for pressure drop measurements. As shown in Figure 2-6, at very low Reynolds number, the entire fin bank consists of steady laminar flow. A Karman Vortex Street occurs far into the fin bank at a Reynolds number of about 1000. As Reynolds number is increased further, the wake shedding of the fins and the unsteadiness of the flow shifts closer to the first row of fins. At very large Reynolds numbers, turbulent flow exists in nearly the entire fin bank.
Mochizuki et al. [53] also measured the turbulent intensity (TI), defined in Equation 18, in a wind tunnel with the OSF geometry in three different locations; around the 3rd row, midstream, and in the last 3 rows, shown in Figure 2-7.

\[
TI = \frac{\sqrt{u'^2}}{U_m}
\]  

(18)

Figure 2-6: Flow visualization in water tunnel in fully laminar region looking at fin rows 5-8 [56] (top image). Higher Reynolds number flow through entire fin bank [53].
At Reynolds numbers around 1000, the TI is low in the first 3 rows and comparable to a plain fin geometry, but high and equal at the midstream and last row locations. At a Reynolds number of about 6000, the turbulence intensity throughout the fin bank is very similar because of the wake-shedding that occurs even in the first few rows.

Lee et al. [57] used a Laser Doppler Velocimetry (LDV) technique for velocity measurements in the fin core, as shown in Figure 2-8. This allowed for a non-intrusive, yet quantitative measurement of velocity through the OSF core. The goal of their study was to augment heat transfer by modifying the plate spacing so a downstream fin would receive the wake from an upstream plate. Because of this, only a few true OSF geometries were tested with a focus on the enhanced fin banks. However, AM could easily manufacture the enhanced fin spacing where traditional stamping could not.
A similar study was performed by Springer et al. [58] using an LDV technique to study the flow field in a louvered fin geometry used for compact heat exchangers. The study used 20 times scaled up fins to increase the resolution of the velocity measurements and to have accurate measurement positioning. The measurements were able to show that for the tested geometry, the louvers were directing the flow even at low Reynolds numbers, which was not apparent previous computation fluid dynamics (CFD) simulations with reduced rows numbers.

Numerical studies of the OSF geometry or staggered plates have shed insight into the flow physics and heat transfer of this type of geometry. As computation resources have grown, the assumptions and approximations have been able to be reduced. A good example from Patankar et al. [3] shows how a fin with finite thickness can impact the recirculation flow structures behind the fins. Other numerical studies in two dimensional, three dimensional [59]–[62] and unsteady cases, are able to capture some of the major flow structures around the fin, shown in Figure 2-9.

Figure 2-8: Normalized velocity contours of the OSF geometry at a Reynolds number of 2750 (left) Modified with different plate spacing (right) [57].
The k-ε turbulence model seems to predict the flow field best [64] and can predict \( f \) and \( j \) factors reasonably well for smooth fins; however, capturing the transition to turbulence can be a challenge [65]. Pham et al. [66] was one of the first to use large-eddy simulation to capture the evolution of wake shedding in the streamwise direction of offset strip fins. Figure 2-10 shows how the wake shedding and flow instabilities are not prevalent on the most upstream fins but start to increase in magnitude as the latter fins are reached. The wake shedding behavior was quantified by the Strouhal number, Equation 19, to capture the nondimensional frequency of the flow. At first, the wake shedding behind a fin consisted of a single Strouhal number, but as the flow traveled further downstream and the shear layer was destabilized, additional Strouhal numbers were measured. Pham et al. did not find any wake shedding behavior at Reynolds of 1400 but the fins modeled were relatively slender with a length-to-thickness ratio of 80 and pitch-to-thickness ratio of 12.

\[
\text{Strouhal number} = \frac{\omega t}{u}
\] (19)
Pham et al. [66] also reports the relative contributions of friction factor from form drag and wall skin friction. At a Reynolds number of 500, the skin friction is about 40% of the friction factor contribution, but this decrease to 17% at a Reynolds number of 10,000. This sort of information is very difficult to gather from experiments and showcases advantages of running computational simulations. Another example of this was shown by Saidi et al. [67] that found the wavy flow oscillation in their offset strip fin geometry occurred at a frequency of ~68 Hz for a Reynolds number of 1124.
Uniqueness of Current Research

As shown in the reviewed literature, the offset strip fin has been widely studied. The flow field of the OSF is well understood from qualitative flow visualization and quantitative laser-based techniques and numerical simulations. Additionally, there are many available correlations to accurately predict the pressure drop and heat transfer of conventionally manufactured fins based on the geometric definitions. Currently, however, it is unclear what performance effect surface roughness might have on OSF geometries because traditional manufacturing methods do not create appreciable surface roughness. The inherent roughness and dimensional tolerance impacts from metal additive manufacturing for thin fin features are unknown, which limits the ability of designers to use this new fabrication technique for complex heat exchangers.

The following studies add to current literature through experimental measurements of rough AM OSF to determine how the surface roughness impacts the pressure losses and heat transfer in the fin array. The OSF geometries tested are representative of currently used geometries in compact heat exchanger applications. Surface characterization through profilometry and SEM scans show roughness values an order of magnitude larger than traditionally manufactured smooth OSF. Additional velocity measurements in true-scale smooth and rough OSF are performed to gather local information about flow field and turbulence levels between fins. Measurements in true-scale AM OSF have not been shown before in public literature.
Chapter 3

Design And Evaluation Of An Additively Manufactured Aircraft Heat Exchanger*

Abstract

Additive manufacturing (AM) technology has significant potential to improve heat exchanger (HX) performance through incorporation of novel geometries and materials, but there is limited understanding of AM HX functionality relative to conventionally manufactured components. This study compares the performance of conventionally-built plate-fin air-liquid crossflow heat exchangers (i.e., aircraft oil coolers) to additively manufactured heat exchangers of similar geometry. To replicate internal features, three dimensional X-ray computed tomography scans were performed on the conventionally-built heat exchanger. A baseline AM model of the conventional design was designed, as well as an AM model with additional enhancement features on the air-side. The two AM heat exchanger geometries were constructed using a laser-based powder bed fusion process with AlSi10Mg aluminum-alloy powder. Visual inspection of the as-built AM HX indicated significant surface roughness throughout and some cracks in the fin-tube joint only at the edges of the heat exchanger. Overall heat transfer was increased by about 10 percent for the baseline AM and by 14 percent for the enhanced AM heat exchanger when compared to the conventionally built baseline heat exchanger. Measured air-side pressure drop for the AM heat exchangers was double that of the conventionally built baseline heat exchanger. Overall, this study indicates potential for improved heat transfer and demonstrated functionality of AM HX in realistic applications.

**Introduction**

Heat exchangers (HX) are used in many different applications to transfer heat between fluid streams, often to get rid of waste heat generated during a process. In the application considered here, aircraft engines generate waste heat from friction in engine components (e.g., bearings, pistons, other moving parts), which is transferred to engine oil. Oil temperature must be carefully controlled to avoid oil viscosity changes that can cause insufficient lubrication. The heat transferred to the oil is typically dissipated to the airflow around the aircraft engine through a specific type of heat exchanger called an oil cooler.

Aircraft engine oil coolers are typically a brazed plate-fin design due to the compact nature and ease of manufacturing. The plates and fins can be manufactured at high volume by stamping processes, which reduces per-part cost although tooling costs can be high. To achieve a desired performance, the required number of plate-fin components can be estimated and assembled into a single HX. Aircraft oil coolers have been used in the aviation industry for many years and have been optimized for weight, performance, and cost. However, for specialized military and aerospace applications, the performance of the heat exchanger is often the primary design goal. Complex designs can improve performance but are often difficult and/or costly to fabricate with conventional manufacturing techniques. Additive manufacturing (AM) through layer-wise fabrication (such as powder-bed fusion) may allow complex HX designs that increase heat exchanger performance and reduce part weight, while limiting the number of individual components required for the final part, as well as conformal HX geometries for space-limited applications.

The metal additive technique for this study is a laser-based powder bed fusion process described in [68], and is depicted in Figure 3-1. The part is built layer by layer in 2D cross-sections on a metal build platform, using a high power laser to weld a thin layer (typically 20-40 µm) of powdered metal in specific locations according to the solid areas of the 3D CAD model. After each
layer is complete, more powder is transferred to the top of the surface (for this study, by using a flexible recoating blade to drag powder across the part) and the new powder layer is welded to the existing solidified material using the laser. This process repeats until the part is built up to its final height. At the end of the build, the entire part, including the internal channels, is filled with powdered metal. This is suctioned or blown out prior to removal of the part from the machine. Other processes generally required are a stress-relieving heat treatment and removal of the part from the build platform.

![Diagram of laser-based powder bed fusion process](image)

Figure 3-1: Laser-based powder bed fusion process simplified showing build volume with part and powder hopper with re-coater blade used to spread of next layer of material (left). View through machine window of heat exchanger building (right).

Because the additive process is very different than the traditional way that heat exchangers are manufactured, little is currently understood about the design tradeoffs required to build an AM HX and the potential performance enhancement (or reduction) that may occur. The objective in this work was to design, additively manufacture, and evaluate a heat exchanger for light aircraft applications and compare it to a conventionally manufactured model. The conventional HX was scanned via a three-dimensional X-ray computed tomography (CT) technique to develop a baseline CAD model of nearly identical geometry for AM fabrication. The heat exchanger was then fabricated using a laser-based powder bed fusion process and tested in a performance rating rig developed specifically to characterize liquid and air stream heat transfer and pressure drop. Heat transfer enhancement features were then added to the baseline AM heat exchanger design to
investigate the ability to fabricate complex features in AM and assess their impact on performance. Please note that some portions of this paper were presented at the 2017 American Institute for Aeronautics & Astronautics SciTech conference, paper number AIAA2017-0902 [69].

**Previous studies**

Heat exchanger design is largely dictated by the process requirement, environment in which the heat exchanger operates, and available manufacturing techniques. This study is focused on liquid to air cross-flow heat exchangers used in the aviation industry. These heat exchangers must be as compact, lightweight, and as efficient as possible to accommodate size and weight limitations on aircraft [70]. Aircraft heat exchangers are used in fuel-air aftercoolers, air-conditioning radiators, electronics cooling, engine oil coolers, and hydraulic systems [71], [72].

These types of heat exchangers are generally classified as compact heat exchangers (CHE), with well-known variants: plate (PHE), plate-fin (PFHE), printed circuit (PCHE), and spiral heat exchangers (SHE), which all utilize specific features to enhance heat transfer [73]. The fabrication methods used in the manufacturing process for each type of HX directly affect the final cost for the part [71]. A complex HX consisting of a brazed assembly with numerous components with small, intricate features will have high performance but will be expensive to manufacture using conventional methods, especially if only a few parts are needed.

Limitations of conventional manufacturing techniques may allow for additive manufacturing (AM) to become a viable option for creating compact, high-performance heat exchangers. Many different types of AM processes exists; each with their own benefits and weaknesses [68]. However, to use AM for an air-liquid heat exchanger, the AM process must be able to create internal channels, thin and small features, and use a material that is thermally
conductive and durable. With these constraints taken into account, a laser-based power-bed fusion (PBF) process is selected for this study.

Laser-based PBF is starting to find many applications in heat transfer and thermal management. Stimpson, et al. [35] demonstrated the use of PBF AM techniques to fabricate heat sinks with micro-channels having hydraulic diameters of less than 0.5 mm. Heat transfer can be further enhanced in micro-channels when compared to straight smooth channels by making the channels wavy to disrupt the boundary layer [42], or by adding pins [31]. Micro-channels are an effective design feature to increase heat transfer but result in high pressure drops and low flow rates. Furthermore, micro-channels can be significantly impacted by surface roughness, which is a function of layer thickness, material, powder particle size distribution, laser parameters and part orientation among other factors [39]. Stimpson, et al. [36] determined a correlation for heat transfer and pressure drop in additively manufactured micro-channels by accounting for the significant roughness.

The majority of studies involving AM HX have considered heat sinks, which are typically designed to dissipate excess heat through conduction into the heat sink material and convection away from its surface. The overall thermal resistance of a heat sink is reduced by increasing the sink material thermal conductivity, the available surface area for convection, or the convection heat transfer coefficient. Ventola, et al. [74] used PBF AM to intentionally create different amounts of surface roughness on an external heat sink fin. Surface roughness increased heat transfer as much as 40% when compared to a smooth external heat sink fin due to the disruption of the thermal boundary layer, but no impact on pressure drop was presented. Wong, et al. [29] used PBF AM to create novel 3-D heat sinks, and found that the innovative geometries were superior in both heat transfer and pressure drop relative to a cylindrical pin-fin geometry. Dede, et al. [24] used topology optimization to design a pin fin heat sink for impingement cooling, and found slightly better overall performance (accounting for heat transfer and pressure drop) for the optimized geometry relative
to conventional designs. Although this study does not consider topology optimization for heat transfer, it appears to be a promising means of exploiting PBF AM for heat sinks.\cite{23,75--78}, it appears to be a promising means of exploiting PBF AM for heat sinks.

Other heat transfer applications have also recently demonstrated use of AM in novel configurations. Thompson, et al. \cite{79} and Ibrahim, et al. \cite{80} constructed functional oscillating heat pipes using a titanium alloy, taking advantage of the surface roughness and porosity to assist fluid wicking behavior in the heat pipe. Recent work by Arie, et al. \cite{10} demonstrated a functional polymer-based heat exchanger using a line-welding AM technique.

There is increasing interest in understanding how AM technology can impact heat transfer applications for thermal management. However, most studies to date have considered heat sink geometries that involve only one working fluid. These studies are necessary to generate design correlations for heat transfer with AM-fabricated parts \cite{36}, but there are other important considerations for functional heat exchanger devices. Specifically, additive manufacturing might enable the use of moderate to poorly-conductive materials (such as stainless steel) or the elimination of braze joints. Furthermore, the fluid and heat transfer boundary conditions for a cross-flow heat exchanger are complex and difficult to capture except with a fully functional device.

This study presents the first-of-its-kind direct performance assessment of a fully functional air-to-liquid metal heat exchanger manufactured via laser-based PBF, and we compare it to a conventionally manufactured HX design of the same geometry. Although this does not take advantage of the design freedom of AM, it provides a more direct comparison of the benefits and drawbacks of design limitations and expected performance variation for an AM HX.
Oil Cooler Design

The conventionally manufactured oil coolers for this study were supplied by Airflow Systems, Inc. and are used for engine oil cooling on performance racing aircraft. The oil cooler geometry used in this study (see Figure 3-2) is a finned tube crossflow style of construction fabricated from aluminum, with flattened tubes in the liquid-side that draw from and dump to large plenums attached to the tubes. The liquid tubes contain heat transfer enhancement features on the inside to promote convective heat transfer and provide additional heat transfer surface area. On the air-side, louvered fins span between adjacent tubes to promote air mixing with high heat transfer surface area. The fins are connected to the liquid tubes via brazing. The total inlet area on the airflow side is 105.1 cm$^2$, although the airflow open area is only 85.1 cm$^2$ due to the finite liquid tube thickness. The spacing between tubes (fin length, $F_l$) is 6 mm, the fin thickness ($F_t$) is 0.30 mm, and the liquid tube internal height ($T_h$) is 3.2 mm.

![Figure 3-2: Interior of conventionally manufactured oil cooler, with liquid and air flow paths indicated.](image-url)
Two conventionally manufactured oil coolers were provided; one model (20002A) is denoted as the baseline traditionally manufactured (BTM), and a second model (2002X) has air-side enhancements and is denoted as the enhanced traditionally manufactured (ETM). Details of the air-side enhancements in the ETM model are not described to protect company proprietary information.

An X-ray computed tomography (CT) scan was performed on the baseline (BTM) oil cooler to capture the internal and external features. The CT scan had a minimum resolution of 100 µm, which was sufficient to resolve the air-side fins and liquid-side turbulators. From the CT scan, a CAD model of the baseline oil cooler was developed, taking advantage of repeating features. On the air-side, the fin density, fin thickness, louver shape, and open flow area were replicated exactly in the CAD model relative to the CT scan. For the liquid-side, the number, shape, and thickness of turbulators, as well as the tube height and wall thickness, were exactly replicated. The liquid manifold volume, restrictions in the manifold between each plate-fin subassembly, and the open area from the manifold into the liquid tubes were also replicated. This HX is denoted as the baseline additively manufactured model (BAM).

Some features of the HX had to be slightly modified to enable it to be built in the PBF process with minimal post-processing. This is because geometries such as down-facing surfaces below a certain angle with-respect to the substrate, build orientation of the part, internal channels above a certain size, and material islands will not be produced accurately [81]. The issue is that these features cannot be properly supported by the loose powder metal underneath them, and internal supports cannot be accessed for removed. While many argue that “complexity is free” when it comes to AM [82], the reality is that even AM cannot fabricate all of the necessary internal features, particularly metallic components with intricate internal features [83]. Consequently, the internal and external geometry of the traditional oil cooler was slightly altered to accommodate limitations of the PBF AM process.
The major geometry changes needed to create a feasible additively manufactured oil cooler were:

1. Curved walls on the sides of the liquid manifold region were straightened and angled to prevent unsupported surfaces (see Figure 3-3).

2. Air-side and liquid-side extended features were given a constant angle of 15° from vertical in order to compensate for the build orientation (see Figure 3-4) so that they would not require support when the HX was built.

3. The exterior edges of the manifold were squared-off for simplicity and to reduce the number of required external supports when the part was built (see Figure 3-5).

Figure 3-3: Comparison of the baseline oil cooler (left) and CAD model used for the AM build (right), indicating changes to the manifold walls.

Figure 3-4: Close up view of the liquid-side turbulators and air-side louvered fins between the baseline oil cooler (left) and the CAD model used to build the AM HX (right).
A second oil cooler design was also created, to test the ability to 3D print small enhancement features and assess their impact on performance. This final design is denoted as the enhanced additively manufactured model (EAM). The majority of the design was exactly the same as the baseline AM HX, but small vortex generators (VG) were added to the air-side, in the space between fins, as shown in Figure 3-6. Vortex generators were only added to the upward-facing tube surface (in the HX build orientation) since downward-facing VG would require additional support structures. VG were added to every other row of fins, with 5 VG per row in the stream-wise direction (spacing of 18.4 mm between successive VG), all with the same orientation of 45 degrees relative to the flow direction. The VG had a design thickness of 0.36 mm and a peak height of 1.25 mm. The VG were not designed to a specific performance metric, but they were included to assess manufacturability.

Figure 3-5: Comparison of manifold exterior shape between AM (left) and conventional HX (right).
Both the BAM and EAM oil cooler designs were oriented at 45 degrees from the vertical direction on a build plate and evaluated for support structure requirements. External support structures were added to the models to support the mounting flanges and the manifold exteriors. No internal support structures were required due to the aforementioned design changes when built at this orientation on the laser-based PBF. A small sectioned piece of the baseline heat exchanger was also designed with no outer walls so that optical analysis could be performed on the air-side and liquid-side features. Figure 3-7 shows the sectioned part, which was built at the same time as the full-scale HX’s so that it would be representative in terms of material properties, roughness, and build resolution.

Figure 3-6: View of vortex generators in the enhanced second design of the AM oil cooler.

Figure 3-7: Sectioned cube of the interior of the baseline AM HX.
The two AM HX designs were fabricated on an EOS M280 PBF machine at Penn State, using recycled AlSi10Mg aluminum powder [84], which is similar to alloys used in aluminum castings and in PBF AM. The machine was fitted with a flexible recoating blade made of silicone to minimize recoater forces impacting thin build features, and EOS-recommended machine parameter settings were used. Figure 3-8 shows the two full size AM HX in the EOS machine immediately after fabrication. The total build time was 160 hours for this build orientation.

Before detachment from the build plate, the AM HX’s were heat treated to relieve thermal stresses, using a thermal soak at 300°C for 2 hours in an argon environment as recommended by EOS. This also has the benefit of slightly improving the thermal conductivity to 173 W/m-K [84], [85]. The parts were cut from the substrate using a wire EDM (electrical discharge machining) process, and excess support structures were also removed.

Figure 3-8: The two AM HX designs, with external supports attached to the build plate.
The sectioned cube and the full-size HX’s were visually inspected for feature resolution and build issues. After the build was complete, but before heat treatment, some cracks were noted on the air-side fins. Figure 3-9 shows a close-up of fin cracking on the small sectioned cube, which was representative of the full-size HX. The cracking only appears in the inlet or outlet rows of the air-side fins, and it does not propagate into the rest of the core. About 20 percent of the fins on the inlet or outlet face had some cracks, although none were severe enough to completely detach the fin. These issues are likely due to a combination of thin features, lack of constraint toward the edges of the AM HX, and residual stresses induced during the PBF process. Additionally, many air-side fins manufactured with voids through the thickness of the feature which is easily shown when the surface is backlit. Modifications of the machine build parameters could influence and reduce the defects found on the air-side fins [86].

![Figure 3-9: Examples of cracking at the outermost row of air-side fins as well as voids through fin thickness.](image)

Figure 3-10 shows close-up views of the various HX features through a microscope. The major design features are resolved, although they have significant surface roughness as is typical from the PBF process. Table 3-1 presents measurements of fin thickness between the conventional and AM fabricated parts, which indicates good agreement to intended design. Roughness measurements in Table 3-1 were obtained by an optical profilometer at a location on the air-side
fin of the small sectioned cube. As expected, roughness levels are significantly higher for the AM HXs relative to the traditionally manufactured HXs. The as-built surface roughness was not removed so that the performance impact could be assessed in this study.

Figure 3-10: Close-up views of the liquid-side turbulators, air-side fins, and air-side vortex generators in the AM HX's

Table 3-1 also compares the dimensions and mass of the oil coolers, with the AM HX being about 34% heavier than the BTM model on which they were based. The additional mass in the AM oil coolers is most likely due to increased wall thickness in the manifold regions, which was incorporated as a precautionary measure to reduce the chances of pressure-induced leaks. In this initial study, the mass of the additively manufactured oil cooler was not optimized, but it would be important to reduce for aerospace applications.
Table 3-1: Size, weight, and design information for the heat exchangers

<table>
<thead>
<tr>
<th>Oil Cooler</th>
<th>BTM</th>
<th>ETM</th>
<th>BAM</th>
<th>EAM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall size (L x W x H) [cm]</td>
<td>21 x 9.25 x 11.5</td>
<td>20 x 9.25 x 11.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass without fittings [kg]</td>
<td>0.927</td>
<td>0.965</td>
<td>1.241</td>
<td>1.248</td>
</tr>
<tr>
<td>Average air-side fin thickness [mm]</td>
<td>0.30 ± 0.02</td>
<td>0.33 ± 0.04</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fin pitch ($F_p$) [mm]</td>
<td>2.3</td>
<td>&lt;1.7</td>
<td>2.3</td>
<td>2.3</td>
</tr>
<tr>
<td>Louver pitch ($L_p$) [mm]</td>
<td></td>
<td></td>
<td>9.2</td>
<td></td>
</tr>
<tr>
<td>Area-based average surface roughness (Sa) [µm]</td>
<td>0.316†</td>
<td></td>
<td>24.0*</td>
<td></td>
</tr>
<tr>
<td>Area-based RMS surface roughness (Sq) [µm]</td>
<td>0.407†</td>
<td></td>
<td>30.8*</td>
<td></td>
</tr>
<tr>
<td>Area-based peak-valley surface roughness (Sz) [µm]</td>
<td>8.31†</td>
<td></td>
<td>163.2*</td>
<td></td>
</tr>
<tr>
<td>Manufacturing material</td>
<td>Aluminum</td>
<td>AlSi10Mg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Manufacturing method</td>
<td>Stamping, brazing</td>
<td>AM - PBF</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of vortex generators</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>1,155</td>
</tr>
</tbody>
</table>

†Based on sectioned HX; *Based on small test cube

Figure 3-11 shows all four models as well as close-up views of the air-side fins for each of the oil coolers tested in this study. All heat exchangers have the same liquid-side features, and the AM heat exchangers have the same air-side features as the baseline traditionally manufactured geometry (BTM). The enhanced traditionally manufactured (ETM) model has an increased number of air-side fins relative to the other designs which is often the best option for increasing heat transfer for a heat exchanger without large changes in design. This heat exchanger (ETM) was tested to further validate the rig against correlations, but also to compare the AM parts to the more standard approach of heat transfer augmentation using high fin density. The baseline AM heat exchanger (BAM) is intended to replicate the baseline traditionally manufactured geometry, with the aforementioned changes due to limitations in the PBF process. The enhanced additive manufactured heat exchanger (EAM) is the same geometry as the BAM but has small vortex generators added on the air-side, attached to the liquid tube wall. Note that the EAM did not replicate any of the features of the ETM model, but was instead an enhancement of the BAM. The intent was to try to get equivalent performance without the added weight of high fin density.
From the images in Figure 3-11, and confirmed by the measured values in Table 3-1, surface roughness is greatly increased from the traditional to the additive process which is a design consideration. The additive heat exchangers are the same geometry with the exception of the vortex generators, and were fabricated in the same build using common laser settings and uniform powder, so the roughness on all features for both AM HX is assumed to be equal.

![Figure 3-11: Summary and naming convention of all four heat exchangers tested.](image)

**Performance Test Rig**

The test rig shown in Figure 3-12 was designed and constructed to measure heat transfer and pressure drop in the air stream and liquid stream of the heat exchangers. The heat exchangers were placed in an insulated box to minimize heat loss to the surroundings. Although the normal liquid-side working fluid is oil, water was used in this study to avoid complexities of heating and
pumping oil, as well as to minimize fluid property uncertainty. The water is heated up to 70°C using four hot water heating elements that dissipate a total power of up to 7000 watts. The water pump allows for a maximum flow rate of 0.35 kg/s. Since all of the heat transferred into the air is replaced by the liquid-side heaters, the system is able to reach a thermal equilibrium during testing, which enables long run times for accurate performance data collection. The liquid-side inlet and outlet temperatures are measured with calibrated E-type thermocouples (± 0.15°C uncertainty) located in the water flow immediately upstream and downstream of the heat exchanger. The static pressure drop on the water side is recorded using two Dwyer absolute pressure transducers with a range up to 103kPa (15 psi), which are located immediately upstream and downstream of the heat exchanger.

Figure 3-12: Heat exchanger performance rating test rig.

The air-side flow is supplied by a high-volume, low pressure blower driven by a variable frequency drive, that can deliver up to 0.4 kg/s of air at room temperature. A venturi flowmeter is installed in the 20 cm (8") line upstream of the heat exchanger to record the delivered air mass flow rate. Flow conditioning, including straws and screens, is used upstream of the oil cooler in order to
create a more uniform velocity profile entering the heat exchanger. Velocity profiles upstream and downstream of the oil cooler have been recorded by pitot probe traverses and are symmetric, but slightly higher near the centerline of the pipe, shown in Figure 3-13. This is expected because there is no smooth convergence from the pipe into the oil cooler fins.

![Diagram showing pitot probe traverses](image)

**Figure 3-13**: Pitot probe traversed across centerline of heat exchanger in horizontal and vertical directions to understand velocity profiles at heat exchanger inlet.

The air temperature increase across the oil cooler is measured with three thermocouples at the inlet and four at the exit of the flow path. Air-side pressure drop is obtained using two pitot tubes installed in the flow path: one probe was seven centimeters upstream of the HX while the other was two centimeters downstream of the HX. The probes were connected to a differential pressure transducer to measure the static pressure difference between the two probes.

For the pressure drop measurements on the air-side, the total uncertainty at low flow rates (0.13 kg/s) is calculated to be about 3% while at the higher flow rates (0.26 kg/s) the total uncertainty is under 0.7%. The air mass flow rate total uncertainty was about 2% and 0.7% at low and high flow rates, respectively. The uncertainty analysis was performed for one oil cooler using a Kline-McClintock uncertainty analysis [87] and assumed constant for all other tests since all the probes and data recording process are constant between heat exchangers.
The heat transfer both from the water stream and to the air stream was measured by using the mass flow rate, specific heat, and measured temperature change of each corresponding fluid. For low air-side flow rates (0.1 kg/s), the uncertainty in total heat transfer rate was calculated to be 10% and 3% for the water and air, respectively. At higher air mass flow rates (0.25 kg/s) the total heat transfer rate uncertainty was 5% and 2.25% for the water and air, respectively. The gain or loss of energy of each fluid was calculated independently, and the energy balance between the two fluids was within the respective uncertainty for a given air mass flow rate. All heat transfer data shown is taken from the air-side instrumentation due to lower measurement uncertainty.

**Experimental Performance Results**

The oil coolers were tested for (1) overall pressure drop and (2) overall heat transfer in both the air and liquid streams. This section first describes the pressure drop results and comparison to existing literature. The heat transfer results are presented next and compared between the various oil coolers tested.

**Pressure Drop**

Figure 3-14 shows the dimensional air-side pressure drop across the oil cooler for a range of air mass flow rates. The result for the baseline traditionally manufactured (BTM) oil cooler can be directly compared to publicly available data published by the manufacturer [88] for the same model, and shows excellent agreement confirming the accuracy of the performance test rig. For the enhanced traditionally manufactured (ETM) oil cooler tested in this study (7 oil tubes, 8 air-side rows), there is no publicly available data, but there is public data [89] for a larger oil cooler of the same air-side fin geometry, with 13 tubes (14 air-side rows). To enable a comparison, the total mass
flow rate across the larger oil cooler was multiplied by 8/14 (the ratio of air-side rows), and the result is shown in Figure 3-14. This estimate assumes that the mass flow rate of air through each air-side row is equal. As expected, the ETM oil cooler has a larger air-side pressure drop due to the higher density of extended surfaces when compared to the BTM oil cooler. Also, the adjusted pressure drop from the manufacturer agrees well with the measurements.

![Pressure Drop Graph](image)

Figure 3-14: Air-side pressure drop versus air mass flow rate for all oil coolers tested.

Both of the additively manufactured oil coolers (BAM, EAM) had a significantly higher pressure drop than the traditionally manufactured models, on the order of two times larger for a given air mass flow rate. This was not unexpected due to the significant surface roughness that was observed. There was a negligible difference between the BAM and EAM oil coolers, suggesting that the small VG features in the EAM model were not aggressive enough to impact pressure drop. Future work should investigate more impactful features.

The pressure drop data was also converted to friction factor based on the form for corrugated louvered fins described by Chang, et al. [90] and Ryu and Lee [91]:

\[
        f = \frac{A_c}{A_s} \frac{2\Delta p}{\rho u_{in}^2}
\]

(20)
where $A_c$ is the minimum flow area:

$$A_c = (F_p - F_t)F_l$$  \hspace{1cm} (21)

$A_s$ is the surface area (on a fin flow passage basis):

$$A_s = (2 * (F_p - F_t) + 2 * F_l)F_d$$  \hspace{1cm} (22)

and $u_{in}$ is the average velocity at the inlet of the fin row, determined from the overall massflow, overall airflow area into the oil cooler, and incoming air density. Note that the inlet and outlet losses were neglected here since the static pressures were measured close to the oil cooler. Also note that the measured fin thickness was used for the area calculations for the AM oil coolers. In Figure 3-15, the friction factor is plotted versus Reynolds number based on the louver pitch spacing:

$$Re_{lp} = \frac{\rho u_{in}L_p}{\mu}$$  \hspace{1cm} (23)

![Figure 3-15: Friction factor for the louvered fins on the air-side of the oil coolers tested.](image)

The BTM and ETM oil cooler friction factors agree well with the correlations as expected, since the louver surfaces are smooth. Furthermore, the range of $Re_{lp}$ tested indicates transitional behavior, from laminar-like flow at low $Re_{lp}$ to a curve with a more shallow slope at higher $Re_{lp}$ (beyond the range of the correlations). Relative to the BTM, the BAM and EAM models exhibit
high friction factors at low $Re_{Lp}$. This is likely due to the significant surface roughness, and it is a similar trend to that seen by Stimpson, et al. [35] for high roughness in additively manufactured micro-channels. Also, the additively manufactured oil coolers tend to reach an asymptotic friction factor value at a lower value of $Re_{Lp}$, suggesting early onset of turbulent-like behavior.

Figure 3-16 shows the water-side pressure drop for the two baseline designs (BTM, BAM). The other coolers were not tested due to similarity of the liquid-side geometry, which was the same for every oil cooler. Surprisingly, the liquid-side pressure drop of the BAM oil cooler was about half that of the BTM model, despite the aforementioned surface roughness in the additively manufactured HX. The causes of this are not fully understood, but they are likely due to the simplifications in the liquid tube manifold described earlier and less significant impact of surface roughness on the liquid-side. Note that the number and size of turbulator features was kept constant between all oil cooler models.

![Figure 3-16](image_url)

Figure 3-16: Water-side pressure drop versus water mass flow rate for the baseline traditional and additive oil coolers.
Heat Transfer

For each of the four oil coolers, at least three different air mass flow rates were tested, each at three different initial temperature difference in degrees Celsius (ITDc) between the air and water streams. It was found that the heat rejection normalized by the ITDc collapsed to a single value as shown in Figure 3-17 by the BTM data points. This is expected since the flow regime is not changing as a function of ITDc; only the temperature gradients are increased linearly. The different ITDc data points for the other heat exchangers are averaged on Figure 3-17 for easier viewing. It is shown that the ETM has a greater heat rejection than the BTM as expected due to the additional extended fin features. For the AM heat exchangers, the EAM rejected heat slightly more efficiently than the BAM (~4%) for a given air flow rate. Furthermore, both AM heat Figure 3-14 exchangers outperformed the BTM heat rejection by over 10% at the highest flow rates. However, the air-side pressure drop for the AM heat exchangers was roughly double that of the traditionally manufactured heat exchangers as shown in . Future work will investigate the level of surface roughness and the potential to optimize it for this application.

Finally, Figure 3-17 indicates that both of the additively manufactured heat exchangers provide as much heat rejection as the enhanced traditional model (ETM) despite not having the aggressive enhancement features of the ETM. Thus, further heat transfer performance gain may be achievable by designing aggressive heat transfer surfaces. An advantage of the AM technique is that these enhancements may be able to be added at minimal additional fabrication cost. Alternatively, it may be possible to design a heat exchanger that performs as well as the ETM but is lighter than the BTM, by removing unnecessary material from the baseline design, especially in the manifold.
Conclusions

This study has investigated the potential for additive manufacturing to generate functional heat exchangers and the implications of the AM process on HX design and performance. A traditionally-manufactured aircraft oil cooler was replicated using a laser-based power bed fusion additive manufacturing process. To demonstrate the capability of novel designs enabled via AM, an enhanced heat exchanger was also designed and fabricated with air-side vortex generators. The traditional and AM HX’s were tested in a newly developed rig which was validated by comparing the performance of the traditionally manufactured baseline to public data.

Both of the AM heat exchangers increased the heat transfer by about 10%, relative to the traditionally manufactured heat exchanger on which the geometry was based; however, the air-side pressure drop roughly doubled. Vortex generators added to one of the AM heat exchangers had little effect on air-side pressure drop but did enhance heat transfer slightly. Additionally, the vortex generators added very little additional mass to the oil cooler.

The AM process did produce a HX with noticeable build defects on the air-side fins (i.e., cracked fins and voids). The liquid-side features may also exhibit build defects (less likely because
the features are more stout), but these cannot be seen until a CT scan of the heat exchanger is performed or the heat exchanger is sectioned, which is planned for future analysis. From the completion of this initial study, it is clear that modifying the original heat exchanger design for production via PBF AM is necessary but could likely yield additional performance benefits.

Further work should be done to investigate how the natural surface roughness of the PBF process affects heat transfer and pressure drop in compact heat exchangers. It is believed that the consequential surface roughness is the main reason for increased heat transfer and pressure drop rather than the geometry changes. Using the louvered fin correlations by Chang et al. [90] and Ryu et al. [91], for a 25% increase in fin thickness at a louver pitch Reynolds number of 1000, the f-factor would change by only 6% while j factor would change by even less. These are much smaller changes than observed experimentally, which suggest that a large part of the difference observed is due to roughness. There may be post-processing techniques to remove the roughness [92], but it is unclear how to avoid polishing away fine features such as fins or vortex generators. Finally, understanding the durability of AM heat exchangers is important when reducing the wall thicknesses to reduce the overall part weight.

Acknowledgments

The authors would like to acknowledge William Genevro and Airflow Systems, Inc. for providing the oil coolers and advice about the subject, and Northrop Grumman Electronic Systems for donating the metal powder. The authors would also like to thank Griffin Jones and Alex Dunbar in Penn State’s Center for Innovative Material Processing through Direct Digital Deposition (CIMP-3D) for their help with software support. Material and fabrication costs for the AM heat exchangers was provided by CIMP-3D.
Chapter 4
Additively Manufactured Overall Pressure Loss and Heat Transfer Performance of Offset Strip Fins used in Compact Heat Exchangers.

Abstract

Heat exchangers for a wide range of applications have been studied in detail, and many resources are available to predict overall heat transfer and pressure drop performance. However, with the relatively new technology of metal additive manufacturing and the resultant surface roughness, the traditional correlations and design considerations might need to be adjusted. As a result, two metal additively manufactured offset-strip fin heat exchanger geometries with different fin spacing are explored for heat transfer and pressure drop performance compared with traditional correlations. Deviation between the additively manufactured geometries and previous correlations for smooth fins are found, and are further amplified as the surface roughness to hydraulic diameter ratio is increased. Furthermore, the surface roughness from the additive process results in a constant friction factor behavior at high Reynolds numbers. Considerations for both the laminar and turbulent flow regimes are needed for correct performance prediction. A final offset strip fin geometry with a change in the fin spacing every third of the way through the flow path was tested, which is difficult to design and fabricate traditionally but simple to make using additive manufacturing. The orientation of fin spacing, wider spaced to tightly spaced or tightly spaced to wider spaced, did not have a significant effect on pressure drop or heat transfer. However, a method for predicting the performance of a spatially varying geometry is found, which will become more important as additive manufacturing increases possible complexity of heat exchanger designs.
Introduction

Thermal management design is becoming more important as power consuming devices are becoming smaller and more compact [93], resulting in new challenges. To ensure correct operation temperatures, heat exchangers are used to reject excess heat by using one or more fluids. For aerospace applications, thermal management is becoming a main challenge as the reduction on fuel and emissions is forcing all aerospace components to be reduced in size and weight but still operate efficiently. In particular, electric aircraft having multiple motors, high power consuming electronics, and potential for battery overheating are just a few of the many thermal challenges. The use of metal additive manufacturing to create custom and high-performance heat exchangers for a wide range of thermal management applications could be valuable to increase system efficiencies and a necessary step for electric powered aviation.

A common geometry used in compact heat exchangers known as the offset strip fin (OSF) has been well documented and studied for performance. These heat exchanger surfaces are manufacturing by folding or stamping sheet metal to form the fins, then brazed onto the plate wall that carries the other heat transfer fluid. Consequently, the stamping process creates a smooth fin surface with the only roughness coming from burred fin edges from the mold form having wear or surface fouling during operation [3], [4]. However, when this same geometry is formed using metal additive manufacturing (AM), the resultant surface roughness inherent from AM can be ten times higher than found with traditionally manufactured stamped fin surfaces [22]. The roughness on metal AM surfaces does impact the heat transfer and pressure drop performance on geometries that are relatively simple [35], [36] and quite complex [21], [94], which deviates the pressure drop and heat transfer performance predictions. However, the impact of surface roughness on AM OSF geometries is not well understood.
Metal AM OSF geometries tested in this study were produced using a laser-based powder bed fusion (PBF) process sometimes referred to as selective laser melting (SLM). This process uses a high-powered laser that melts a thin layer of the powdered metal, \( \sim 30 \) µm, that forms the solid sections of the part. After a layer is complete, powdered metal is recoated over the previous layer and then process starts again, resulting in a layer-by-layer additive process. Once the part is fully built, all remaining powdered metal must be removed from the part before the heat treatment is performed to avoid sintering of any remaining metallic powder that could cause a flow blockage. The AM OSF was fabricated on an EOS M280 PBF machine at The Penn State University using stainless steel 316L metallic powder [95].

During the metal AM build, support material and the part are built as homogenous feature and the support material is removed afterwards through machining or, more typically, wire electric discharge machining (EDM). This additional material used to support the part during the build is required because any previous layer of pure powdered metal cannot support solid material above it. Furthermore, support material prevents the current layer being built from distorting caused by thermal stresses during the PBF process. However, if the build angle of the part is large enough relative to the build direction, the solid material can be self-supporting with negligible distortion and does not require support material.

The PBF AM process is able to create unique and optimized shapes but the major advantage of using offset strip fins is their robust yet simple design, which is a reason they are so commonly found in compact heat exchange applications. Furthermore, the offset strip fins themselves do not required support material during the PBF process, which is important for internal locations where access to remove the support material is blocked. Additionally, since the fins are built at an angle and all features are perpendicular to each other, the roughness on all features can be predicted somewhat well since all surfaces are either plus 45° or minus 45° from the build direction. The metal AM roughness on circular and other more complex features is a bit harder to quantify because
the angle continuously changes relative to the build direction [38], [43]. Lastly, even with the simple design of the OSF geometry, optimization of the fin thickness, height, spacing, and length, can be varied to produce the highest performing geometry combination [96], [97] for a given operating condition.

**Previous Studies**

Offset strip fins have been studied extensively by a number of different researchers with Kays and London [47], Manglik and Bergles [9], and Webb et al. [6] being some of the most notable in overall performance correlation. In most cases, having a fin that is very thin relative to the fin length and fin spacing gives best performance [54] which can be a challenge for metal AM because features around 0.25 mm and smaller do not often manufacture well [98]. Many fundamental and numerical studies have been performed on a slightly modified offset strip fin called staggered parallel plates [99]. The staggered plate geometry matches the offset strip fin geometry ignoring the extra material on the channel walls from the fin stamping process. The staggered plate studies are usually scaled up to gain more measurement resolution and, in some cases, use optical and flow visualization techniques. For example, Mochizuki et al. [53], [55] scaled up the staggered plates to a thickness of 1.6 mm for dye-injected flow visualization in a water tunnel. Mochizuki found that as the Reynolds number was increased, wake shedding behind the plates started to occur further and further upstream. A similar set up was used by Dejong et al. [56] to capture a feathery-like wake which were reported to be small-scale transverse vortices. Dejong also captured local mass transfer on the fin surfaces by using the naphthalene sublimation technique which is has measured values analogous to local heat transfer values. At Reynolds numbers above about 750, the highest local heat transfer on fins can be at 20% down the length of the fin due to boundary layer separation and recirculation near the leading edge. A nearly identical result was found from a numerical
analysis by Toubiana et al. [65] which captured the vortex roll-up at the leading edge of the fins. Lee et al. [57] used a LDV technique for velocity measurements in offset strip fin cores as well as studied the flow field impact of various plate arrangements. The modified designs to augment heat transfer that Lee et al. [57] and Peng et al. [64] showed are a great example of an easily additively manufactured geometry when compared to traditional manufacturing methods.

Numerical studies of the OSF geometry or staggered parallel plates has shed insight into the flow physics and heat transfer of this type of geometry. A good example from Patankar et al. [3] shows how considering a fin with finite thickness can have an impact on the recirculation flow structures behind the fins. Other numerical studies in both two dimensional, three dimensional [59]–[62], and unsteady flow which, are able to capture some of the major flow structures around the fin. Pham et al. [66] was one of the first to use large-eddy simulation to capture the evolution of wake shedding in the streamwise direction of offset strip fins. Another example of this was shown by Saidi et al. [67] that found the wavy flow oscillation in their offset strip fin geometry occurred at a frequency of ~68 Hz for a Reynolds number of 1124. From a 2-D unsteady simulation, Zhang et al. [100] found oscillatory flow motion at a Reynolds number of 720 and a slightly increasing Strouhal number as Reynolds number was increased. Toubiana et al. [65] compared different turbulent flow solvers and goes into detail about the advantages and disadvantages of each. Some numerical solvers can calculate up to 44% different in \( f \) and 27% different in \( f \) prediction when compared to the most complex flow solver, LES. The numerical studies also allow for the relative contribution of pressure loss from shear and pressure drag to be determined [9]. At low Reynolds numbers, the skin friction can be ~40% of the total loss contribution while at high Reynolds numbers, the form drag contributes to over 80%.

The flow field of an OSF geometry with the random-like roughness created from the AM process has not been explored because traditional manufacturing processes do not produce this result. However, structured roughness has been added to surfaces to enhance heat transfer
performance for heat exchange applications which is well documented by Webb et al. [6]. Likewise, researchers such as Mahmood et al. [101] scaled up a dimpled surface, a common structured roughness, and performed flow visualization, and spatially resolved heat transfer. The high resolution from the scaled-up geometry show vortex pairs shedding from the dimples and boundary layer disruption augmenting heat transfer; a similar result to a numerical simulation by Lu et al. [102]. However, the roughness created from the metal AM process is more random-like which is not analogous to the structured roughness that has been previously studied.

**Fluid and Heat Transfer Parameter Definitions**

The offset strip fin is relatively simple compared with other heat exchanger fins but there is still some disagreement in the literature on how to define hydraulic diameter which is used in Reynolds number and correlation calculations [103]. The most common definition for hydraulic diameter \( D_h \) is outlined by Manglik and Bergles [9] but the current study proposes to slightly alter the definition based on the nature of additively manufactured fins, as shown in Figure 4-1. The differences in \( D_h \) between the OSF and the geometry in this study are from the extra material that exists due to the folding process in traditional offset strip fin manufacturing, which reduces the flow area by the thickness of the fin material which is folded flat to braze to the roof and floor of the flow channel. In an additively manufactured part, this extra material is not necessary—the fins are directly attached to the floor and roof as the part is built. This current arrangement is similar to previous studied geometries called staggered plates that are a simplification of the offset strip fin, as described by a few previous researchers such as Manson [99] and Mochizuki et al. [55]. The staggered plate geometry is typically scaled up for high resolution measurements to understand the fundamental physics in the OSF geometry [104] with the \( D_h \) being defined as four times the cross-sectional area divided by the wetted perimeter. However, even for the simplified staggered plate
geometry, Reynolds number has been defined using the thickness of the fin rather than the hydraulic diameter [104].

For the additive offset strip fins tested, the commonly used performance parameters of friction factor, $f$, and nondimensionalized heat transfer, $j$, are defined in Equations 24 and 25.

$$ f = \frac{\Delta P D_h}{2 \rho u_{avg}^2 L} \quad (24) $$

$$ j = StPr^{\frac{2}{3}} \quad (25) $$

The Stanton number used in $j$ correlation is defined as follows:

$$ St = \frac{U_{avg}}{\rho u_{avg} C_p} \quad (26) $$

with the convective heat transfer coefficient, $U$, is defined in Equation 27

$$ U_{avg} = \frac{m C_p (T_{out} - T_{in})}{\eta R A_{surface} \Delta T_{lm}} \quad (27) $$

Figure 4-1: Simplified offset strip fin geometry for additive manufacturing with geometric definitions and schematic based off Manglik and Bergles [9].
where $A_{surface}$ is the total wetted surface area which includes all four surfaces of the fins as well as the lower and upper walls of the channels and $\eta_T$ is the total surface efficiency.

The log-mean temperature difference $\Delta T_{lm}$ is found using the inlet and exit air temperatures as well as the calculated wall temperature on the inside of the coupon. The wall temperature is found from a one-dimensional conduction analysis knowing the temperature of the heater block, the thermal conductivity of the sample [95], and length of each section of material between the heater block and the coupon inner wall. The temperature difference between the heater block and inner wall of the coupon was a maximum of 4°C at the highest heat flux condition. Despite the thermal contact surfaces being sanded smooth, thermal paste [105] was added between the heated blocks and the coupon outer wall to fill any small voids. The volume of thermal paste was recorded and was assumed to be spread evenly across the sample resulting in a thickness of 15µm which was accounted for in the 1D conduction analysis.

The total surface efficiency, $\eta_T$, and fin efficiency, $\eta_f$, are accounted for since the fin material is stainless steel which has a thermal conductivity of about $14 \frac{W}{m-K}$ [95]. The total surface efficiency and fin efficiency are shown in Equation 28 and Equation 29.

$$\eta_T = (1) \frac{A_{wall}}{A_{surface}} + (\eta_f) \frac{A_{fin}}{A_{surface}} \tag{28}$$

with a wall efficiency of 1 and a fin efficiency defined as:

$$\eta_f = \tanh \left( \frac{m(h+t)}{2} \right) \tag{29}$$

where

$$m = \left( \frac{2U_{avg}}{k_f t} \right)^{\frac{1}{2}} \tag{30}$$
For these calculations, \( U_{avg} \) is the average convective heat transfer coefficient on the fin surface. The process of determining the convective heat transfer coefficient is implicit with \( U_{avg} \) being calculated from Equation 27 with \( \eta_T \) equal to one. An iterative process is performed until \( U_{avg} \) is within 1% for both Equation 27 and Equation 29.

**OFS Geometries Studied**

A total of 5 offset strip fin geometries were tested: three metal and two plastic. The three metal additively manufactured offset strip fin geometries shown in Figure 4-2, with different fin spacings, were experimentally tested for heat transfer and pressure loss performance. The geometries were manufactured additively from stainless steel powder with a thermal conductivity of 14 W/m-K [95]. Another two plastic offset strip fins with the same geometry as the tightly and wider spaced metal fins were printed using a resin-based printer, which results in smooth fin surfaces to isolate the effect of roughness. A Photocentric Liquid Crystal Pro resin-based printer with a 137 \( \mu \)m lateral resolution and 100 \( \mu \)m vertical layer height was used to manufacture the plastic OSF. Daylight Pro Hard resin [106] was selected for its strong mechanical properties and post-cured with high powered curing lights. The plastic parts were only used for pressure loss analysis because of the low temperature tolerance and poor thermal conductivity of the material. Both the metal and plastic OSF geometries are required to be built up at a 45° angle from the build direction in order to avoid support material being used inside the fins. The minimum self-supporting angle for different materials and additive processes does vary, but 45° is usually sufficient.
The variable spaced (VS) fin has a step change in fin pitch every third of the way through the flow path, shown in Figure 4-3. The geometry includes the tightly and wider spaced fin spacing as well as a fin spacing that is even larger. The use of a geometry like this could be used to obtain a more even temperature distribution if an applied heat flux is not uniform or in boiling applications to avoid high temperatures and possible burnout [107] by providing additional heat transfer surfaces. However, a method to predict the heat transfer and pressure loss performance of a geometry like this is not well defined. To get detailed pressure loss measurements, pressure taps are installed in the first and last row of each of the three discrete fin spacings. Air flow can go in either direction into the coupon for pressure loss and heat transfer performance measurements, by switching its orientation in the test rig.
The experimental rig, shown in Figure 4-4, was created to test additively manufactured offset strip fin arrays that are 126x63x5 mm. The fin arrays have 36, 18, or 9 rows of fins across the width of the coupon and 24 rows of fins along the length of the coupon. Many rows of fins allow the flow to be periodic and unaffected by the side walls of the coupon. The rig performance measurements were validated with a machined coupon with smooth rectangular channels and an aspect ratio of 0.5 height-to-width, as seen in Figure 4-5. This geometry was chosen because of the prevalence of existing correlations of Nusselt number and friction factors in current literature. Additionally, the solid sections of the channel are kept thick to ensure a uniform wall temperature on each channel side to replicate the correlation’s isothermal wall boundary condition. To verify, a fin efficiency analysis [108] shows the lowest efficiency being over 99% at the highest air flow rate tested, implying a very uniform temperature in the solid. The experimental data is compared with
existing correlations, shown in Figure 4-6 and Figure 4-7, which agrees well, indicating that the test rig and its instrumentation were accurate over the entire range of interest.

Figure 4-4: Sectioned view of experimental rig used to test additively manufactured samples for heat transfer and pressure loss performance.

Figure 4-5: Geometry used to baseline the rig and compare to existing correlations, picture taken before seam was sealed shut.
For heat transfer tests, a total of 42 type-E thermocouples are used with three at the inlet of the coupon and six at the exit to measure incoming and outgoing flow temperature, respectively. Other thermocouples are used to monitor coupon wall temperatures and to perform a 1D conduction heat loss analysis from the insulation. From an energy balance analysis, the sum of thermal energy

Figure 4-6: Baseline geometry experimentally tested and compared with friction factor, $f$, correlations, [108]–[110], to verify correct rig set-up.

Figure 4-7: Baseline geometry experimentally tested and compared with Nusselt number correlations, [108], [110], [111], to verify correct rig set-up.
imparted to the air through the coupon walls and the conduction heat loss through the insulation was never more than 10 percent different than the electrical power supplied to the heaters. Any imbalance larger than 10 percent was not included in the data, which always occurred at low flow rates. High flow rates had the least imbalance of around 5 percent.

During pressure loss measurements, the top heater block and insulation was removed to gain better access to the coupon and allow for pressure taps to be installed in the coupon upper wall. All static pressure tap locations are in the first and last row of fins except for the overall pressure drop of the variable spaced (VS) fin geometry which are taken just before and after the inlet and exit of the coupon, respectively.

**Surface Characterization of Fins**

Metal and plastic fins from a smaller sample piece created during the same builds as the test coupons were used to analyze the surface roughness using a Nexview 3D Optical Profilometer. The downward and upward facing surfaces of the metal fins were scanned and the images stitched to cover a large area of the fin at high resolution shown in Figure 4-8. The scan has a lateral resolution of 0.803 µm and a total scan width and length of 1.477 mm.
The plastic OSF were also characterized with the same equipment and each roughness value is shown in Table 4-1. It was found that the plastic fins had a similar roughness on the upward and downward facing surfaces of the fins and a very low roughness value when compared to the metal parts. The roughness of the plastic fins is about twice that of traditionally manufactured folded aluminum fins used in commercial heat exchangers [112] but still an order of magnitude less rough than the metal additive manufactured fins.

Table 4-1: Measured surface roughness of additively manufactured fins

<table>
<thead>
<tr>
<th>Fin Material</th>
<th>Roughness property</th>
<th>Upward facing [µm]</th>
<th>Downward facing [µm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>AM stainless steel</td>
<td>$S_a$</td>
<td>15.5</td>
<td>26.4</td>
</tr>
<tr>
<td></td>
<td>$S_q$</td>
<td>21.7</td>
<td>33.0</td>
</tr>
<tr>
<td>AM plastic</td>
<td>$S_a$</td>
<td>2.4</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$S_q$</td>
<td>3.1</td>
<td></td>
</tr>
</tbody>
</table>

Figure 4-8: Profilometry measurements show different roughness types on upward and downward facing surfaces on the metal offset strip fins
The surface of a few metal AM offset strip fins were also viewed with a scanning electron microscope (SEM) at two magnifications to optically view the downward and upward sides of the fins. Figure 4-9 shows how the upward facing surfaces of the print appear smoother with fewer ball-like structures of partially melted powdered metal when compared with the downward facing surfaces. This could result in different boundary layer growth on each side of the fin surface.

![SEM images](image.png)

Figure 4-9: SEM images taken in the center of the downward and upward facing metal AM OSF fin surfaces at two different magnifications.

**Results**

The heat transfer and pressure drop results are presented by first exploring the two OSF geometries that have constant fin spacing throughout the coupon. The experimental data is compared with previous correlations of the same geometry. Next, the heat transfer and pressure
drop of the OSF geometry with variable spaced (VS) fins are examined to understand the local and overall OSF performance. Last, all three metal AM OSF geometries are compared for overall performance by comparing the ratio of heat transfer to pressure drop which is commonly performed for heat exchanger fins.

**Constant Spaced OSF**

The friction factor, $f$, data from the plastic (smooth) and AM (rough) parts are compared with OSF correlations from Mochizuki et al. [104], in Figure 4-10 and Figure 4-11, with a regression analysis discussed later in this section. As mentioned previously, the static pressure at the inlet and exit of the geometry are taken in the first and last rows of fins, respectively. The static pressure drop and the geometry of the fins are used to calculate $f$ as a function of Reynolds number.

![Figure 4-10: Friction factor of plastic (smooth) additively manufactured offset strip fins.](image)
The friction factor, $f$, for the smooth OSF agrees reasonably well with the Mochizuki et al. correlations, especially after a Reynolds number of around 100. For most practical applications, the use of OSF geometries at Reynolds number below 100 is not typically performed or studied; so, it is unclear why this increase in $f$ occurs. However, Peng et al. [113] experimentally and numerically found a similar increase in $f$ at very low Reynolds number using an OSF geometry that is referred to as serrated fins in their study.

The metal AM OSF geometries have $f$ values that are larger than the smooth fin for all Reynolds number. At the highest Reynolds numbers $f$ values for the metal AM OSF geometries becomes nearly constant. The tightly spaced fins ($s = 1.40$ mm) have the highest $f$ at all Reynolds number and seem to transition from laminar to turbulent flow at a lower Reynolds number. Furthermore, $f$ for both AM OSF is higher even at low Reynolds, which has also been measured in rectangular metal AM channels [36]. Other researchers exploring large ratios of roughness-to-diameter [114], [115] propose that the increased $f$ at low Re numbers could be caused from: (1) the rough elements reducing the open flow area and making the traditional calculation of hydraulic

Figure 4-11: Friction factor of metal additively manufactured (rough) offset strip fins with power law regression.
diameter incorrect or (2) the rough elements causing larger surface shear stresses. The decrease in overall flow area due to surface roughness is not the case in this study since the fin thickness of the tested geometry are measured from one side of the fin to the other which makes all roughness features inclusive. Additionally, the fin thickness of the metal AM OSF would need to be increased by 75% to match with the smooth fin correlations, indicating that the roughness is not significantly reducing the open flow area. The increased $f$ is caused by a combination of added surface area from the roughness and by each fin developing a new boundary layer on the leading edge causing locally high surface shear stress. The most tightly spaced OSF ($s = 1.40$ mm) has a total of 864 fins which cause constant disruption and re-building of the hydraulic boundary layer similar to a study by Gloss et al. [116] which measured a doubling of $f$ in the deep laminar regime with a 1000 roughness elements that are one-tenth the channel height.

An interesting trend in $f$ noticed in a few locations, but most pronounced at a Reynolds number of 1200 for the wider spacing OSF ($s = 3.15$ mm), is a small increase followed by a decrease in $f$ as Re is increased. To ensure that this was not caused by instrumentation and was flow based, additional data points were taken at much smaller flow rate increments, see Figure 4-12, with data taken by both increasing and decreasing Reynolds number to rule out any hysteresis effects.

The unique $f$ behavior for these Reynolds numbers is attributed to changes in flow separation and reattachment behavior as stated by Michna et al. [48], who measured a similar trend at high Reynolds numbers. The Reynolds numbers in this region do appear to be around the expected flow transition from laminar to turbulent regimes. There are other studies that indicate similar small transitional behaviors in $f$ [63], [117] with this occurrence most prevalent when the fins are relatively short in length but widely spaced which allows wake shedding and oscillatory flow motion to occur. When the fin thickness is large relative to the fin spacing, a center core flow develops and is bounded by the fins [3], reducing oscillatory motion and wake shedding.
As mentioned previously, only the metal AM geometries could be tested for heat transfer performance. The non-dimensional heat transfer, \( j \), for the metal OSF geometries, incorporating fin efficiencies into the data reduction, is shown in Figure 4-13. The AM fins with the wider spacing (s = 3.15 mm) align with the traditional smooth fin correlation for laminar flow but do not match the predicted increase in heat transfer for higher Reynolds numbers. The tightly spaced (s = 1.40 mm) fins have a high \( j \) compared with the smooth fin correlation in the laminar flow region but follow the correlation at higher Reynolds numbers. It was not possible to test a wide range of Reynolds numbers due to experimental limitations; so, a power law regression is fit to the measured data and extrapolated to higher and lower Reynolds number values which seems to parallel the correlations. The main reason for a smaller range for the heat transfer test was due to the limitations of the coupon heating system used in the experimental set up. At the lowest Reynolds numbers, the exiting air temperature was very near the coupon wall temperature creating large measurement uncertainties that are excluded from the plots. The upper limit on the external temperature was restricted to a maximum of \( \sim 220 ^\circ C \).

Figure 4-12: Additional data points taken around Reynolds number of 1200.
To develop correlations for the metal AM OSF behavior, a power law regression is used for $f$ at high and low Reynolds numbers and then linked using an asymptotic correlation method. The simpler $j$ power law regression and the more involved $f$ correlation are shown in Equations 31, 32, 33, and 34.

$$j_{s=3.15} = 0.494 Re^{-0.525}$$  \hspace{1cm} (31)  

$$f_{s=1.40} = 1.164 Re^{-0.597}$$ \hspace{1cm} (32)  

$$f_{s=3.15} = [(14.963 Re^{-0.813})^{2.86} + (0.05 Re^{-0.003})^{2.86}]^{0.35}$$ \hspace{1cm} (33)  

$$f_{s=1.40} = [(15.223 Re^{-0.767})^{10} + (0.0969 Re^{-0.005})^{10}]^{0.10}$$ \hspace{1cm} (34)  

The $f$ regression follows the form $f = (f_{laminar}^n + f_{turbulent}^n)^{\frac{1}{n}}$, which allows the laminar $f$ to be prominent for low Reynolds number and the turbulent $f$ to be prominent for high Reynolds numbers [118], [6]. The exponent, $n$, determines the weighting at high and low Reynolds numbers. In other words, a very high $n$ value would immediately change the overall regression as if the two functions intersected each other with no interpolation. Consequently, the tightly spaced OSF ($s = 1.40$ mm) has a quick transition from laminar to turbulent flow; so, a high $n$ value best represents the data.
Likewise, a lower $n$ value is used for the wider spaced OSF ($s = 3.5$ mm) since the transition from laminar to turbulent flow is more gradual. It is notable that if the laminar and turbulent regressions are similar to each other, then a lower $n$ value can be used since less interpolation is needed. The exponent associated with the Reynolds number is the slope of the regression when plotted on logarithmic scale. For turbulent flow, the exponent on the Reynolds number for the $f$ is nearly zero since the $f$ is almost independent of Reynolds number in Equations 33, and 34.

The advantage of having a regression is the ability to compare the $j/f$ performance ratio, known as the goodness factor [119], for a wide range of Reynolds numbers and not just the values experimentally tested. Figure 4-14 shows the goodness factor using the $j$ regression with the measured $f$ data points since there are not necessarily $j$ results at the same Reynolds number as the $f$ results. For the fin geometry with the wider spaced fins ($s = 3.15$ mm), the maximum ratio of heat transfer to pressure drop occurs at a Reynolds number around 1000 while the tightly spaced fins ($s = 1.40$ mm) has a global maximum for performance around a Reynolds number of around 600. The correlations proposed by Mochizuki et al. [104] have an equal goodness factor for each OSF geometry in the laminar regime due to the identical exponents on Reynolds number in their correlation. In the laminar regime, their correlation predicts that the tightly spaced fins ($s = 1.40$ mm) should have a slightly higher goodness ratio than the wider spaced fins ($s = 3.15$), which was also true for the AM fins. Similarly, in the turbulent regime, the correlation developed here accurately predicts that the goodness ratio will be higher for the wider spaced fins than for the tightly spaced fins. Overall, the AM fins have lower goodness factor ratios for all Reynolds numbers due to the increased $f$ caused by the rough surfaces.
As mentioned previously, the variable spaced (VS) fin geometry could be a beneficial design when a heat exchanger application has a non-uniform heat flux and wall temperatures need to be kept constant. The overall friction factor, $f$, for the variable spaced (VS) fins was found by measuring the static pressure drop upstream and downstream of the entire coupon. The static pressure drop across each separate fin spacing was also measured with the pressure tap layout shown in Figure 4-3. Additionally, pressure drop measurements are taken with the coupon orientation having the most tightly spaced fins at the inlet, as well as, the reverse orientation with the tightly spaced fins at the exit. The geometry of the VS fins changes through the coupon so calculating an average $f$ is difficult because the hydraulic diameter and average velocity in the fin array changes. One option is to use the global average fin spacing to obtain geometric and fluid parameters which results in $f$ values shown in Figure 4-15. By this metric, $f$ for the VS geometry
exceeds that of the tighter spaced fins for low Reynolds numbers and matches $f$ at high Reynolds number. The resulting globally averaged VS geometry would have an equivalent fin spacing of 3.73 mm. Clearly, $f$ of the VS OSF with a globally averaged geometry well exceeds the pressure drop of an OSF of the same geometry.

To understand why $f$ for the VS geometry is so large, the distribution of pressure drop attributable to each subsection in the VS array is shown directly in Figure 4-16 as a function of mass flow rate. Although the tightly spaced segment ($s = 1.40$ mm) contributes the most to the total array pressure loss, its contribution tends to decrease with increasing Re while the moderately spaced and widely spaced section picks up a higher percentage of pressure loss. The artificially high $f$ from averaging the entire geometry into one fin spacing is caused by not accounting for the different contributions of pressure drop for each fin spacing.

Figure 4-15: Overall friction factor performance for the variable spaced fins (VS) with the results being nearly identical for both flow directions.
When the pressure drop for each subsection of the VS geometry is converted into a friction factor $f$ from its respective geometry, the resulting $f$ values are more in line with the constant spaced fins as previously tested. Figure 4-17 and Figure 4-18 show $f$ values for each subsection of the VS geometry with flow going in either direction through the coupon. At low Reynolds numbers the fin array orientation has little impact on $f$ for each section of fins, but there are clearly different behaviors at the highest Reynolds numbers, with the constant friction factor, $f$, increasing with decreasing fin spacing. When the tightly spaced fins are at the inlet of the coupon, as in Figure 4-17, $f$ for the next two sections of fins are higher than the reverse flow orientation Figure 4-18. By having the tightly spaced fins at the inlet, the downstream fin rows experience higher levels of turbulence which cause flow instabilities and ultimately result in wake shedding which might be less present in the reverse flow orientation. Previous research has shown that at higher Reynolds numbers the main contributor of pressure losses in OSF arrays are due to form drag caused by wake shedding behind the fins [66].

Figure 4-16: The contribution of total pressure drop for each section of the VS fins.
The most tightly spaced fins \((s = 1.40 \text{ mm})\) in the VS array do not reach a constant \(f\) that is as high as \(f\) for the 24 rows of constant spaced fins of same geometry, shown previously in Figure 4-17.

Figure 4-17: Calculated \(f\) for each subsection of the VS fin geometry with flow going from tight to wide spaced fins.

Figure 4-18: Calculated \(f\) for each subsection of the VS fin geometry with flow going from wide to tight spaced fins.

The most tightly spaced fins \((s = 1.40 \text{ mm})\) in the VS array do not reach a constant \(f\) that is as high as \(f\) for the 24 rows of constant spaced fins of same geometry, shown previously in Figure
4-11. Having only 8 rows of fins in the VS geometry results in a lower \( f \) due to the flow behavior dependence on the number of rows. Initial boundary layer development in the first row of fins causes the largest \( f \) due to the flat velocity profiles at the inlet. In the subsequent rows, the flow through the array begins to establish but does not separate from the fins, and \( f \) is reduced. Once instabilities and flow separation on the fin surfaces being in the array, \( f \) rises. Pham et al. [66] calculates that at a Reynolds number of 3000, \( f \) for row 5 can be 50% less than in row 15.

From the pressure drop measurements for each subsection of the VS fins, it is confirmed that using the global average fin spacing to obtain geometric and fluid parameters produces an artificially large \( f \) as shown previously in Figure 4-15. To calculate a more representative \( f \) for the VS geometry, the method needs to account for the increased \( f \) and pressure drop associated with the most tightly spaced fins. This is accomplished by calculating the Reynolds number and \( f \) in each section of fins for a given mass flow rate and then averaging the result. This accounts for the tightly spaced fins having a lower Reynolds number and higher \( f \) than the widely spaced at the same mass flow rate due to difference in hydraulic diameter and average velocity. It is important to use the same mass flow rate in the calculation and not the same Reynolds number because only the mass flow rate is held constant throughout the fin array. This method produces a more reasonable \( f \) when compared with \( f \) for each respective fin section in the VS geometry, see Figure 4-19, as well as when compared with the constant spaced fins, shown in Figure 4-20.
Figure 4-21 shows that $j$ for the VS fins is nearly identical and independent of which direction the flow travels through the coupon; this is the same result as for $f$. Since the temperature after each section of fins is not measured, calculating the heat transfer contribution for each fin
section cannot be performed. Therefore, the globally average fin spacing to obtain geometric and fluid parameters results in an artificially high $j$, as seen in Figure 4-21. Although it cannot be confirmed without temperature measurements, the larger than expected $j$ is most likely due to the tightly spaced fins having higher heat transfer than is represented in the geometry averaging for the VS fins.

Figure 4-22 shows the goodness ratio, $j/f$, for the VS geometry using globally averaged results. The VS goodness factor is reasonably in line with the other AM OSF geometries tested even though both $f$ and $j$ values were over predicted. Consequently, by using the goodness factor ratio, the method of averaging the geometry is not significant and results in an accurate overall performance prediction. The tightly spaced OSF have the highest levels of goodness ratio in the laminar regime but gets significantly reduced in the turbulent regime from high pressure losses. The widely spaced ($s = 3.15$ mm) and VS geometry seem to benefit from flow mixing at higher Reynolds numbers without a significant increase in pressure loss. The VS OSF geometry has the highest overall performance at Reynolds numbers above 1000 which is a result of the widest spaced fins in the VS array having very low pressure drop with some heat transfer. This is similar to how plain fins that have the highest goodness factors compared to any other fin type due to the extremely low pressure drop performance and a maximum performance around a Reynolds number of ~5000. However, the plain fin would result in a large, heavy heat exchanger to reject the required heat load, which is not ideal for compact heat exchanger applications.
Figure 4-21: Overall $j$ performance for the variable spaced fins (VS) with the results being nearly identical for both flow directions.

Figure 4-22: Goodness factor for all three metal AM coupons
Conclusions

A test rig was designed and baselined to experimentally test the performance of AM heat exchanger surfaces called OSF. The metal OSF were fabricated from stainless steel via the laser powder bed fusion process and compared with smooth fin counterparts and traditional correlations. The fin surfaces were characterized through surface scans and imaged with a scanning electron microscope to view the resultant surface roughness from the AM process which was an order of magnitude larger than the roughness found on traditional heat exchanger surfaces. The resultant surface roughness from the metal AM process caused the friction factor, \( f \), to be larger when compared with a similar smooth fin geometry in both the laminar and turbulent flow regions. It was previously thought that surface roughness does not impact \( f \) for low Reynolds numbers, but the constant reforming of the boundary layer on each fin results in larger shear stress and added surface area. In the turbulent flow regime, the rough fin surfaces cause a leveling out of \( f \) which is common to rough channels.

A last OSF geometry with fin spacing that changed every third of the way through the coupon was tested to form a method of overall performance prediction. Just averaging the entire geometry to calculate a \( f \) and \( j \) over-predicts both of these performance values. A more representative method is to normalize the local performance of each section of fins and then average the result. This allows for each section to be represented and weighted correctly to make a more accurate performance average. However, if the overall geometry is averaged with both the over predicted \( f \) and \( j \), the ratio falls in line with expected performance ratio.

This study concludes that if AM OSF are used for compact heat exchanger applications, then operating at high Reynolds numbers will have additional pressure loss compared with smooth OSF. The increased pressure losses are more prominent than the augmented heat transfer at higher Reynolds numbers which severely lowers overall performance. To justify the use of AM for heat
exchanger applications other performance considerations could be improved such as weight, size, and durability to offset the increased pressure drop.

**Acknowledgements**

The authors would like to acknowledge and thank Nicolas Espinosa and Volvo for their technical input and their expertise on the design of the heat exchanger fins tested. The authors would also like to thank Corey Dickman in Penn State’s Center for Innovative Material Processing through Direct Digital Deposition (CIMP-3D) input on the additive build process and post-processing information. The authors would also like to thank Josh Langer (Penn State’s ExCCL research group) for assistance with measurement capability.
Chapter 5

Flow Field Measurements in a Metal Additively Manufactured Offset Strip Fin using Laser Doppler Velocimetry

Abstract

Metal additive manufacturing (AM) of heat exchanger enables custom and conformal designs for a wide range of applications. However, one challenge with metal AM is the resultant surface roughness formed when using this process. The goal in this study is to explore how this roughness impacts a commonly used heat exchanger surface called an offset strip fin (OSF). Two OSF of the same geometry are tested: one with an average fin roughness of 34 µm from metal AM and the other with an average fin roughness 2.5 µm, used as a baseline. The roughness from the metal AM process increased pressure losses and transitioned the flow to turbulent-like behavior at lower Reynolds numbers when compared with the smooth fin. Laser Doppler Velocimetry measurements captured the row number in the fin array where transition from laminar to turbulent-like flow occurred. The location of transition from low to high turbulence levels occurred earlier in the fin array as the Reynolds number was increased for the smooth and rough fins. Wake profiles of time-averaged axial velocity were similar between the rough and smooth fins, with the rough fins having higher levels of turbulence intensity and less symmetric wake profiles. Overall, this study indicates that a pressure loss penalty is associated with using metal AM OSF due to the resultant surface roughness.
Introduction

Thermal management is becoming increasingly important as power consuming devices are becoming smaller and more compact [93]. In order to ensure a correct operation temperature, heat exchangers are used to reject excess heat from the device, using one or more fluids. In aerospace applications, thermal management is becoming the forefront of challenges in designing electric aircraft. Heat generated internally to the aircraft from electric motors, high power consuming electronics, and battery charging/discharging must be dissipated external to the aircraft, without significant weight increase. The use of metal additive manufacturing to create custom and high-performance heat exchangers for a wide range of thermal management applications could be valuable to increase system efficiencies and may be a necessary step for electric powered aviation.

A common geometry used in compact heat exchangers known as the offset strip fin (OSF) has been well studied and documented for performance parameters. This heat exchange enhancement surface is commonly manufactured by folding or stamping sheet metal to create the fins and then is brazed onto a tube or plate wall that carries the other heat transfer fluid. The stamping process creates a smooth fin surface, with the only roughness coming from burred fin edges from the mold form having wear or surface fouling during operation [3], [4]. However, when an OSF is created using metal additive manufacturing (AM), the resultant surface roughness can be an order of magnitude higher than found with traditionally manufactured stamped fin surfaces [22]. This surface roughness could have an impact on the flow mechanics of offset strip fins and result in a deviation from the commonly used performance parameters: \( f \) (non-dimensional pressure drop) and \( j \) (non-dimensional heat transfer). Previous studies have quantified the impact of AM roughness on heat transfer and pressure drop performance for some geometries that are relatively simple [35], [36] and quite complex [21], [94], but few have explored heat exchanger fins that are common for air-side heat transfer enhancement.
The metal AM OSF tested in this study was produced using a laser-based powder bed fusion (PBF) process often referred to as selective laser melting (SLM). This process uses a high-powered laser that melts a thin layer of the powdered metal, ~30 μm, that forms the solid sections of the part. After a layer is complete, powdered metal is recoated over the previous layer and then the process starts again, resulting in a layer-by-layer additive process. Once the part is fully built, all remaining powdered metal must be removed from the part before a stress-relieving heat treatment is performed to avoid solidification of any remaining metallic powder. Additional material is sometimes required to support a part during the SLM process because the powdered metal cannot physically support a solid section of material above it. Furthermore, support material prevents the current layer being built from distorting caused by thermal stresses during the PBF process. This support material is removed after the build has been completed.

The metal SLM AM process is able to create unique and optimized shapes, but the major advantage of using offset strip fins is their robust yet simple design which is a reason that they are so commonly found in compact heat exchange applications. The OSF is able to enhance heat transfer by providing a high surface area for heat transfer but minimizes pressure loss due to the streamlined fin shape and low blockage area of the flow. Overall weight of an AM heat exchanger can be kept low by having the fins be as thin as possible and further optimized by distributing the fins spatially as shown with the variable spaced fins in Chapter 4. During the SLM process, a heat exchanger populated with the offset strip fin geometry will not require support material as long as all surfaces are angled above the self-supporting angle. Support material in locations that cannot be accessed and removed once the build is complete will cause a flow blockage and reduce overall performance. The AM OSF surface roughness characterization will only require a few surfaces since the fins are rectangular and perpendicular to each other. The metal AM roughness on circular channels and other more complex features, such as louvered heat exchanger fins [120], are harder to quantify because the angle continuously changes relative to the build direction [38], [43]. Finally,
a justification to explore this simpler heat exchanger geometry is the large amount of work on the fluid dynamics and heat transfer of OSF, so modeling them in AM provides a clean comparison regarding the effects of roughness.

**Previous Studies**

Offset strip fins have been studied extensively by a number of different researchers with Kays and London [47], Manglik and Bergles [9], and Webb et al. [6] being some of the most notable in providing overall performance correlations. Many fundamental and numerical studies have been performed on a slightly modified offset strip fin called staggered parallel plates [99]. The staggered plate geometry matches the offset strip fin geometry, but ignores the extra material that would normally be present on the channel walls from the fin stamping process. The staggered plate studies are usually scaled up to gain more measurement resolution and, in some cases, use optical and flow visualization techniques. For example, Mochizuki et al. [53], [55] scaled up the staggered plates to a thickness of 1.6 mm for dye-injected flow visualization in a water tunnel. Mochizuki found that as the Reynolds number was increased, wake shedding behind the plates started to occur further upstream in the fin array. A similar set up was used by Dejong et al. [56] captured a feathery-like wake behind the staggered plate which was reported to be small-scale transverse vortices. Dejong also captured local mass transfer on the fin surfaces by using the naphthalene sublimation technique which is analogous to local heat transfer values. Lee et al. [57] used a LDV technique for velocity measurements in offset strip fin cores as well as studied the flow field impact of various plate arrangements. The modified designs to augment heat transfer that Lee et al. [57] and Peng et al. [64] studied are a great example of an easily additively manufactured geometry when compared to traditional manufacturing methods.
Numerical studies of the OSF geometry and staggered parallel plates has shed insight into the flow physics and heat transfer with this type of geometry. A good example from Patankar et al. [3] shows how considering a fin with finite thickness can have an impact on the recirculation flow structures behind the fins. Other numerical studies in both two dimensional, three dimensional, [59]–[62] and unsteady flow, are able to capture some of the major flow structures around the fin. Pham et al. [66] was one of the first to use large-eddy simulation to capture the evolution of wake shedding in the streamwise direction of offset strip fins. Another example of this was shown by Saidi et al. [67] that found the wavy flow oscillation in their offset strip fin geometry occurred at a frequency of ~68 Hz for a Reynolds number of 1124. From a 2-D unsteady simulation, Zhang et al. [100] found oscillatory flow motion at a Reynolds number of 720 and a slightly increasing Strouhal number with increasing Reynolds number.

The flow field of an OSF geometry with the random-like roughness created from the AM process has not been explored because traditional manufacturing processes do not produce this result. However, structured roughness has been added to surfaces to enhance heat transfer performance for heat exchange applications which is well documented by Webb et al. [6]. Likewise, researchers such as Mahmood et al. [101] scaled up a dimpled surface and performed flow visualization, and spatially resolved heat transfer. The high resolution from the scaled-up geometry show vortex pairs shedding from the dimples and boundary layer disruption augmenting heat transfer; a similar result to a numerical simulation by Lu et al. [102]. However, the roughness created from the metal AM process is more random-like which is not analogous to the structured roughness that has been previously studied.
**Offset Strip Fin Designs and Characterization**

Two OSF of the same geometry are experimentally tested in this study: a metal AM rough OSF geometry created from the PBF process and a smooth polymer OSF created from a resin-based AM process. One challenge is that the surface roughness created from the metal additive process does not scale as the part is scaled up which necessitates experimentation on true scale metal parts. The OSF geometries including geometric definitions are shown in Figure 5-1. This current arrangement is similar to previously experimented geometries called staggered plates that are a simplification of the offset strip fin, as described by a few researchers such as Manson [99] and Mochizuki et al.[55]. The staggered plate geometry is commonly used in more fundamental studies because it still describes the physical behavior of the OSF geometry, despite not including the material typically left on the tube wall when stamping fins from sheet metal. The AM process can create the staggered plate geometry directly which results in an overall simplified geometry with same underlying heat transfer enhancement physics.

![Figure 5-1: Simplified offset strip fin with geometric definitions and schematic based off Manglik and Bergles [9] and linear laser path for measurements](image-url)
The metal AM OSF was fabricated on an EOS M280 PBF machine at Penn State University’s CIMP-3D using AlSi10Mg powder [84]. Figure 5-2 shows how the geometry for this study was angled and supported for the metal AM process. The support material and the part are built as a homogenous feature, and the support material is removed afterwards by traditional machining. This additional material used to support the external sections of the part during the build is required because any previously layer of pure powdered metal cannot support solid material above it when below the self-supporting angle of ~45°. A section of the outer wall of the fin array was left open and replaced with a borosilicate glass window for optical access during the LDV measurements. The plastic AM OSF was fabricated from a Photocentric Liquid Crystal Pro resin-based printer with a 137 µm lateral resolution and 100 µm vertical layer height. Daylight Pro Hard resin [106] was selected for its strong mechanical properties and post-cured with high powered UV curing lights.

Figure 5-2: Both the metal AM and plastic AM OSF samples were printed from the same geometric model with support material removed by machining.
Since both samples are additively manufactured, the as-manufactured geometries are listed in Table 5-2 and compared with the designed value. The measurements listed are taken with calipers that were validated for accuracy with measurement standards similar to the fin dimensions. The AM OSF physical dimensions are fairly consistent across the 20 measurements taken, with the metal AM fins having the most deviation from the design fin thickness.

Table 5-1: Fin dimension measurements of manufactured OSF geometries

<table>
<thead>
<tr>
<th>Measurement ± standard deviation</th>
<th>Thickness $t$</th>
<th>Spacing $s$</th>
<th>Length $l$</th>
<th>Height $h$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design value [mm]</td>
<td>0.50</td>
<td>3.30</td>
<td>5.00</td>
<td>5.00</td>
</tr>
<tr>
<td>Smooth Fin [mm]</td>
<td>0.507 ± 0.014</td>
<td>3.293 ± 0.014</td>
<td>4.877 ± 0.030</td>
<td>5.010 ± 0.026</td>
</tr>
<tr>
<td>Metal AM [mm]</td>
<td>0.625 ± 0.013</td>
<td>3.175 ± 0.013</td>
<td>4.997 ± 0.015</td>
<td>4.954 ± 0.018</td>
</tr>
</tbody>
</table>

The metal offset strip fin surface roughness was analyzed using a Nexview 3D Optical Profilometer which measures the height of the surface but operates on a line-of-sight measurement. The downward and upward facing surfaces of the metal fins were scanned and the images stitched to cover a large area of the fin at high resolution; shown in Figure 5-3. The scan has a lateral resolution of 0.803 µm and a total scan width and length of 1.477 mm. Measurement uncertainties of the surface roughness are calculated from multiple scans and machine specifications [121]. For the metal AM OSF geometry, the roughness values on the downward-facing surfaces were larger than the roughness on the upward facing surfaces which is a common result of the PBF process [122]. Additionally, the histogram of surface heights for the downward-facing surface is much wider than the upward-facing surface, indicating a large range of length scales.
The plastic AM OSF were also characterized with the same equipment as the metal AM fins. Table 5-2 summarizes the roughness values for all parts investigated. The plastic fins had a similar roughness on the upward and downward facing surfaces of the fins and a very low roughness value when compared to the metal parts. The roughness of the plastic fins is about twice that of traditionally manufactured folded aluminum fins used in commercial heat exchangers [112], but it is still an order of magnitude less rough than the metal AM fins. Post-processing of rough AM surfaces through abrasive flow machining has been studied by a few researchers [123]–[125], which can reduce the roughness, $R_a$, to less than 3µm. However, it is unclear if this process would be too aggressive on the thin heat exchanger fins, which is why a plastic part with exactly the same geometry was built as a smooth baseline. The metal AM OSF tested in this study are left *as-built* to understand the impact of roughness on the fluid mechanics.

Figure 5-3: Profilometer measurements show different roughness types on upward and downward facing surfaces on the metal offset strip fins.
Table 5-2: Measured surface roughness of additively manufactured fins

<table>
<thead>
<tr>
<th>Fin Material</th>
<th>Roughness property ± standard deviation</th>
<th>Upward facing [µm]</th>
<th>Downward facing [µm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>AM AlSi10Mg</td>
<td>$S_a$</td>
<td>14.6 ± 1.8</td>
<td>54.1 ± 4.2</td>
</tr>
<tr>
<td></td>
<td>$S_q$</td>
<td>20.3 ± 4.7</td>
<td>63.4 ± 9.3</td>
</tr>
<tr>
<td>AM plastic</td>
<td>$S_a$</td>
<td>2.4 ± 0.4</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$S_q$</td>
<td>3.1 ± 0.7</td>
<td></td>
</tr>
</tbody>
</table>

**Experimental Configuration**

**LDV Measurements**

The experimental test set-up for this study was used previously for similar designed coupons to take heat transfer and pressure drop measurements discussed in Chapter 4. There is no difference in the flow delivery system except that the incoming flow also has seeding particles for the LDV measurements. The seeding particle fluid used in this experiment is liquid Di-Ethyl-Hexyl Sebecat (DEHS) which is atomized into ~1 µm sized particles using compressed air. The seeded air passes through the coupon and then vents to atmosphere, similar to an open-loop wind tunnel.

The LDV beams are generated from a TSI Powersight laser head connected with optics for beam steering and focusing. The 532 nm wavelength beams are routed through optical cables to a 350 mm probe head with a 362 mm standoff distance. The length of the measurement volume created by the beam pair crossing point ($l_B$) is 1.35 mm and the diameter ($d_B$) is 117 µm, as shown in Figure 5-1 and qualitatively in Figure 5-4. The length of the beam is about 20% of the total fin height while the diameter of the beam is about 3% of the fin spacing; note that any recorded velocity
is spatially averaged through this measurement volume size. The probe head is placed on a traverse system that can move the probe in the X and Y directions with a minimum movement of 10 µm and in the Z direction of 5 µm. To avoid any backlash in the traverse screw, the probe head is only moved in one linear direction when moving to a new location. The optical window chosen was 3.2 mm (0.125 inch) thick borosilicate glass which is optically clear and scratch resistant. To avoid laser reflections back from the glass, the beams enter the glass at a 2° angle rotated about the x-axis as defined in Figure 5-4.

The LDV system is able to capture velocity measurements anywhere from 50 samples per second up to a few thousand depending on the flow seeding, laser power, and signal filtering. As a result, a study was done to find the number of recorded samples required to have a statistically converged average velocity and turbulence intensity at three different velocity and turbulence

Figure 5-4: (upper) Experimental set-up of additively manufactured offset strip fins with LDV beams. (lower) Showing scale of measurement volume compared to scale of fin geometry.
intensity (TI) values. Figure 5-5 shows that the average velocity and turbulence intensity does not significantly change as more samples are collected, but it takes about 15,000 samples for the measurement average to be within 1% of the measurement average at 100,000 samples. This indicates that a total sample count of around 20,000 to be sufficient for an accurate velocity and TI measurement.

Figure 5-5: A total sample count of 15,000 velocity measurements are required to be within 1%

The measurements with the highest level of turbulence intensity required the largest number of samples for the average value to converge to a statistically stationary result. The turbulence intensity value is calculated from the discrete velocities recorded by the LDV and is defined by Equation 35 and Equations 36.

\[ u' = u_i - \bar{u} \]  \hspace{1cm} (35)

\[ TI \ (\%) = \frac{100}{\bar{u}} \sqrt{\frac{1}{N} \sum_{i=1}^{N} (u'_i)^2} \]  \hspace{1cm} (36)
Friction Factor

The pressure drop characterization of the two geometries are compared against correlations by Mochizuki et al. [104] because it best matches with the geometries being tested. Mochizuki et al. define the hydraulic diameter, Reynolds number, and Fanning friction factor, \( f \), using Equations 37-39:

\[
D_h = \frac{4A_c}{P} = \frac{4sh}{2(s+h)} \tag{37}
\]

\[
Re = \frac{u_mD_h}{v} \tag{38}
\]

\[
f = \frac{\Delta PD_h}{2\rho V_{avg}^2 L} \tag{39}
\]

To get pressure drop measurements, static pressure tap ports are printed into the OSF geometry directly, as shown in Figure 5-4. The static pressure tap locations are centered between the fins in the first and last row. During the pressure drop tests, the static pressure and temperature is measured in the upstream plenum to calculate the air density using ideal gas law.

It should be noted that both OSF geometries being tested have a sheet of glass as the top lid of the flow channel. This is notable because the glass is extremely smooth (~4 to 50 nm [126],[41]) compared with the surfaces of the OSF and makes up \~1/7 of the total surface area. It would be difficult to decouple the effect of having this extremely smooth surface on only one side of the coupon but it could be assumed this would have a larger effect on the metal AM OSF geometry since this causes a larger difference in overall channel roughness.

Measurement Uncertainty

The reported measurement values from the experiments are Reynolds number, \( f \), mean axial velocity, and turbulence intensity. These calculated values come from the air mass flow rate,
physical dimensions of the geometry, and the LDV system, each with a measurement uncertainty. The bias uncertainty analysis was performed for one OSF coupon using a Kline-McClintock uncertainty analysis [87] and assumed constant for all other tests since all the instrumentation was kept constant between tested geometries. The precision uncertainty is calculated by taking ten repeated measurements of pressure drop at the same flow rate, but turning off the flow between tests to understand instrumentation repeatability. This process is performed for a low and high rate with the Student t-test confidence interval of 95% used to calculate measurement precision. Additionally, the OSF coupon was removed and reinstalled in the rig to ensure that the installation process was repeatable and accounted for in the total uncertainty analysis.

The Reynolds number has a maximum total uncertainty of 3.2% which occurs at the lower Reynolds number values. The $f$ values have an uncertainty of 9.1% and 3.7% at the lowest and highest Reynolds numbers, respectively. The $f$ uncertainty was kept to a minimum by using four different pressure transducers appropriately sized over the wide range of Reynolds numbers. The bias uncertainty of the velocity measured by the LDV system is provided by the manufacturer at 2%. The precision uncertainty of the velocity and TI measurement from traversing to the same location 10 times was 0.43% and 2.7%, respectively. The resultant total LDV uncertainty for velocity and TI are 2.1% and 3.4%, respectively.

**Results**

The friction factor, $f$, of the smooth and AM rough fins are calculated by measuring the static pressure drop across the OSF and shown in Figure 5-6. At mid-range Reynolds numbers, the smooth fins agree well with Mochizuki et al. [104] but deviates slightly at the lowest and highest Reynolds numbers. The AM rough OSF has a similar linear trend in the laminar region but is about 20% higher in $f$ than the smooth fin and respective correlation, shown in Figure 5-7. At Reynolds
numbers below about 500, the AM rough OSF has about a consistent 20% increase in f over the smooth fin geometry; however, beyond a Reynolds number of 500, the AM rough OSF starts to deviate from the smooth fin f resulting in a larger difference between the two OSF. At the highest Reynolds numbers, the smooth fin and AM rough fins have a difference in f of over 50%.

Figure 5-6: Friction factor measured for the metal AM rough OSF and the smooth OSF compared with correlations from Mochizuki et al. [104].

Figure 5-7: The difference in measured f between the smooth and rough OSF geometries.
One thing to note is that the correlation provided by Mochizuki et al. [104] does not account for the fin thickness in the laminar flow regime. This makes both OSF geometries have very similar correlations even though the fin thickness is slightly different. In the turbulent flow regime, the correlation does account for fin thickness and according to the correlation, an 11% increase in $f$ between geometries should exist. The larger deviation of $f$ between the smooth and rough fins from the experimental measurement, however, is likely due to the additional surface roughness of the metal OSF.

**Flowfield Measurements**

The two types of LDV measurements are denoted as *wakes* and *row-by-row*, which are all taken along the laser path line detailed in Figure 5-1 and shown from a top view in Figure 5-8. The laser path line is halfway up the height of the fin, $h/2$, and halfway along the fin’s length, $l/2$. The *wakes* capture the wake profile of the upstream row by measuring averaged axial velocity in rows 3, 7, and 17 with row locations described previously in Figure 5-4. The wake measurements are all taken on the laser path line by traversing the measurement volume by steps of ~100 µm across the fin spacing. Measurement locations that form the wake profile are normalized by the fin spacing and range from negative one to one, with zero being the centerline of the upstream fin.

The *row-by-row* velocity and TI measurement location is the location on the laser path line where the time-averaged axial velocity is maximum between the normalized fin spacing $Y = -1$ and $Y = 0$. This location is found by traversing the laser measurement volume starting at $Y = -1$ toward $Y = 0$ until the maximum velocity is reached. The measurement location is at the peak of the wake profile that occur around $Y = -0.5$ as shown in Figure 5-9. Velocity values at this location are useful to understand how the maximum velocity in the wake profile changes as Reynolds numbers and row location increase. Additionally, the location of highest velocity has low levels of TI and
indicates the onset of wake shedding if TI levels increase. These row-by-row measurements are performed for rows 3 through 21 at seven different Reynolds numbers for both the AM and smooth OSF geometries.

![Image](image.png)

Figure 5-8: Resin-based (smooth) AM OSF geometry tested showing non-dimensional wake distance

**Wake Profiles**

Normalized wake profiles for Reynolds number of 300 and 1000 are taken at row locations of 3, 7, and 17 for the smooth and rough metal AM OSF geometries as shown in Figure 5-9 and Figure 5-10. The physical distance between each wake profile data point is 101.6 µm (0.004 inch) which is slightly smaller than the beam measurement diameter of 117 µm. The mean axial velocity profiles for the smooth and rough fins in Figure 5-9 do not change significantly from row 3 to row 17 for the low Reynolds number of 300. Both the smooth and rough OSF geometries have maximum normalized velocities around 1.5 that occur near $Y = 0.5$ and $Y = -0.5$. In addition, the turbulence levels across the wake are low except at the locations closest to the fin surface, in both fin geometries. Wake shedding is not present at a Reynolds number of 300 for either OSF array.
suggesting that the increased $f$ between the smooth and rough fins must be a consequence of increased surface shear and additional shear surface area from the roughness.

Figure 5-10 shows the wakes for the smooth and rough AM OSF geometries at a Reynolds number of 1000. The smooth OSF wake profile is fairly symmetric with respect to the velocity and TI measurements. The maximum velocity increases from row 3 to row 7 with a narrower crest.
which is suggestive of flow acceleration due to boundary layer thickening on the fins. From row 7 to 17, the wake profile becomes flatter with a decreased maximum velocity. Additionally, the velocity deficit in row 17 at \( Y = 0 \) is reduced relative to row 7; both of these effects are suggestive of more turbulence in the flow. The TI levels for the smooth fin increase as the row number increases with a local minimum at the center of the wake which is caused by lower shear levels, similar to cylinder wake shedding experiments [127].

Wake profiles for the AM rough OSF geometry are less symmetric but still exhibit some of the same trends that are seen with the smooth fin geometry. For example, the turbulence intensity levels increase as the row number is increased. For the AM rough OSF, the velocity deficit from the upstream fin wake seems to be similar for rows 7 and 17 suggesting a faster wake recovery which could be cause by a more turbulent flow in an earlier row number when compared with the smooth fin. The wakes from the AM OSF geometry also exhibit less of a symmetric velocity and TI profiles behind the upstream fin (\( Y = -0.2 \) to \( Y = 0.2 \)) which is attributed to the different roughness surfaces on the upskin versus downskin sides. This is most apparent in row 3 at \( Y = -0.1 \) where the velocity deficit is larger than at \( Y = 0.1 \) due the rough side of the previous fin surface forming a larger boundary layer. Row 3 is most affected by the difference in roughness on each side of the fin because the velocity profile is still developing from the uniform velocity profile at the inlet of the fin array. As the flow progresses further into the array, the fins themselves have a larger impact on the bulk fluid motion than the roughness on the fin surfaces in the downstream wake.
Figure 5-10: Wake profiles of time averaged axial velocity and turbulence intensity of the (a) smooth and (b) AM rough (b) OSF geometries at Reynolds number = 1000.

Figure 5-11 directly compares the wake profiles at rows 3 and 17 for the smooth and rough fins at a Reynolds number of 1000. The symmetric wake profiles of the smooth fin highlights the asymmetry of the rough fin in both row locations. In row 3, the rough OSF has increased normalized velocity when compared with the smooth fin due to greater boundary layer squeezing early in the
fin array. However, near the end of the fin array in row 17, the rough OSF has equal or lower normalized velocity due to increased TI levels mixing out the wake. As a result, the wake profile in the downstream rows of the AM OSF becomes less distinct as the velocity deficit at \( Y = 0 \) is reduced.

![Graph showing normalized velocity profiles](image)

Figure 5-11: Comparison of wake profiles at row 3 and 17 for the rough and smooth OSF at Reynolds number of 1000.

**Row-by-Row Measurement**

Figure 5-12(a) shows how the maximum normalized velocity and TI for the smooth OSF geometry change with downstream distance and Reynolds number. For the low Reynolds number of 300, the maximum normalized velocity is relatively unchanged through the fin array. However, at a Reynolds numbers of 750 and above, the maximum velocity appears in an earlier row number...
which is attributed to the flow being squeezed by the developing boundary layers on the fins. This result was also shown in numerical studies where a maximum axial velocity is 11% greater in row 4 than in row 12 at Reynolds number of 1400.

At Reynolds numbers below 450 in Figure 5-12(b), the TI levels are low and fairly constant through the smooth OSF coupon, and the flow is completely laminar. However, as Reynolds number is increased to 600, the TI levels start to increase significantly. The increase occurs at an earlier row number in the fin array with increasing Reynolds number. This is due to the development of unsteady flow in the OSF array, which eventually transitions to turbulent-like behavior. It is interesting to observe that the TI levels decrease slightly after peaking in the first few rows. The reason for this can be revealed by inspecting the velocity histograms from the LDV system Figure 5-12(c). In the first few rows of the OSF array, the axial velocity histogram shows a very narrow distribution of velocities which results in a low TI. As the flow travels downstream, the highest TI appears when the velocity histogram shows a bimodal (double-peak) behavior. This creates a high TI because a large number of high velocities are far from the average velocity. It should be noted that the LDV measurement is only capturing one component of velocity, and oscillatory motion as described by Saidi et al. [67] would appear as a bimodal histogram since the velocity at a point changes direction. At a further downstream row, the velocity distribution loses its bimodal behavior and has a wide histogram profile. This lowers the TI levels when compared to the bimodal behavior since the tails of the PDF are generally lower velocities, but this row location still has a higher TI than the entrance of the coupon. In the last few rows of the smooth OSF the TI levels for all Reynolds numbers above 750 seem to have very similar levels around 9%. This is similar to the hot-wire measurement findings by Mochizuki et al. [53] who measured a TI of about 3% at row 5 and 10% at row 11 and 17 in a staggered plate geometry at a Reynolds number of 1500.
Figure 5-12: (a) The row-by-row velocity and (b) turbulence intensity for the smooth OSF, with (c) the velocity histograms at selected rows for a Reynolds number of 900.
The same row-by-row axial velocity and TI measurements were also performed on the metal AM OSF samples as shown in Figure 5-13. High levels of normalized velocity occur near the inlet of the fin array with a decrease in the downstream rows. This result matches with the full wake profiles, Figure 5-10, with the decreased maximum velocity in the downstream rows due to large TI levels lessening the magnitude of the wake. Additionally, the metal AM OSF turbulence intensity transitions to high TI levels at lower Reynolds number when compared with the smooth fins, with nearly complete transition to high TI for Reynolds numbers > 750.

Interestingly, at the highest Reynolds number tested, the TI levels are about the same for both the smooth and rough OSF geometries. This indicates that the flow patterns behind the fins are similar, yet, a difference in pressure loss was measured. As a result, the additional flow losses in the rough AM fins must be caused by additional shear stresses on the fin surface or added surface area from the roughness elements.
Figure 5-13: The row-by-row velocity and turbulence intensity for the metal additively manufactured rough OSF.
Conclusions

Pressure loss and LDV measurements were performed in a smooth and rough OSF geometry to understand how roughness from the AM process impacts the flow field in the fin array. The metal OSF geometry was fabricated from AlSi10Mg via the SLM process and the resulting surface roughness was an order of magnitude larger than the roughness found on traditional heat exchanger surfaces. The resultant surface roughness from the metal AM process caused the friction factor, $f$, to be larger when compared with the smooth fin geometry in both the laminar and turbulent flow regions. In the turbulent flow regime, the rough fin surfaces cause a leveling out of $f$ which is common to rough channels. The rough OSF showed characteristics of flow transition from laminar to turbulent-like flow sooner than the smooth OSF which was confirmed by the velocity and turbulence intensity measurements. The roughness from the AM process caused instabilities and wake shedding from the upstream fin causing higher turbulence levels at lower Reynolds numbers compared with the smooth OSF. Differing roughness on either side of the AM fins caused asymmetric wake profiles which where most apparent near the inlet of the fin array where the velocity profile is still developing.

Similar levels of TI were measured in the rough and smooth OSF at a low Reynolds number of 300 and a high Reynolds number of 1350. However, $f$ was larger in both cases for the rough AM OSF geometry. This suggests that the additional pressure losses from the resultant AM roughness must be caused by larger surface shear stress and added surface area for shear to occur when roughness is present and not additional losses due to turbulent dissipation in the fin wakes.

This study concludes that if AM OSF are used for compact heat exchanger applications, then additional pressure loss will exist compared with traditionally manufactured smooth OSF. Operating at lower Reynolds numbers helps avoid the high levels of increased $f$ for an AM OSF and it may be possible to offset the detriment of increased pressure drop by optimizing the heat
exchanger in other ways such as creating a conformally shaped heat exchanger that maximize overall heat transfer surface area and allows the heat exchanger to be placed closer to the thermal source, improving overall system efficiencies. To justify the use of AM for heat exchanger applications, other performance considerations could be improved such as weight, size, and durability to offset the increased pressure drop.

Acknowledgements

The authors would like to acknowledge and thank Corey Dickman and Ryan Overdorff at Penn State’s Center for Innovative Material Processing through Direct Digital Deposition (CIMP-3D) input on the additive build process and post-processing information.
Chapter 6

Conclusions

The work described in this dissertation investigated the impacts of using metal additive manufacturing for compact heat exchanger applications. This section summarizes the important conclusions from the work and recommends future studies. First, the major findings associated with the full-size metal additively manufactured heat exchangers are presented. Second, the offset strip fin heat transfer performance, pressure drop performance, and flow field measurements are discussed. Lastly, future work to gain more understanding of metal additive manufacturing for compact heat exchangers is summarized.

Full Size Heat Exchanger Conversion

The metal additive manufacturing (AM) process was able to replicate the geometry of a typical full-sized, traditionally manufactured, oil cooler used for small aircraft. Some design changes to the traditionally manufactured oil cooler were required for the AM process to avoid support material being required in any inaccessible location. However, the geometry significant to heat transfer and fluid mechanics, such as fin spacing, fin thickness, open flow areas, etc. were kept constant during the design conversion. The metal AM process was able to resolve these features but had performance deviations compared with the traditional design with twice the air-side pressure drop and about a 10% increase in heat transfer. Defects from the AM process were apparent on the air-side fins which included fin cracking and fin porosity. It is difficult to fully understand the defect’s impact on overall heat exchanger performance, however, it is hypothesized that these defects do not improve overall performance, and negatively impact heat transfer more
than pressure drop because the cracks and porosity of the heat exchanger fins remove the thermal conduction path from the base of the fin. If the fin was split halfway up its span, \( h/2 \), then this might not have a large impact on heat transfer since it could be assumed that this is at the adiabatic plane of the fin, if the heat transfer conditions on the fin are symmetric. However, it was observed that the cracks were biased toward one side of the fin and not at the fin mid-span. The porosity has a similar effect as the cracked fin with less solid material to conduct heat into the air stream. The main goal of the fin is to add more surface area into the air-side flow path with a large temperature difference; fin porosity would not help achieve this. In addition to the observed defects, the measured surface roughness from the AM process was an order of magnitude greater than the measured roughness on the traditionally manufactured geometry. It is believed that this surface roughness which exists on all features of the AM heat exchanger causes additional surface shear and earlier transition to turbulent-like flow.

The direct replication of an existing heat exchanger geometry using AM without design improvements does not take full advantage of AM, but demonstrates the current capabilities of the AM process. Furthermore, the metal AM heat exchanger was hydrostatically tested to twice the rated burst pressure without failing which gives confidence that AM can be used in critical applications. The modification of the original heat exchanger design would yield benefits, especially when customized for a specific application but will have larger pressure losses than expected due to the surface roughness.

**Offset Strip Fin Heat Transfer, Pressure Loss, and Flow Field Measurements**

The impact of surface roughness on pressure loss and heat transfer has been studied at a fundamental level which usually is driven by an application and manufacturing process. However, roughness on heat exchanger fins has not been studied because traditionally manufactured heat
exchanger fins are smooth, which will not be the case for metal additively manufactured (AM) heat exchangers. To understand this effect, a test rig was designed and baselined to experimentally test the performance of AM heat exchanger surfaces called offset strip fin (OSF). The metal OSF were fabricated from 316L stainless steel alloy via the powder bed fusion process and compared with smooth fin counterparts and traditional correlations. The fin surfaces were characterized through surface scans and imaged with a scanning electron microscope to view the resultant surface roughness from the AM process which was an order of magnitude larger than the roughness found on traditional heat exchanger surfaces. The resultant surface roughness from the metal AM process, caused the friction factor, $f$, to be larger when compared with a similar smooth fin geometry in both the laminar and turbulent flow regions. It was previously thought that surface roughness does not impact $f$ for low Reynolds numbers, but the constant reforming of the boundary layer on each fin results in larger shear stress and from the added surface area. In the turbulent flow regime, the rough fin surfaces cause a leveling out of $f$, which is common in rough channels. The OSF array with the more tightly spaced fins had larger $f$ when compared with the AM OSF of wider spaced fins.

Interestingly, the traditional smooth fin correlations correctly predicted that the overall performance of the AM fins would be higher for the tightly spaced fins in the laminar regime than for the wider spaced fins. Similarly, in the turbulent regime, the smooth fin correlations correctly predicted that the wider spaced AM fins would have higher overall performance. This is useful from an engineering standpoint to have confidence that trends between smooth and rough OSF geometries are similar, but the rough OSF has larger pressure losses.

A third OSF geometry, with fin spacing that changed every third of the way through the coupon, was tested to form a method of overall performance prediction. A simple approach of just averaging the entire geometry to calculate $f$ and $j$ over-predicts the both these performance values. A more representative method is to normalize the local performance of each section of fins and
then average the result. This allows for each section to be represented and weighted correctly in order to calculate a more accurate performance average. However, if the overall geometry is averaged with both the over predicted $f$ and $j$, then the ratio falls in line with expected performance ratio.

To gain a better understanding of the impact that the resultant surface roughness from the metal additive manufacturing (AM) process has on an offset strip fin geometry, velocity measurements inside an AM OSF geometry were performed. These measurements are challenging due to the fact that the AM roughness does not scale as the geometry is scaled. Consequently, these measurements taken are the first of its kind to explore the velocity in a true-scale AM OSF array. This study also explored the velocity and pressure loss measurements of a true-scale resin-based additive manufacturing process to create the exact same geometry with a smoother surface for comparison. The resin-based fin surfaces had roughness comparable to traditionally manufactured fins while the metal AM fins had surface roughness values more than ten times higher. The resultant surface roughness from the metal AM process caused $f$ to be larger when compared with the smooth fin geometry in both the laminar and turbulent flow regions. In the turbulent flow regime, the rough fin surfaces cause a leveling out of $f$ which is common to rough channels. The rough OSF showed characteristics of flow transition from laminar to turbulent-like flow sooner than the smooth OSF which was confirmed by the velocity and turbulence intensity measurements. The roughness from the AM process caused instabilities and wake shedding from the upstream fin causing higher turbulence levels at lower Reynolds numbers compared with the smooth OSF. Differing roughness on either side of the AM fins caused asymmetric wake profiles which were most apparent near the inlet of the fin array where the velocity profile is still developing.

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must be caused by larger surface shear stress and added surface area for shear to occur when roughness is present and not additional losses due to turbulent dissipation in the fin wakes.

These studies conclude that if AM OSF are used for compact heat exchanger applications, then additional pressure loss will exist compared with traditionally manufactured smooth OSF. Operating at lower Reynolds numbers helps avoid the high levels of increased $f$ for an AM OSF and it may be possible to offset the detriment of increased pressure drop by optimizing the heat exchanger in other ways such as creating a conformally shaped heat exchanger that maximize overall heat transfer surface area and allows the heat exchanger to be placed closer to the thermal source, improving overall system efficiencies. To justify the use of AM for heat exchanger applications other performance considerations could be improved such as weight, size, and part durability to offset the increased pressure drop.

**Recommendations for Future Work**

The conclusions from these studies show the benefits and consequences of using metal additive manufacturing for heat exchanger applications, but also warrant future studies. Understanding the fundamental physics and interaction that the AM surface roughness has on the fluid flow over the fin surfaces would shed insight into the reasoning behind the increased pressure losses. Measuring boundary layers on the fin surface at different locations in a fin array and at multiple Reynolds numbers would give a better understanding of the surface roughness impact. However, performing these measurements at true scale is extremely difficult; so, scaled up OSF geometries with representative AM roughness is required. Furthermore, considerations to the differing type of roughness dependent on the material used in the AM process is not well understood. The majority of public studies only test AM geometries created from the same material, and which roughness parameters such as $S_a, S_q$ and $S_z$ are most important for heat transfer fins
would be valuable information. Related to this, another area of research involving numerical simulations to accurately predict the performance of rough AM OSF which could reduce the need for experimental testing and result in numerical optimize the AM OSF geometry. However, the random-like roughness from the AM process is not easily implemented in current CFD codes and would need validation through experimental research. This validation could be achieved by comparing scaled up experimental results to CFD results with the AM roughness.

A research area that is only recently being explored is the use of surface finishing methods to reduce roughness on AM parts. If all the roughness could be removed from the AM OSF, then traditional correlations would accurately predict the performance of the heat exchanger and the increased pressure losses would be eliminated. However, many of the surface finishing techniques do not remove material evenly and could cause other unpredicted changes in performance. Some methods of mechanical flow abrasion might be too aggressive for the thin heat exchanger fins and cause cracking, bending, or removal of the entire fin. This would necessitate that the fins are built thicker than required from a heat transfer perspective and thinned by the abrasive flow. However, certain sections of the heat exchanger that are stronger and designed to hold pressure, such as the liquid-side, could handle the flow abrasion force and result in reduced pressure losses.

Another area of research is to create correlations that account for the AM roughness could be formed for heat exchanger designs similar to those formed for traditionally manufactured OSF. If the ratio of surface roughness to fin spacing could be incorporated into the traditional smooth fin correlations, then the correlations would work for a wide range of geometries and still capture the underlying physics of OSF. Other fin types besides OSF could also have correlations but defects on more complex fins such as louvered fins shown in the Appendix pose additional issues with performance prediction. However, if the fundamental physics of roughness on heat exchanger surfaces is understood, then analogies between different fin types could predictively link the overall performance of untested fins.
Last, a promising use of metal additive manufacturing is to find existing heat exchanger designs and figure out the design inefficiencies caused by the current, traditional manufacturing process. These inefficiencies such as flow recirculation, low fin efficiency, local hot spots, and over designed safety factors could be easily corrected with metal additive manufacturing. Iteratively improving the heat exchanger design will result in a higher performing heat exchanger part. The last step for any heat exchanger is to make it conformally shaped for compactness and improved performance. Other design changes for certain applications could also be easily added with metal additive manufacturing such as, thicker fins at the inlet for increased durability to combat foreign object damage (FOB), fins could be intentionally made thinner and act as fatigue samples to indicate when the actual fins are going to crack, embedded instrumentation could easily be installed to monitor heat exchanger life with temperature and pressure sensors, and some fins could act as struts throughout the heat exchanger to reduce damage from thermal and pressure cycling. These examples might not be areas of research, but relate more to practical design where metal additive manufacturing surpasses traditional manufacturing techniques.
References


2013.


Appendix

Comparison of Metal Additively Manufactured Louvered Fins from Identical CAD Geometry*

The louvered fin geometry is a commonly used heat transfer surface found in compact heat exchangers and augments heat transfer by disrupting the hydrodynamic boundary layer which mixing the flow. The fins are typically manufactured from sheet metal and the louver is created by pressing and turning a small section of the fin. Traditionally manufacturing processes are able to create this geometry in mass quantities with different fin parameters being louver height, louver pitch, louver angle, fin thickness, fin pitch, and fin height that are used for performance correlations.

The louvered fin geometry is difficult to metal additively manufacture directly because the parts of the louvered surface falls below the self-supporting angle required for SLM. The louver fin shape can be modified as described in Chapter 3 to limit the amount of surface area below the self-supporting angle to improve the final built quality. However, even with the modified louver fin geometry defects on the curved features of the louvers can still result. Figure 0-1 shows sectioned views of two different heat exchangers printed from the same geometry. Build I louvered fins manufactured well and the full louvered fin curvature is apparent. The louvered fins in Build II did not print well and had the largest defects on the curved sections of the louver. Figure 0-2 shows how the defects on the downward facing surfaces of Build II are more prominent than on the upskin surfaces. The defects on the louvers are visible throughout the heat exchanger but are not always identical to each other which makes performance prediction difficult. Quantifying these defects is difficult because a surface scan would only capture to location roughness of the defect but not the shape and a CT scan could capture the overall shape of the defect, but not each defect.

is identical. Furthermore, another SLM machine and set of processing parameters might produce different results than presented from Build I and Build II.

![Build I and Build II](image)

Figure 0-1: Internal cross section of louvered heat exchanger fins from two different heat exchangers, bottom right fin of Build I was removed for inspection.

![Upskin and downskin surfaces](image)

Figure 0-2: Upskin and downskin surfaces of the louvered fins from build II

The differences in as-built geometries played a role in the heat transfer and pressure drop performance of the two heat exchangers. The Build II heat exchanger had a 72% increase in
pressure drop and a 34% increase in overall heat transfer conductance when compared with Build I. Difficulty in predicting the performance of traditionally manufactured louvered fins exists which only becomes more of a challenge when roughness and defects become additional design variables.
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