CHAPTER FIVE

Review of advances in convective heat transfer developed through additive manufacturing

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Abstract
Opportunities exist to make revolutionary impacts on convective heat transfer designs for high temperature applications as the development of three-dimensional metal printing technologies continue to advance. Not only can complex heat exchange designs be further exploited because of manufacturability, but the development of these new designs can be done cheaper and faster. This chapter highlights the importance of understanding the additive manufacturing (AM) processes in light of how the AM processes impact heat transfer, particularly for millimeter-sized channels and thin
fins, which are used in many convective cooling applications. In particular, the surface roughness inherent in the AM process is a key contributor to performance. Surface roughness is a strong function of the build parameters as well as the component build direction. In addition to proposing a new correlation for predicting friction factor and heat transfer of internal channels and thin fins made using AM, a range of channel shapes and novel geometries are highlighted. Finally, this review details how AM has been used to build functional heat exchangers along with the expected performance relative to conventional designs.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>AR</td>
<td>aspect ratio of channel</td>
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<tr>
<td>Dh</td>
<td>hydraulic diameter</td>
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<td>Dh,design</td>
<td>design intent hydraulic diameter</td>
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<td>DL</td>
<td>laminar equivalent diameter</td>
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<td>f</td>
<td>Darcy friction factor, $f = \frac{\Delta P}{\frac{1}{2} \rho u^2}$</td>
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<td>H</td>
<td>channel height</td>
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<td>h</td>
<td>fin height</td>
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<td>j</td>
<td>Colburn j-factor</td>
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<td>J</td>
<td>objective function</td>
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<td>Ka</td>
<td>sandgrain roughness</td>
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<td>L</td>
<td>channel length</td>
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<td>l</td>
<td>fin length</td>
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<td>n</td>
<td>number of datapoints</td>
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<td>Nu</td>
<td>Nusselt number, $hD_h/k$</td>
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<tr>
<td>NuDh</td>
<td>Nusselt number calculated using $D_h$ as characteristic length, $hD_h/k$</td>
</tr>
<tr>
<td>Nu√Ac</td>
<td>Nusselt number calculated using $\sqrt{A_c}$ as characteristic length, $h\sqrt{A_c}/k$</td>
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<td>Pr</td>
<td>Prandtl number</td>
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<tr>
<td>Ra</td>
<td>arithmetic mean roughness</td>
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<td>Re</td>
<td>Reynolds number, $uD_h/\nu$</td>
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<td>Re√Ac</td>
<td>Reynolds number calculated using $\sqrt{A_c}$ as characteristic length, $u\sqrt{A_c}/\nu$</td>
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<tr>
<td>KU</td>
<td>kurtosis of roughness</td>
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<td>Rq</td>
<td>root-mean-square roughness</td>
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<td>mean roughness depth</td>
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<td>fin spacing</td>
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<td>Sa</td>
<td>area based average surface roughness</td>
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<td>St</td>
<td>Stanton number</td>
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<td>t</td>
<td>fin thickness</td>
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<tr>
<td>W</td>
<td>channel width</td>
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<td>z_ref</td>
<td>reference surface height</td>
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<td>z_surf</td>
<td>roughness height</td>
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1. Introduction

Through decades of research including theory, experimentation, and computation, the heat transfer community has developed a fundamental understanding of highly-engineered surfaces for effective convective heat exchange for a myriad of applications. These designs have evolved to ones that have high surface area densities and ones that promote turbulent transport with the constraints being the amount of pumping power available, space availability, and the ability to manufacture the designs. As advances occur in additive manufacturing (AM), also known as metal 3D printing, new geometric designs are enabled that have the potential of increasing complexities and performance of convective heat exchange while reducing part number, weight, manufacturing waste, and even costs depending upon the application. It is for these reasons that particular industry sectors, such as aerospace and energy, are embracing this printing technology. In 2019, the additive manufacturing industry was estimated to be worth over $9B [1], which is only a fraction of the $12 trillion manufacturing industry; however, industries are moving quickly in using AM for rapid prototyping to such applications as tooling and even production parts. New companies in this space are emerging and even more important are new and improved AM capabilities. The AM process also lends itself well to what many companies are moving toward and that is a digital thread for their products.

There is a suite of AM options for metal, including selective laser melting (SLM), electron beam melting (EBM), and selective laser sintering (SLS) as technologies capable of printing parts that are particularly exciting for opening up new heat exchange geometries. Specifically, sintering methods are of
interest because sintering allows parts to be manufactured from high temperature alloys. However, because the metal powder is sintered, the microstructure can differ from conventional manufacturing using the same material. Coupled with optimization methods, AM offers a wealth of opportunities to create cooling geometries that would not be possible using traditional manufacturing methods.

With each printing technology, there are associated process parameters that ultimately result in a component’s geometric tolerances, surface morphology and roughness, and component quality. The intention of this review is not to evaluate the effect of all possible process parameters on the overall part quality. This review will, however, discuss some of the influential process parameters including the build direction, laser scan speed, and surface contouring and how those resulting surfaces affect the heat transfer and pressure loss characteristics for small channels, in particular. It is important to recognize that AM is not the answer to everything but, as advances continue to develop, 3D metal printing technologies will continue to widen the opportunities and application spaces. One key consideration, in terms of convective heat exchange, is that the component surfaces using AM are inherently different from that of smooth, machined surfaces. For external surfaces, post-processing is possible through a number of means, but with that comes increased costs. For small internal channels that are often used to convectively cool components, current post-processing methods are either nonexistent or limited especially as passages become more complex, intricate, and smaller. For this very reason, it is important to develop an understanding of how AM surfaces impact convective heat transfer and the associated pumping power, which is the focus of this chapter.

Given the growing interest and investments in developing AM, the community has already demonstrated the use of novel geometries to increase heat exchange ranging from microchannels to full heat exchangers. For the purposes of this review, we will focus on using AM for advancing heat exchange components including the following topics: (i) key AM parameters that affect the part quality and ultimately heat exchange and pressure losses for small channels; (ii) novel surfaces enabled through AM; and (iii) full heat exchanger studies that have used AM. This chapter will also provide correlations based on a wide array of experiments that have been conducted using millimeter-sized channels that have been made using additive manufacturing, which can assist the community in predicting the augmentation effects of the inherent surface roughness in AM parts. Although methods are being developed to computationally simulate such rough
surfaces, which is difficult at best, these methods are still highly reliant upon the experimental results given the random nature of the inherent AM roughness features.

2. Metal printing methods and post-processing

The use of metals for the fabrication of AM heat exchange components is appealing to the energy and aerospace communities, in particular, due to the development of high temperature metal alloys, production of microscale features, high specific strength and stiffness relative to other non-metal materials, and desirable thermal properties. Of the seven categories of AM processes characterized by the American Society of Design and Materials group [2], metal 3D printing typically occurs through metal powder bed fusion (PBF) techniques. PBF is predominately used given its ability to resolve small features for true-scale components, material availability, surface morphology, postprocessing methods, and cost relative to the other metal AM processes [3]. This section of the paper describes the build sequence of a typical PBF process and highlights the key differences between the different PBF processes. Post processing methods and common part defects will also be described and related to how those defects can impact the thermal material properties of an AM component.

2.1 Printing methods

Powdered bed fusion (PBF) processes are distinguished by the energy sources used in the process [4] such as those already mentioned including selective laser melting (SLM), electron beam melting (EBM), selective laser sintering (SLS), and direct metal laser sintering (DMLS). The SLS and DMLS processes are nearly identical except that the DMLS process exclusively uses metal compared to the use of nonmetal materials for the SLS process. A typical PBF process is seen in Fig. 1, where powdered metal is selectively melted from an energy source (either electron beam or laser-based) that follows a profile outlined from the design intent digital model [4]. In more detail, the powdered metal is melted by the energy source to form a liquid pool, also known as a melt pool, which then rapidly cools and solidifies. After the energy source has scanned a cross-sectional profile for a particular metal layer, the build platform is lowered to deposit another layer of powder using a recoater blade or another device such as a roller. This process is then repeated until the component is built [5]. A benefit in using the PBF process relative to other metal AM processes is that unfused residual
powder can be recycled and blended with virgin powder for a future build. However, it must be noted that the use of recycled powder can impact the repeatability and quality of an AM part [6]. Ahmed et al. [6] observed that external surface roughness increased from 8% to 17% depending on the amount of times the powder was recycled relative to virgin powder.

In comparing the various metal additive manufacturing energy sources, the EBM process uses a beam that melts the metal powder in a vacuum. The build plate for the EBM process is preheated to prevent process instabilities and then maintained at a specific substrate temperature. The higher temperatures experienced in the EBM process results in more favorable material and microstructural properties compared to laser sintering processes (SLS and DMLS) [7]. An important distinction between EBM and other PBF methods are that higher surface roughness and reduced feature resolution [8,9] is experienced using EBM relative to SLM and DMLS.

The SLM and DMLS processes use high power lasers rather than an electron beam to either melt or sinter the powder to form a component [10]. As the technology of SLM and DMLS has improved, the SLM and DMLS processes are considered to be identical processes for fully melting the metal powder. The terms SLM and SLS are used interchangeably. DMLS is regarded as the most commonly used PBF process in the AM industry [11]. Throughout the remainder of this paper, we will use the term AM to represent 3D metal printing with the majority of the results specifically obtained using DMLS to manufacture the various heat exchange designs.

After printing, postprocessing techniques such as heat treatments are used to reduce residual thermal stresses that develop during the build to produce

![Fig. 1 Schematic of a general PBF process showing the substrate, energy source, component, powder, and recoater blade/roller.](image)
more favorable material properties and prevent further warpage that can occur when removing the part from the build plate [12]. Heat treatment techniques are applied directly to the part, while attached to the build plate. For further detail, DebRoy et al. [12] provides an overview of the material properties that are influenced by the postheat treatment techniques. After the parts have been heat treated, the parts are postprocessed by removing the components from the build plate as well as removing any support structures that were designed into the part for building purposes (important for unsupported surfaces). In addition to residual thermal stresses, parts from any PBF process can experience multiple defects that are not present in traditional subtractive or casting manufacturing methods. Such defects include porosity, delamination, surface defects, and microstructural impurities [13]. Additional postprocessing methods can occur to reduce surface defects through surface treatments and minimize porosity by use of a Hot Isostatic Pressure (HIP) treatment [14].

Common treatments to reduce the surface roughness on external surfaces include conventional mechanical methods such as sand blasting and mechanical grinding or acid/electrical methods such chemopolishing and electropolishing. The complex three-dimensional internal passages that AM can fabricate for highly effective heat exchange makes conventional post-processing surface treatment methods difficult or even impossible to apply for internal channels [15]. In some cases, however, internal surface treatments are done using either electropolishing [16] or chemopolishing [17,18].

2.2 Powdered metals and properties

There is a wide range of material alloys that have been made available in powders for 3D printing including nickel (Ni) [19], cobalt (Co) [20], iron (Fe) [21], titanium (Ti) [22,23], and aluminum (Al) [24] with a complete review given by Ngo et al. [25]. Nickel-based materials such as Inconel 718 are among the most common material for heat transfer components seen in literature [26–31] because of the applicability at high temperatures. Understanding the influence an AM process has on component properties, part porosity, and how those properties change is an important consideration for AM heat transfer applications [32,33]. Thermal properties such as thermal conductivity can differ between a sample of the AM powder feedstock and that of an AM part [32]. Furthermore, due to the possibility of porosity in AM parts, the as-built thermal conductivity of an AM sample cannot be assumed to be equal to the thermal conductivity of the same bulk material used for conventional manufacturing methods [34].
Studies such as Alkahari et al. [32] observed that the thermal conductivity of AM samples decrease with increasing porosity. Simmons et al. [34] observed the same result and also showed that the thermal conductivity of a sample can change depending on the orientation of the sample when specific laser processing parameters are used. The study by Simmons et al. included fabricating several AM samples and showed that once critical values of the laser process parameters (energy density and laser speed) are met, the thermal conductivity depends upon the orientation of the sample with respect to the surface of the substrate. Their results indicated the thermal conductivity of samples built parallel to the plate was lower relative to samples built in a vertical orientation. The work done by the authors of this paper, however, showed that there was not change in the thermal conductivity depending upon build orientation if the standard, recommended process parameters are used for AM.

Sélo et al. [35] showed for Al-based parts that different heat treatments can change the thermal conductivity by as much as 44% compared to the as-built, nonheat treated part. Additionally, the study found that heat treatments can reduce the anisotropic behavior of the thermal conductivity. Similarly, Butler et al. [36] observed higher thermal conductivity for heat treated Al-based samples compared to as-built (nonheat treated) samples. For these reasons, it is important to consider the influence processing parameters, orientation of a sample, and heat treatment has on the resulting thermal properties.

Table 1 presents measured thermal conductivity values that have been reported in the literature for AM specimens (not powder feedstock). The limited amount of thermal property data, shown in Table 1, highlights the need for future measurements of as-built AM thermal properties.

2.3 Design considerations and process parameters

AM parts are built using a design intent digital model to construct the desired part. There are additional steps needed prior to building a component from the powder, which is briefly described in this section in terms of component accuracy and surface roughness and morphology.

The procedure of tasks required to fabricate an AM component are what is known as a set of process parameters. Some of the key parameters include the laser scanning speed and direction, recoater blade direction, contouring strategy, powder thickness, the laser power, and laser contouring. Changing the energy input (laser speed and laser power) can directly impact the melt
pool resulting to changes in surface morphology and structural properties of AM components [12,16,42,43]. Hanzl et al. [44] provides additional detail in how these parameters can influence material properties while Calignano et al. [45] provides further information on surface morphology.

Powdered bed fusion methods slice the digital design model into multiple two-dimensional cross-sections where each slice represents an individual layer generally between 20 and 40 μm. As illustrated in Fig. 2, the power source such as the laser scans the powder layer whereby the distance between the laser beam passes is known as the hatch distance as measured between the beam center. Hatching occurs in the region of the part that contains no exposed surfaces and is segmented into several specific regions known as the core and skin. Depending on the severity of unsupported surfaces, unique hatching patterns and different energy inputs are applied to a part. Surfaces fabricated at angles other than perpendicular (90°) to the surface of the substrate contain unsupported/overhanging surfaces. These unsupported surfaces are known as either a downskin surface, which is built upon powder, or an upskin surface, which is built on solidified material from a previous layer as illustrated in Fig. 2. Energy input is generally adjusted for the laser depending on if the surface is a downskin or upskin surface.

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<tbody>
<tr>
<td>316L SS</td>
<td>40</td>
<td>14.1 [34]</td>
<td>13.9–14.2 [37,38]</td>
<td>DMLS</td>
<td>Solution Annealed Heat Treatment</td>
</tr>
<tr>
<td>Inconel 718</td>
<td>40</td>
<td>9.8 [29]</td>
<td>10.0 [39]</td>
<td>DMLS</td>
<td>Solution Annealed Heat Treatment</td>
</tr>
<tr>
<td>Hastelloy X</td>
<td>40</td>
<td>10.0</td>
<td>11.1 [40]</td>
<td>DMLS</td>
<td>Solution Annealed Heat Treatment</td>
</tr>
<tr>
<td>AlSi10Mg</td>
<td>25</td>
<td>122.5 [35]</td>
<td>113.0 [41]</td>
<td>DMLS</td>
<td>No heat treatment, value is averaged from both build directions</td>
</tr>
<tr>
<td>AlSi10Mg</td>
<td>25</td>
<td>146.5 [35]</td>
<td>113.0 [41]</td>
<td>DMLS</td>
<td>Solution Annealed Heat Treatment, value is averaged from both build directions</td>
</tr>
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Table 1: Thermal conductivity of AM samples compared with non-AM feedstock.

Review of advances in convective heat transfer
While hatching provides the infill to a part, a laser contour resolves the edges of a part by performing a laser pass that follows the part outline. Similar to hatching, a contour pass follows the same upskin and downskin naming convention. Unlike typical hatching, contours are sometimes applied several times to the current layer; however, this depends on the AM machine and the process parameters that are set. As seen in Fig. 2, the first contour typically follows a dimensional error correction outline known as the beam offset. The beam offset compensates for dimensional error related to the laser spot diameter [45].

Optical profilometer measurements are presented for an example downskin, Fig. 3A, and upskin, Fig. 3B, whereby the two surfaces are from a representative AM coupon that was built at a 45° angle. The reason for the differences shown in Fig. 3 has to do with gravitational effects on the melt pool as the component is being built using the laser powdered bed fusion process. The downskin surface is clearly very rough with large protrusions while the upskin surface also shows varying levels of roughness albeit much less. In comparing these two optical scans, it is important to note that the legend scale starts with a negative value, which relates to the difficulty of determining the datum for a surface. The arithmetic roughness levels, which will be described next, is also listed for the two surfaces in which a datum line was defined based on a curve fit of the scanned data in the point cloud.
The data in Fig. 3 has important consequences when considering the build direction of an AM component but can also be an opportunity should one side of the design require higher heat exchange, which would be promoted by the downskin side.

The term surface roughness can be quantified using multiple definitions such as arithmetic mean roughness ($R_a$), root-mean-square roughness ($R_q$), mean roughness depth ($R_z$), skewness ($R_{sk}$), and kurtosis ($R_{ku}$) [47]. Like many roughness studies, most of the AM literature follows convention by citing the arithmetic mean roughness [26,29,31,46,48]. The importance of this variable for convective heat transfer principles was validated by the work conducted by Stimpson et al. [31] who assessed numerous ways of characterizing roughness levels for scaling purposes of the heat transfer and friction factor specifically for AM surfaces. The arithmetic mean roughness, $R_a$, represents the average deviation of the surface relative to a reference value, which is a one-dimensional measurement. The arithmetic mean roughness values reported throughout this paper for those studies associated with the authors’ research have been calculated using computed
tomography (CT) scan data unless otherwise noted as given by the mathematical definition in Eq. (1).

\[ R_a = \frac{1}{n} \sum_{i=1}^{n} |z_{surf} - z_{ref}| \]  

The arithmetic mean roughness values for circular channels reported in this paper are calculated using a method similar to Klingaa et al. [49]. The inherent curvature of a circular channel makes defining a reference line challenging. For example, in the case of a circular channel depending upon the build direction, an ellipsoid may be more representative of the surface profile in which the ellipsoid would define the reference line (datum).

### 2.4 Part accuracy and surface morphology

A host of reasons exist as to why AM parts are inherently rough that include poor melt pool control, powder particles that are attracted to the melt pool during the process, and the stair stepping effect resulting from the layerwise process. Accordingly, there are a range of distinct surface roughness types present in AM that range in scale from small partially melted particles, which range between 20 and 60 μm, to large dross formations illustrated in Fig. 4. A general overview on the types of AM surface roughness include partially melted particles, hatch spacing, balling, lack of fusion, and dross. All of these types of roughness in Fig. 4 depend upon the process parameters and can even be tailored to enhance heat transfer [53]. Partially melted particles are a result of the energy not being sufficient enough to fully melt the powder feedstock [54]. Hatch spacing can also result in distinct roughness patterns caused by the spacing between multiple single laser tracks illustrated by the large rib-like ridges between the laser passes in Fig. 4 [50]. The work of Snyder et al. [53] amplified these ridges by manipulating the hatch spacing and several other laser parameters to influence pressure loss and heat transfer in AM channels.

Balling, also illustrated in Fig. 4 is caused by flow (Rayleigh) instabilities within the melt pool resulting in spherical balls being broken off from the melt pool track [51]. The amount of balling present, increases with increasing scan speed [54]. In the case of lack of fusion, there is insufficient energy used to fully fuse the powdered metal that can create voids within a part and contribute to porosity [55]. The largest roughness type in Fig. 4 is dross formation that results from surplus material created from improper control of the melt pool. Dross formation is usually seen on overhanging surfaces (downward facing) as result of gravity and the surfaces being built upon unsupported powder feedstock.
Fig. 4  Range of roughness types experienced in AM parts which are impacted by design consideration and the AM process [27,50–52].
To fabricate AM components, build considerations include the component build direction and placement on the substrate, powder removal, and placement of support structures. Factors such as the location of the component on the substrate can influence the deviation from design intent [56–59] and was shown by Kleszczynski et al. [56] to be attributed to the laser incidence angle. Oter et al. [58] and Sendino et al. [59] observed that depending upon where the components are fabricated relative to the center of the substrate plate, various levels of surface roughness and deviations occur. A key consideration is the placement of support structures that add to conduction paths during the processing directly impacting the thermal stresses and resulting warpage [60]. Observations from Oter et al. [60] showed that a high density of support structures can result in lowering residual stress and warpage but adds to the complexity and costs.

The work of Snyder et al. [61], Mingear et al. [16], and Wildgoose et al. [29] showed internal channels deform relative to their design intent differently for different build directions. The build direction is defined as the angle from a part surface to the surface of the substrate as illustrated in Fig. 5. Surfaces that are perpendicular from the substrate are at a 90° build direction (vertical) while surfaces parallel with the substrate (horizontal) are at a 0° build direction. As seen in Fig. 6A, multiple computerized tomography (CT) scans along the length of a 0.51mm diameter channel built at a 45° angle shows the deviations from a design-intended round channel. These deformations happen at the top of the channel, which was the downward facing (unsupported) surface. To mitigate these deformations, a build...
correction in the digital model can be applied as has been reported by Snyder et al. [26] who used a teardrop correction as shown in Fig. 6B. Kamat et al. [62] also showed that a teardrop, diamond, and ellipsoid channel shape can reduce geometric deviations on the downfacing surfaces. Measurements by Wildgoose et al. [29] indicated that build direction starts to exhibit a noticeable impact to the surfaces and geometric deformations of circular channels once the build angle is below 60°, which will be further discussed later in this review.

Similar to small channels, other heat transfer augmentation features such as pin fins and film cooling holes, often used in cooling turbine blades, are affected by the build direction. Multiple pin fins with different cross-sections from Ferster et al. [63] are shown in Fig. 7 whereby the channel containing these features were fabricated at a 45° build angle. Kirsch et al. [30] observed that lowering the pin fin spacing resulted in an increase in surface roughness and geometric deviation from design intent, but also showed that this was dependent upon the location of the coupon placed on the substrate plate. The shaped film cooling holes featured in Fig. 8 emphasize the capabilities of the AM process to produce sub-millimeter holes. Shown in Fig. 8 is the diffuser portion of a shaped film cooling hole as was presented by the study of Stimpson et al. [64] to illustrate the magnitude of variation between an electro-discharge-machined (EDM) shaped hole that is conventionally done in Fig. 8B for an entrance diameter of 0.38 mm to that of an AM hole in Fig. 8A having the same 0.38 mm and in Fig. 8B for a hole twice the diameter of 0.76 mm. Their results implied a limit as to the feature size and also presented the complexities associated with the build direction on particular features.

Fig. 6 Cross-sections from CT scan data of a circular 0.508 mm channel (A) and a 0.508 mm teardrop (B) correction both at the 0° build direction [26].
Fig. 7  CT scan images of different shaped pin fins that were fabricated at the 45° build direction [63].
The importance of characterizing the resulting AM component through detailed CT scans was illustrated by Stimpson et al. [31] who showed hydraulic diameters for a 1mm rectangular channel can differ by as much as 10% for different builds and can differ by as much as 20% relative to the design intent across multiple build directions [29]. Wildgoose et al. [29] found that regardless of hydraulic diameter for circular channels, the hydraulic diameter increased just from the build direction increasing from $0^\circ$ to $90^\circ$ as seen in Fig. 9. They also found that not only does the hydraulic diameter of a circular channel change with build direction so does the variation of the diameter along the length of the channel. As seen by the histograms of the CT scans along the length of a 1mm circular channel in Fig. 10, there is a wide range of diameters that occur for a horizontal build ($0^\circ$) that tightens in distribution in the case of a vertical build ($90^\circ$).

Circularity describes how close a cross-section is to that of a perfect circle. Fig. 11A from Wildgoose et al. [29] shows how circular channels become increasing noncircular as build direction decreases below $60^\circ$. Similar to circularity, the variation of centroids along the length of the
Fig. 9 Deviation of the hydraulic diameter from design intent of circular channels across a range of build directions [29].

Fig. 10 Histogram of the measured diameter using CT scans along the length of a 1 mm circular channels for different build directions [29].

Fig. 11 Impact of build direction on the circularity (A) and concentricity (B) of 0.75, 1, and 1.25 mm circular channels [29].
channel (concentricity) becomes noticeable below 60° as seen in Fig. 11B. It has been observed through multiple studies that as build direction increases from 0° to 90°, concentricity decreases [29,61]. These two figures of merit in evaluating AM components quantify the fact that AM components can result in some waviness along the build and that build direction, which is sometimes not possible to control due to the complexity of the component, is an important parameter.

3. Heat transfer and friction factors for small channels

For meeting high heat flux demands, single phase microchannels are often used as a passive cooling concept in such applications as for gas turbines components, for electronics cooling, and even for solar concentrators. Over time, there has been a shift to microchannels but for the purposes of this review, we will consider channel dimensions on the order of millimeters in cross-section, which is in the realm of possibilities for printing using AM.

As previously discussed, the surfaces of AM parts are intrinsically rough. Ventola et al. [65] were the first to show the enhanced heat transfer provided by the roughness on additive surfaces. In their study, the external heat transfer coefficients of additively manufactured fins were found to be 40% higher than the same smooth geometry. While external surfaces can be smoothed through postprocessing, it has already been discussed that complex internal channels cannot be easily postprocessed. Very few studies [66] were found to have experimentally investigated internal friction factor and heat transfer including the fully turbulent flow regime of additively manufactured channels with a constant cross-section except for those from the same research group as the authors’. Of those few studies, Parbat et al. [67] presented Nusselt numbers for a rectangular AM channel which served as a comparison between the same geometry with wall jets implemented. Collins et al. [68] characterized the friction factor and heat transfer of a 500 μm diameter square channel and compared the results to standard correlations for laminar flow. This section of the review discusses the most basic effects of roughness, build direction and other process parameters on resulting heat transfer and pressure losses for microchannels followed by a useful correlation based upon the data currently available.

Since flow in rough pipes has been studied for decades, much exists in the literature regarding this topic. However, only a small number of studies examined flow in millimeter-sized channels, and even fewer looked at channels with high relative roughness. Huang et al. [69] studied flow through
pipes with relative roughness \( (K_s/D_h) \) values as high as 42%. They observed that as relative roughness exceeds about 7%, the friction factor in the laminar regime is higher than the theoretical Blasius values and that transition to turbulent flows occur at Reynolds numbers lower than expected based on smooth channels. Dai et al. [70] compiled a large sampling of friction factor data indicating increased friction factors when relative roughness levels exceed 2% for several different channel geometries.

Stimpson et al. [28] was the first to report friction and heat transfer measurements for millimeter-sized channels made using AM as shown in Table 2 with a wide variety of dimensions. The sample dimensions from Stimpson et al. [28] with the different channels is seen in Fig. 12 all made using Inconel 718 with the exception of the baseline “smooth” channel made from aluminum. In a related study, Snyder et al. [26] analyzed the effect that build direction had on flow structures in similar AM parts paying particular attention to the different roughness features generated by the different build orientations for the part. The authors of both studies reported relative roughness values (based on an arithmetic mean) ranging as high as 25 \( \mu m \) and proved that existing correlations for friction factor and heat transfer no longer held true for surfaces with such high roughness values. Stimpson et al. [28] evaluated a number of different scaling parameters using

### Table 2: Inconel 718 and smooth baseline coupon descriptions [28].

<table>
<thead>
<tr>
<th>Coupon name</th>
<th>( W_{design} ) [( \mu m )]</th>
<th>( H_{design} ) [( \mu m )]</th>
<th>( D_h, ) design [( \mu m )]</th>
<th>AR</th>
<th>( K_s/D_h )</th>
<th>( K_s/D_h )</th>
<th>Cross-sections</th>
</tr>
</thead>
<tbody>
<tr>
<td>L-1 \times \text{In}</td>
<td>457</td>
<td>1016</td>
<td>631</td>
<td>2.22</td>
<td>0.017</td>
<td>0.19</td>
<td></td>
</tr>
<tr>
<td>L-2 \times \text{In}</td>
<td>914</td>
<td>2032</td>
<td>1261</td>
<td>2.22</td>
<td>0.009</td>
<td>0.08</td>
<td></td>
</tr>
<tr>
<td>M-1 \times \text{In}</td>
<td>305</td>
<td>660</td>
<td>406</td>
<td>2.16</td>
<td>0.025</td>
<td>0.45</td>
<td></td>
</tr>
<tr>
<td>M-2 \times \text{In}</td>
<td>610</td>
<td>1321</td>
<td>834</td>
<td>2.17</td>
<td>0.015</td>
<td>0.22</td>
<td></td>
</tr>
<tr>
<td>S-2 \times \text{In}</td>
<td>610</td>
<td>610</td>
<td>610</td>
<td>1.00</td>
<td>0.020</td>
<td>0.23</td>
<td></td>
</tr>
<tr>
<td>Cyl-Al, smooth</td>
<td>–</td>
<td>–</td>
<td>635</td>
<td>–</td>
<td>–</td>
<td>0</td>
<td>Round</td>
</tr>
</tbody>
</table>
the coupons geometries shown in Table 2 to determine the best value to correlate the data and concluded that the arithmetic mean value \( (R_a) \) was sufficient. They based this conclusion from numerous AM coupons that were CT-scanned and measured from an optical profilometer in which a range of already published variables were evaluated.

The data from the Stimpson et al. [31] study are shown in Figs. 13 and 14 for friction factor and for Nusselt number of those geometries presented in Table 2 as well as the smooth, round coupon, which agrees with accepted
correlations. Fig. 13 shows that for the smooth channel the friction factors agree with well-accepted correlations but start to diverge from the laminar theory for $Re < 700$. The increased friction in this range is the result of entrance effects becoming significant given the coupons were between $20 < L/D_h < 63$, which is in agreement with Langhaar [72]. Length-to-diameter ratios do not typically have such an impact on pressure loss in the turbulent flow region.

The pressure loss data in terms of friction factors for the AM coupons indicate that even at the low Reynolds numbers there is an impact of the roughness relative to what is expected from laminar flow theory. Deviation from laminar flow theory of flow through channels with high roughness was seen in the results of several others who have studied friction factor of mini-channels [67,69,70]. The data in Fig. 13 indicates the highest augmentation of friction factor occurred for the coupon having the highest ratio of arithmetic mean roughness-to-hydraulic diameter ($R_a/D_h$). The data shown in Fig. 13 also indicates the transitional Reynolds number occurs between $700 < Re < 2000$ as indicated by the increases in friction factors. In the turbulent region shown in Fig. 13, the data can be categorized
into four groups that correlate with the highest augmentations occurring for
the highest $R_a/D_h$ ratios. For each of the coupons shown in Fig. 13, a relative
sandgrain roughness ($K_s/D_h$) was computed based upon the Colebrook
correlation [73], Eq. 2, and also given in Table 2:

$$\frac{1}{\sqrt{f_o}} = -2.0 \log_{10} \left( \frac{K_s/D_h}{3.7} + \frac{2.51}{Re \sqrt{f_o}} \right)$$

The table shows the resulting $K_s/D_h$ for each of the coupons which are sig-
nificantly higher than the $R_a/D_h$ values but that the overall order of higher
and lower values agree between the two ratios.

Convective heat transfer data is presented in the form of a Nusselt num-
ber in Fig. 14 whereby the coupons were tested using an imposed constant
surface temperature. Along with this data, the Dittus-Boelter correlation
given by Eq. (3) and Gnielinski correlation given by Eq. (4) are shown where $f$
in the Gnielinski correlation is taken to be the smooth friction factor found
using the Colebrook equation listed above [74,75]. The Dittus-Boelter
equation is only valid for $Re \geq 10,000$, and the Gnielinski correlation is valid
for $3000 \leq Re \leq 5e06$. Within their respective Reynolds number ranges, the
“smooth” data is within ±12% of both correlations. The data in Fig. 14 at
low $Re$ is transitioning to a constant value of 3.66 as would be expected for
data in the laminar regime.

$$Nu_o = 0.023 Re^{0.8} Pr^{0.4}$$

$$Nu_o = \frac{f_o/8(Re - 1000)Pr}{1 + 12.7 \sqrt{f_o/8(Pr^{2/3} - 1)}}$$

Results shown in Fig. 14 indicate that AM channels with the lowest to highest
$Nu$ correspond to the order of the friction factor data presented in Fig. 13;
however, the augmentations do not increase as significantly for Nusselt num-
ber relative to friction factors. This is especially observed for the coupons
having the highest friction factor values. Norris [71], as reported by Kays
and Crawford [76], experimented with pressure loss and heat transfer of flow
through channels with varying levels of roughness resulting in a simplified cor-
relation for Nusselt number augmentation expressed as a function of friction
factor augmentation. He found that the augmentation of Nusselt number
increases with increased friction factor according to the functional relationship
given in Eq. (5), where $n = 0.68Pr^{0.215}$.

$$\frac{Nu}{Nu_0} = \left( \frac{f}{f_0} \right)^n$$
Lines of \( \frac{Nu}{Nu_0} \) for varying \( \frac{f}{f_0} \) using the above correlation are given in Fig. 14 for the purpose of comparison. As a limitation to this correlation, Norris reported that augmentation of \( Nu \) ceases to increase for friction factor augmentation greater than \( \frac{f}{f_0} = 4 \).

Fig. 15 is a comparison of the friction factor vs Nusselt number augmentations where the augmentation is relative to a smooth channel correlation

<table>
<thead>
<tr>
<th>AM Datasets</th>
<th>Non AM Datasets</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel Shape</td>
<td>D_{h, actual} [µm]</td>
</tr>
<tr>
<td>Rectangle</td>
<td>626</td>
</tr>
<tr>
<td>Rectangle</td>
<td>1246</td>
</tr>
<tr>
<td>Rectangle</td>
<td>415</td>
</tr>
<tr>
<td>Rectangle</td>
<td>739</td>
</tr>
<tr>
<td>Square</td>
<td>614</td>
</tr>
<tr>
<td>Rectangle</td>
<td>632</td>
</tr>
<tr>
<td>Rectangle</td>
<td>1275</td>
</tr>
<tr>
<td>Rectangle</td>
<td>469</td>
</tr>
<tr>
<td>Rectangle</td>
<td>920</td>
</tr>
<tr>
<td>Square</td>
<td>669</td>
</tr>
<tr>
<td>Circle</td>
<td>554</td>
</tr>
<tr>
<td>Circle</td>
<td>569</td>
</tr>
<tr>
<td>Circle</td>
<td>528</td>
</tr>
<tr>
<td>Tear drop</td>
<td>623</td>
</tr>
<tr>
<td>Diamond</td>
<td>599</td>
</tr>
</tbody>
</table>

Fig. 15 Comparison of additively produced channels [26,28] with constant cross-sections compared to non AM highly engineered samples from Han et al. [77], Han et al. [78], and Wright et al. [79].
(f₀ and Nu₀) for numerous AM channels over a range of Reynolds numbers. According to the limitation in Norris’s correlation, heat transfer augmentation ceases to dramatically increase past 2.4 when Pr = 0.7. As seen by Fig. 15 a large portion of the AM channels follow this trend; however, there are several outliers as seen by the case of the cylindrical 45° [26] and M-1 × -Co [28] samples. As a comparison, also shown in Fig. 15 are augmentations for highly-engineered ribbed surfaces. The first thing to note is that the augmentation data for simple AM channels overlap with the highly-engineered surfaces that were designed to promote convective heat transfer relative to the increase in pressure losses [77–79]. Also, important to note is that the friction factor augmentations from all the surfaces are significantly higher than the heat transfer augmentations. While some of the highly engineered surfaces in Fig. 15 show increases in augmentation beyond f/f₀ = 4, the majority of the data for the simple AM coupons is in agreement with the trend indicated by Norris in which heat transfer augmentations cease to exist beyond f/f₀ = 4, which was based on an open channel.

3.1 Build orientation effects

The build orientation for components has been reported by many to have a significant impact on the surface finish. Strano et al. [80] studied the roughness for steel AM components as a function of orientation between horizontal and vertical. They found that the roughness was fairly constant at a value of about 15μm for angles between horizontal and 45° beyond which roughness decreased to about 13μm for a fully vertically built channel. Delgado et al. [81] measured external surface roughness of parts manufactured with DMLS from an iron-based powdered metal using various machine parameters and two build orientations. They found that roughness is affected by layer thickness, scan speed, and build direction with the latter having the greatest contribution to the roughness. Delgado et al. reported arithmetic mean roughness values for horizontally built surfaces between 5 < Rₐ < 8μm while values were two to three times greater on vertically built surfaces. Simonelli et al. [82] performed a similar study and observed comparable trends. Ventola et al. [65] found external surface roughness increased resulting from the stair stepping effect inherent with a layer-by-layer AM process. Common roughness values found in literature range between about 5 < Rₐ < 50μm and are strong functions of machine parameters and build direction [45, 65, 80–84].

A significant portion of studies that have stated roughness measurements are from external surfaces while only a few studies have reported internal
surface roughness measurements. Wildgoose et al. [29] varied channel diameters and build directions between horizontal (0°) and vertical builds (90°) to determine the effects on the resulting friction factor and heat transfer. Coupons that were multichannel containing millimeter-sized channels, similar to that shown in Fig. 12 were used for the study. Each of the coupons were CT scanned with some of the data illustrated in Fig. 16 showing the surface outline of the channels taken at three different streamwise positions for the 1 mm diameter channels. It is clear from the CT scan data in Fig. 16 that as the build direction decreases from vertical (90°) to horizontal (0°), the channels become more oval-shaped with the horizontal build exhibiting a deformed top half of the channel, which is the downward facing surface. Also shown in Fig. 16 is that the centroid of the channel shifts slightly at the lower build angles such that the center is misaligned at various streamwise locations along the channel.

Fig. 17 shows the variation in the channel surface roughness that decreases from 0° to 90° for the three different channel diameters that were
used in the Wildgoose et al. [29] study. These roughness levels are similar to values found in literature. It is not surprising that nearly the same roughness values occurred for all three channel diameters for a particular build direction since the same AM process parameters were used. However, the implications of these results reinforce the importance of build angle on the roughness levels. Surface roughness between 60° and 90° is indistinguishable and begins to increase from 60° to 0°.

The sensitivity of these roughness levels resulting from the build direction are shown in Fig. 18A and B at $Re = 20,000$ [29]. While the friction factor decreases as build angle increases, the heat transfer augmentation peaks between 30° and 45°. Snyder et al. [26] also observed the highest heat transfer augmentation at 45° compared to 0° and 90°. As would be anticipated from the roughness values shown in Fig. 17, which decrease with build angle, it would be expected that augmentations decrease if it were simply a function of roughness. These results point to the importance of the surface morphology that results from a particular build angle and the nature of the roughness. Consider the nature of the roughness, for example, as illustrated in Fig. 19 showing the surface with three different metal particles (A, B, and C). Each of these particles have different effective diameters and each are sintered differently to the surface. For C, the particle is well sintered to the surface resulting in a high potential for conduction and simulating a good fin while for B, the particle is barely touching the surface indicating a poor fin but still contributing to fluid drag that results in pressure loss.

Despite the 60° coupons having similar surface roughness, friction factor augmentations, and geometric tolerances as compared to the 90° coupons in Fig. 18, the 60° coupons observed an 8% higher heat transfer augmentation compared to the 90° coupons. Based on these results, orienting an additive

![Fig. 18 Friction factor (A) and Nusselt number (B) augmentation of circular channels with different diameters across multiple build directions [29].](image)
component with an optimal build angle can result in sizable increases in heat transfer without additional pressure losses. Implementing a channel shape correction to the circular channel at the $30^\circ$ and below may reduce deformation on the downward facing surface leading to a reduction in friction factor.

### 3.2 Print process parameters effects

As has been discussed, random roughness levels resulting from the AM process can augment heat transfer equal to highly engineered surfaces for a given pressure drop as illustrated in Fig. 15. While numerous process parameters exist for AM, it is possible to tailor those parameters to achieve different surfaces as was done in the exploratory work by Snyder et al. [53]. Fig. 20 presents some of

![Fig. 19 Scanning electron microscope micrographs showing the upward facing surfaces at low (A) and high magnification (B) highlighting the range of partially sintered metal particles [31].](image1)

![Fig. 20 Scanning electron microscope micrographs of surfaces with differing AM process parameters to form smooth (A), rough (B), and ribbed (C) surfaces [53].](image2)
those results where they attempted to alter the surface roughness. Other than the work by Snyder et al., the only other study that took the approach of changing process parameters to affect the part performance was done by Fantozzi et al. [85] who varied the laser power, scanning speed, and hatch spacing to control the pressure losses through a porous material. However, flow through porous media is significantly different than high Reynolds number turbulent channel flows.

The effect of different process parameters on surface roughness has been investigated by many who have found that the predominant parameter affecting surface roughness is the amount of energy input by the laser [55, 86–89]. All of these studies saw the same trend where roughness decreased with increasing energy density to a minimum roughness level before increasing with further increases energy density. Also previously discussed is that one of the key parameters to consider when controlling surface roughness is the use of a contour scan, or single laser pass around the edge of each layer which then becomes the surface of the component. The intent of a contour scan is to remelt larger roughness features into smaller ones. The general consensus among studies in the literature is that contours are successful in reducing the surface roughness when used correctly [89–92]; however, the exact physical phenomena are not well understood with some studies showing contradictory results.

To demonstrate whether tailoring the process parameters to develop different types of roughness, Snyder et al. [53] used millimeter-sized channels, similar to those already discussed, as test coupons. For these coupons one of the channel surfaces was defined as a test surface where the AM process parameters were controlled to alter the type of roughness. Only one surface was chosen for the 45° built channels since this was feasible to implement. Roughness variations were brought about by changing the laser power, laser scanning speed, hatch distance, and laser scanning. Resulting from these changes in the process parameters, three different roughness types were printed that are referred to as “smooth,” “rough,” and “ribbed” as illustrated by the scanning electron microscope (SEM) images in Fig. 20.

For the smooth case in Fig. 20A, the hatching parameters were chosen to create a small, stable melt pool during the hatching by using a low scanning speed and laser power, and by using a small hatch spacing. To further reduce the roughness, two consecutive contour passes were utilized to reduce the roughness even further from the hatching phase. The morphology is characterized by a relatively smooth base surface, with small partially sintered metal particles. The rough test surface, shown in Fig. 20B, was built by increasing the laser power, speed, and hatch spacing relative to the smooth
case resulting in a lower surface energy flux than the smooth case. The high laser power caused increased material vaporization leading to irregular roughness features. The “ribbed” morphology in Fig. 20C was printed by utilizing a higher scanning speed and lower laser power than the smooth case resulting in an elongated melt pool caused by Plateau-Rayleigh instabilities [54]. With a high length-to-width ratio, the surface tension of the melt pool attempted to break the melt pool into spheres, minimizing its surface energy. The goal in creating the ribbed surface was to create roughness akin to traditional rib turbulators found in many internal turbine passages, but without changing the roughness magnitude from the smooth case. The flow direction was intended to be perpendicular to the ribs.

The friction factor augmentation relative to a smooth channel is shown in Fig. 21A for the smooth, rough, and ribbed surfaces that were illustrated in Fig. 20 while the heat transfer augmentation is shown in Fig. 21B. Both the smooth and ribbed channels have nearly the same augmentation performance while the fully rough has the highest augmentation of friction factor and lowest augmentation of heat transfer. The relatively poor heat transfer performance of the rough test surface was hypothesized to be the result of the geometric changes to the sidewalls separating the channels. While the parameters for the rough test surface increased the surface roughness of the sidewalls, these parameters also decreased the separating wall thickness reducing the sidewalls’ ability to conduct heat away from the center of the coupon. Snyder et al. [53] quantified this effect through a fin efficiency for the rough test surface, which indicated a drop in the fin efficiency due to
the reduction in the fin (sidewall) thickness by between 5% and 10%. The reduction in fin efficiency was dictated by the low thermal conductivity properties of the nickel-based alloy (Inconel 718) used for this study.

3.3 Correlations to predict performance for small additively manufactured channels

In 2016, Stimpson et al. [31] provided a correlation for predicting the pressure loss and convective heat transfer for additively manufactured channels. In previous work, they noted that using accepted correlations for smooth channels could result in miss-predictions by as much as 300%. Through the work they presented in 2016, they showed a detailed analysis of 10 additively manufactured coupons made from 2 different materials all being rectangular in shape at the 45° build direction with hydraulic diameters ranging from 400 to 1260 μm having \( R_a/D_h \) values as high as 0.025. In developing this correlation, they evaluated the use of five different parameters for scaling heat transfer and pressure losses that included: arithmetic mean roughness (Ra), root mean square of the roughness, mean roughness depth, skewness, and kurtosis. They also evaluated using the autocorrelation function of the roughness levels to determine if there was a dominant length scale, which was not identified, and evaluated the use of a shape factor first introduced by Dirling [93] to correlate drag and heat transfer. The results presented by Stimpson et al. [31] indicated that the best scaling factor to use was the simplest of the variables evaluated namely the arithmetic mean roughness, \( Ra \). They proposed a correlation for calculating a relative sand grain roughness, \( K_s/D_h \), as a function of a measured \( R_a/D_h \), which they determined using Colebrook’s equation, Eq. (6), and their measured friction factors.

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

(6)

If the channel pressure loss is unknown, it is possible to use the \( K_s/D_h \) correlation to calculate the friction factor once the \( R_a/D_h \) value is known from measured values. Once the friction factor is calculated, Stimpson et al. [31] used this friction factor in their newly proposed Nusselt number correlation given by Eq. (7).

\[
Nu = \frac{(Re^{0.5} - 29)Pr\sqrt{f/8}}{0.6(1 - Pr^{2/3})}
\]  

(7)
In using their $K_s/D_h$ correlation to predict friction factor and then heat transfer, they noted as much as a 30% difference; however, in using their direct measure of pressure loss in terms of a friction factor in the Nusselt number correlation Eq. 7, they noted agreement to within better than 15%. These results implied that their $K_s/D_h$ correlation needed further development. Furthermore, the correlation was developed only using multichannel, millimeter-sized square and rectangular coupons that were built at 45°.

Since the 2016 study presented by Stimpson et al. [28], numerous pressure loss and heat transfer tests have been conducted with many more coupons that have included multiple materials, multiple build directions, and multiple geometries (rectangular, circular, teardrop, and diamond). The authors continued the development of their correlation using all available datasets from literature to predict friction factor and Nusselt number performance from AM channels. Similar to previous work, these additional sets of coupons imposed a constant surface temperature and used CT scanning to measure the $R_a$ values. A table indicating the ranges of test coupons and the bounds over which tests were conducted is shown in Table 3.

The studies used to make the correlation include: the rectangular 45° channels with varying diameters [28], circular channels at a single diameter with different build directions [26], varying diameters of circular channels fabricated at four different build directions [29], and rectangular channels built at 45° which contained different surface roughness dictated by changes to laser process parameters [53]. As seen in Table 3, the dataset uses multiple channel shapes (circular, square, and rectangular cross-sections), multiple materials (Inconel 718, Hastelloy X, and Cobalt-Chromium), a variety of hydraulic diameters (500–1400 μm), a range of build directions from 0° to 90°, and different process parameters. Furthermore, only incompressible fully turbulent friction factor values were used for the relative sandgrain roughness, $K_s/D_h$.

In performing several best fit scenarios, the authors have found that an approach similar to that presented by Stimpson et al. [31] best fit the data with the only exception being a more robust correlation for $K_s/D_h$ was needed. Similar to Stimpson et al. [31], $K_s/D_h$ was derived from using the measured friction factors along with the Colebrook relation given by Eq. (6). Fig. 22 shows the results of this analyses and includes the originally proposed correlation given by Stimpson et al. [31], which was based on significantly fewer coupons. While there is some scatter in the data, a proposed best fit was determined to be given by Eq. (8):

$$\frac{K_s}{D_h} = 11 \frac{R_a}{D_h}$$

<table>
<thead>
<tr>
<th>Coupon number</th>
<th>Channel shape</th>
<th>Material</th>
<th>Build direction [°]</th>
<th>$D_h$ [μm]</th>
<th>$R_{pd}/D_h$</th>
<th>$K_a/D_h$</th>
<th>References</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coupon 1</td>
<td>Circle</td>
<td>Inconel 718</td>
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<td>1147</td>
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<td>Inconel 718</td>
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<td>CoCr</td>
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<td>1246</td>
<td>0.010</td>
<td>0.120</td>
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</tr>
<tr>
<td>Coupon 14</td>
<td>Rectangle</td>
<td>CoCr</td>
<td>45</td>
<td>739.0</td>
<td>0.017</td>
<td>0.237</td>
<td>[28]</td>
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<tr>
<td>Coupon 15</td>
<td>Square</td>
<td>CoCr</td>
<td>45</td>
<td>614.0</td>
<td>0.015</td>
<td>0.218</td>
<td>[28]</td>
</tr>
<tr>
<td>Coupon 16</td>
<td>Rectangle</td>
<td>Inconel 718</td>
<td>45</td>
<td>632.0</td>
<td>0.017</td>
<td>0.190</td>
<td>[28]</td>
</tr>
<tr>
<td>Coupon 17</td>
<td>Rectangle</td>
<td>Inconel 718</td>
<td>45</td>
<td>1275</td>
<td>0.009</td>
<td>0.077</td>
<td>[28]</td>
</tr>
<tr>
<td>Coupon 18</td>
<td>Rectangle</td>
<td>Inconel 718</td>
<td>45</td>
<td>920.0</td>
<td>0.015</td>
<td>0.217</td>
<td>[28]</td>
</tr>
<tr>
<td>Coupon 19</td>
<td>Square</td>
<td>Inconel 718</td>
<td>45</td>
<td>664.0</td>
<td>0.020</td>
<td>0.225</td>
<td>[28]</td>
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<td>Coupon 20</td>
<td>Circle</td>
<td>Inconel 718</td>
<td>90</td>
<td>554.0</td>
<td>0.015</td>
<td>0.150</td>
<td>[26]</td>
</tr>
<tr>
<td>Coupon 21</td>
<td>Circle</td>
<td>Inconel 718</td>
<td>0</td>
<td>569.0</td>
<td>0.028</td>
<td>0.310</td>
<td>[26]</td>
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<tr>
<td>Coupon 22</td>
<td>Circle</td>
<td>Inconel 718</td>
<td>45</td>
<td>528.0</td>
<td>0.031</td>
<td>0.380</td>
<td>[26]</td>
</tr>
<tr>
<td>Coupon 23</td>
<td>Rectangle</td>
<td>Hastelloy-X</td>
<td>45</td>
<td>1260</td>
<td>0.009</td>
<td>0.141</td>
<td>[53]</td>
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<tr>
<td>Coupon 24</td>
<td>Rectangle</td>
<td>Hastelloy-X</td>
<td>45</td>
<td>1380</td>
<td>0.010</td>
<td>0.192</td>
<td>[53]</td>
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<tr>
<td>Coupon 25</td>
<td>Rectangle</td>
<td>Hastelloy-X</td>
<td>45</td>
<td>1270</td>
<td>0.010</td>
<td>0.150</td>
<td>[53]</td>
</tr>
<tr>
<td>Coupon 26</td>
<td>Rectangle</td>
<td>Hastelloy-X</td>
<td>45</td>
<td>1230</td>
<td>0.014</td>
<td>0.159</td>
<td>[53]</td>
</tr>
<tr>
<td>Coupon 27</td>
<td>Rectangle</td>
<td>Hastelloy-X</td>
<td>45</td>
<td>1300</td>
<td>0.017</td>
<td>0.224</td>
<td>[53]</td>
</tr>
<tr>
<td>Coupon 28</td>
<td>Rectangle</td>
<td>Hastelloy-X</td>
<td>45</td>
<td>1220</td>
<td>0.015</td>
<td>0.158</td>
<td>[53]</td>
</tr>
</tbody>
</table>
The proposed best fit correlation for $K_s/D_h$ in Fig. 22 is able to better predict relative sandgrain roughness values compared to Stimpson et al. [31], which tends to overpredict the values. The correlation proposed used an expanded data set in predicting relative sandgrain roughness as shown in Table 3. As seen in Fig. 23, the maximum absolute error of the relative sandgrain roughness using the proposed correlation is 71% however the average absolute
error of the datasets using the purposed correlation is much less at 20%. To evaluate the prediction of friction factor, the relative sandgrain roughness was calculated using Eq. (8) in the Colebrook relation in Eq. (6). The friction factor prediction of several samples using the proposed correlation and Stimpson’s correlation is seen in Fig. 24. The proposed correlation is able to predict friction factor values with an average absolute error of 12% while Stimpson’s correlation is able to predict values with an average absolute error of 23%.

For the heat transfer prediction, the proposed correlation by Stimpson et al. [31] given by Eq. (7), was used to predict Nusselt number for the datasets in Table 3. Two methods were used to evaluate the prediction of the Nusselt correlation from Stimpson et al. [31]. The first method assumes that the friction factor is known, while the second method assumes friction factor is unknown thereby needing the $K_s/D_h$ value for the Colebrook relation. As such the second method uses a measured relative arithmetic mean roughness, $R_a/D_h$, that is used to predict relative sandgrain roughness from the proposed $K_s/D_h$ correlation, Eq. (8). A comparison of the two methods for the Nusselt prediction is given in Fig. 25. The average absolute error is
13% using both methods, which is relatively good considering the range of AM build directions, materials, diameters, and process parameters. Fig. 26 illustrates a comparison of predicted and measured Nusselt numbers for specific channel cases using both methods for that of Stimpson et al. [31] as well as the revised $K_s/D_h$ methods.

As seen in Fig. 26, several datasets are shown from Table 3 with Stimpson et al. [31] Nusselt correlation using a known friction factor, the same Nusselt correlation calculated using the Stimpson et al. [31] $K_s/D_h$ correlation, and the proposed best fit $K_s/D_h$ correlation, Eq. 8. The maximum absolute error of Stimpson et al. [31] Nusselt correlation using the datasets in Table 3 is 40% when using a known friction factor and it is 40% when using the proposed $K_s/D_h$ correlation, Eq. 8. When using Stimpson et al. [31] $K_s/D_h$ correlation the maximum absolute error is 48%.

Fig. 25 Accuracy of Nusselt prediction, Eq. 7, using friction factor predicted from the relative sandgain roughness correlation, Eq. 8, and compared with measured friction factor from the studies in Table 3.
In summary, the proposed $K_s/D_h$ correlation is able to predict relative sandgrain roughness within an absolute average error of 20% with a maximum error of 71% and predict friction factor using the Colebrook equation within an average absolute error of 12% and a maximum error of 47% from the dataset in Table 3. Using the Nusselt correlation from Stimpson et al. [31] with the predicted friction factor from the proposed $K_s/D_h$ correlation, Eq. 8, the average absolute error is 13% while the maximum error is 40%. The $K_s/D_h$ correlation, Eq. 8, is limited to straight channels with hydraulic diameters greater than 0.5 mm.

4. Noncircular channel shapes

Noncircular channels have been shown to create secondary flows that influence the resulting pressure losses and convective heat transfer [94–98]. Because there have been limited studies of noncircular channels resulting from difficulties of conventionally manufacturing these shapes, it is important we take a fundamental look at these types of channels. Also of interest is whether the hydraulic diameter is truly the most relevant scaling parameter for noncircular channel friction factor and heat transfer.
4.1 Scaling friction factor and heat transfer for noncircular channel shapes

Hydraulic diameter is often used as a characteristic length for noncircular channels. However, using hydraulic diameter has been put into question as a consistent scaling parameter given the results reported for triangular channels by Duan et al. [94] who showed a 30% lower friction factor compared to a circular channel at the same Reynolds number based using the hydraulic diameter as a basis for comparison. Given the sparse amount of data available Jones [99] examined data taken in rectangular channels and found that an equivalent or hydraulic diameter, \( D_h \), was not sufficient and that effects of aspect ratio were important. He proposed using a laminar equivalent diameter, \( D_L \), which uses an empirically derived equation based on the aspect ratio of the channel to correct \( D_h \). Using \( D_L \) to calculate the friction factors collapsed the results in the laminar regime onto the theoretical \( 64/Re \) line. Jones [99] also found that evaluating turbulent data with the corrected diameter gives a better agreement with known correlations for turbulent pipe flow. Duan et al. [94], however, showed that the square root of cross-sectional area may be more applicable to collapse friction factor data of various smooth channel shapes compared to hydraulic diameter.

Compared to friction factor, there are far less convective heat transfer measurements for different channel shapes. Similar to friction factor, aspect ratio and channel shape have been shown to impact heat transfer [97,100,101]. Leung et al. [101] found that a 60° isosceles triangle contained the highest convective heat transfer relative to a range of apex angles while Kakac et al. [100] indicated that heat transfer coefficients increase with increasing aspect ratios for ellipses. Nikuradse [97] experimentally measured heat transfer coefficients for a range of trapezoids. Duan [102] proposed a general correlation to predict heat transfer results in noncircular channel shapes; however, this correlation was derived from hydrodynamic experiments by using the Colburn analogy instead of from direct experimental heat transfer measurements due to the lack of available noncircular turbulent heat transfer data in literature. Wang et al. [103] computationally showed that the use of hydraulic diameter for regular polygonal shaped channels, such as an equilateral triangle can be off by as much as 28% in Nusselt number based on correlations for circular channels.

To evaluate the influence channel shape has on the pressure loss and convective heat transfer, a variety of straight channel shapes were fabricated at a vertical build angle of 90° using AM on Inconel 718, as shown in Fig. 27.
Nine channel shapes were selected that are challenging and costly to fabricate using traditional manufacturing techniques. The nine channel shapes shared the same perimeter and coupon length to match the convective surface area. These channel shapes were placed in a coupon similar to the one previously shown in Fig. 12 only twice as long to achieve length-to-hydraulic diameter ratios to reduce entrance effects ($33 < L/D_h < 54$). The coupons were all designed to achieve fin efficiencies of greater than 95% such that the constant surface temperature boundary condition could be applied.

After building the coupons, each was CT-scanned to determine the quality. Figs. 27 and 28 provide results from the scans showing the outline of the surfaces for each of the coupons at various streamwise locations along the channel (Fig. 27) as well as the arithmetic mean roughness for individual surfaces for each channel shape (Fig. 28). What is immediately noticeable from the surface outlines in Fig. 27 is that horizontal surfaces (nonangular) located at 12 o’clock and 6 o’clock show significant surface deviations as compared to the other surfaces. These deviations are particularly evident for the square (Fig. 27F), trapezoidal (Fig. 27H), rectangular (Fig. 27G) and triangular channels (Fig. 27I). In contrast, the circular (Fig. 27A), hexagonal (Fig. 27B), and diamond (Fig. 27E) channels do not show such dramatic deformations. In particular, consider the pentagon in which the 6 o’clock horizontal surface is quite deformed but yet the angled surfaces on both sides of the 12 o’clock position are quite flat. And, consider the square (Fig. 27F) compared with the diamond (Fig. 27E), which have the same shape and design intent geometry but were built in different

Fig. 27 Axial slices of channel shapes (circle (A), hexagon (B), pentagon (C), ellipse (D), diamond (E), square (F), rectangle (G), trapezoid (H), and triangle (I)) sharing a constant design intent perimeter fabricated at the 90° build direction.
Fig. 28 Arithmetic mean roughness, \( R_m \), from CT scan data of multiple surfaces among the different channel shapes which are at the same surface orientation as Fig. 27.
orientations resulting in significantly different surface morphologies and roughness levels. The data in Fig. 28 quantifies these effects through an arithmetic mean roughness value for each of the surfaces showing dramatic increases for the horizontal surfaces at 12 o’clock and 6 o’clock for the horizontal surfaces.

Significant ongoing research is taking place to understand why these differences occur in the various surfaces of the noncircular channel geometries. At this time, it is hypothesized to be affected by a combination of the recoater blade, build location, as well as thickness of the external coupon wall. More research is needed to discern these AM effects which are important to keep in mind in evaluating the friction factor and heat transfer data presented in this paper for the various coupon geometries.

For each of the coupons in Fig. 27, pressure loss measurements were made over the laminar, transitional, and turbulent flow regimes whereby the data are presented in two ways: Fig. 29A uses the hydraulic diameter

![Fig. 29](image-url)  
**Fig. 29** Friction factor data over a range of Reynolds numbers calculated using hydraulic diameter (A) and square root of cross-sectional area (B) with benchmark data from Stimpson et al. [28] and smooth noncircular data compiled by Duan et al. [94].
as the scaling factor for Reynolds number and Fig. 29B uses the square root of cross-sectional coupon area. In both cases the AM coupon data is compared with that of smooth noncircular channel data compiled by Duan et al. [94]. Also presented is a smooth circular channel coupon data that was developed as a benchmark by the authors of this paper, which agrees with the well-accepted Colebrook friction factor correlation.

Consistent with results for AM internal passages, Fig. 29 shows the friction factors for the AM coupons were significantly higher than the data given for the smooth channels presented by Duan et al. [94] due to surface roughness. The friction factors in Fig. 29 also show transition to turbulence for the AM coupons occurs at lower Reynolds numbers than for the smooth channels. In considering the AM channels alone in Fig. 29, the transition Reynolds number for the square channels occurs at the lowest Reynolds number given it has more than three times the magnitude of arithmetic mean roughness compared to the diamond, for example. In general, channel shapes that contain high surface roughness and geometric tolerances enter the transitional regime at lower Reynolds numbers.

In evaluating the two scaling factors, the smooth channel data shown in Fig. 29A indicate some scatter in the data whereas using the cross-sectional area as a scaling parameter in Fig. 29B indicate that there is a reduction in the scatter as reported by Duan et al. [94]. The fRe for the AM channels using hydraulic diameter vary by as much as 22% compared to the circle at a Re \(D_h = 30,000\) whereas when using the square root of cross-sectional area, the variation of in \(fRe \sqrt{A}\) of square to circle is 19% at a \(Re \sqrt{A} = 30,000\) as shown in Fig. 28. The area scaling parameter proposed by Duan et al. [94] appears to be a better characteristic length in terms of collapsing smooth noncircular channels as shown in Fig. 29B. However, for the AM noncircular channels the square, trapezoid, rectangle, and triangle exhibit a higher \(fRe \sqrt{A}\) compared to the circle, hexagon, pentagon, ellipse, and diamond. These results are not only related to the roughness but also to the channel shapes as illustrated when comparing the ellipse and the circle which both exhibit the same friction factor although the ellipse has a lower roughness-to-hydraulic diameter ratio. Additionally, the rectangle and triangle have the same friction factor even though the rectangle has a lower relative roughness level.

Due to the different channel shapes having different relative arithmetic mean roughnesses but similar friction factors the proposed \(K_s/D_h\) correlation, Eq. 8, struggles to predict the friction factor for different channel
Fig. 30 shows the accuracy of friction factor of the newly proposed \( K_s/D_h \) correlation, Eq. 8, for several different channel shapes. As seen in Fig. 30, the newly proposed \( K_s/D_h \) correlation, Eq. 8, has a maximum absolute error of 73% and an average absolute error of 30% relative to a maximum absolute error of 109% when using Stimpson et al. [31] \( K_s/D_h \) correlation.

Convective heat transfer coefficients were measured by imposing a constant surface temperature boundary condition and are presented in terms of a Nusselt number using the two aforementioned scaling parameters as shown in Fig. 31. As seen in Fig. 31A, most of the channel shapes uncertainty bars overlap one another signifying Nusselt numbers among the different shapes are nearly indistinguishable. As seen in Fig. 31A, there is a 10% difference between the highest and lowest Nusselt number at \( Re = 50,000 \). Fig. 31B compares the scaling of the data using the square root of cross-sectional area proposed by Duan et al. [94]. The Nusselt numbers using the area shows a somewhat better collapse than using the hydraulic diameter for the various shapes. This result is evident in a comparison between the circle and diamond channel shape. The circle channel Nusselt number, calculated using the square root of cross-sectional area, is 2.4% different than the diamond shape.
Nusselt numbers and friction factors of the various channel shapes are shown in Fig. 33 relative to what occurs for a circular channel built using AM. For the comparison to circular channels, the diamond channels showed the lowest friction factors while the highest friction factors occurred for the square channels. At a limited Reynolds number range, the square, trapezoid, rectangular and triangular channel shapes gave higher heat transfer augmentations relative to the circular channel but also came with an increased friction factor augmentation. These noncircular channel results illustrate that more research should be done in evaluating different basic channel shapes.
Fig. 32 Comparison of Nusselt number prediction using $K_e/D_h$ correlation from Stimpson et al. [31] and the proposed correlation in Eq. (8) of several channel shapes from Fig. 27.

Fig. 33 Nusselt number and friction factor of channel shapes augmented by the Nusselt number and friction factor of the circular channel shape, Fig. 27A at the same Reynolds number.
but that it is critical to consider the manufacturing processes simultaneously, which have a significant impact on the channel shape, surface morphology, and surface roughness.

5. Other types of microchannels and heat transfer augmentation surfaces

Clearly a large area of interest in applications of additive manufacturing for heat transfer is the potential to generate complex internal channel shapes, or features that were previously difficult or impossible to fabricate such as triply periodic minimal surfaces. Just as for circular channels, the surface roughness and dimensional tolerance of AM, however, must be considered.

5.1 Wavy microchannels

Kirsch and Thole [104] investigated wavy microchannels (hydraulic diameter of 0.55–0.58 mm) fabricated via Inconel 718 in a DMLS process. Measurements of the parts indicated approximately 10% larger hydraulic diameter relative to design intent, and relative roughness levels \( (R_a/D_h) \) of 0.015–0.018. Short wavelength wavy microchannels had significant pressure drop but not much heat transfer benefit relative to longer wavelength channels. Fig. 34 shows a simulation of the streamwise velocity in

![Fig. 34 Simulation of wavy microchannels with different wavelengths (A), and measured Nusselt augmentation versus friction factor augmentation (B) compared to other engineered features [28,79,105–107].](image)
the varying wavy channels, where short wavelengths create many flow acceleration/deceleration regions around the bends. Fig. 34 also shows that at a fixed friction factor augmentation of five relative to a smooth channel, a moderately wavy microchannel with a wavelength of 20% of the channel overall length could result in nearly $3.5 \times$ improvement in heat transfer relative to a smooth channel, or $2 \times$ improvement relative to a straight additively manufactured microchannel.

The advantage of AM, however, is the potential to develop more efficient designs that take advantage of the manufacturing flexibility. To this end, Kirsch and Thole performed several studies using an adjoint method to optimize the shape of the wavy microchannel cross section [108], or even optimization of wavy microchannels with communicating passages as shown in Fig. 35 [109]. For optimizations based on maximizing heat transfer,

![Figure 35](image)

**Fig. 35** Wavy microchannels with communicating passages (A), and heat transfer augmentation of adjoint-based optimized designs (B), where the optimization targeted minimum pressure drop ($J_1$), maximum heat transfer ($J_2$), or a combination of the two ($J_3$).
vortical structures common to wavy channels were enhanced, while those same structures were minimized in designs optimized for low pressure drop. Despite the small scale of the channels, several optimization features were reasonably replicated by DMLS, and maximal heat transfer designs did in fact show significant improvement. However, minimal pressure drop designs did not match optimizer results, likely due to the relatively large roughness features inherent to DMLS of microchannels, or probably also limitations of turbulence models in capturing channel secondary flows [48]. Designs based on optimization of heat transfer and pressure drop did result in a 15% improvement in heat transfer for a given friction factor relative to a nonoptimized wavy microchannel. For wavy microchannels with communicating passages, Fig. 35 shows that experimental testing of optimized designs showed 6% lower pressure drop for minimal pressure drop design criteria, or 9% increase in heat transfer for maximal heat transfer design criteria, relative to the nonoptimized baseline. Despite the lack of agreement between the optimizer predicted performance and actual performance, the success of the optimizer and DMLS processes in producing observable changes in performance is encouraging.

Several recent studies have investigated other novel extensions of wavy microchannel shapes [110–112], although more work remains to experimentally validate these in concert with additive manufacturing. Parbat et al. [67] used metal AM to fabricate microchannels with wall jets, where the jet is created by the flow restriction of a small gap between a blockage in the channel and the upper wall of the channel. Regular spacings of the blockage ensured continuous refresh of the wall jet in the space between the blockage and the upper wall. This design results in significant pressure loss, anywhere from 20 to nearly 250 times that of a smooth channel, but also heat transfer augmentation up to nine times that of a smooth channel. The authors also observed excellent agreement to the correlation of Stimpson et al. [31] for their empty DMLS microchannels.

The potential of additively manufactured wavy microchannels in gas turbine cooling applications was recognized by Wimmer et al. [113–115] and Ruhmer et al. [116], in a series of studies incorporating the wavy microchannel design of Kirsch and Thole [104] into a gas turbine ring segment (component of a gas turbine above a rotating turbine blade that contains the hot gases within the flowpath). This part is subjected to extreme temperatures and often has a limited life due to oxidation and cracking. Compared to a conventionally cast ring segment, the additively manufactured ring segment reduced maximum nondimensional metal temperature by 11% and reduced
necessary cooling air by 23%. The better overall control of metal temperature in the AM ring segment also enabled it to extend the number of high–low temperature cycles (commonly experienced as gas turbines start up and shut down, or vary their operation point) by >15% relative to the cast part. These are significant gains that validate the potential of AM for real impact in applications.

5.2 Pin fins, lattices, and triply periodic minimal surfaces

Additively manufactured pin fins were studied by Kirsch and Thole [30] and Ferster et al. [63] for the LPBF process, and by Dupuis et al. [117] for pins made by a cold spray additive manufacturing process (essentially fusion of metal particles via high velocity impact onto a surface). In all cases, the manufacturing processes result in rough surfaces relative to the extensive prior literature in pin fin arrays (e.g., [118–123]). Kirsch and Thole [30] observed an increase in surface roughness with highly packed pin fin arrays, which resulted in very high friction factors without a concomitant increase in heat transfer performance. In fact, they observed the same heat transfer for the least densely packed array but lower friction factor, suggesting that low density packing arrangements are appropriate for LPBF fins, which can also reduce overall weight. Relative to smooth pin fins, all additive pin fin arrays had higher friction factor, and slightly higher or marginal increase in heat transfer. Ferster et al. [63] also investigated LPBF manufactured pin fins, but of varying shapes (see Fig. 7). Fig. 36 shows the heat transfer augmentation vs friction factor augmentation for the various pin shapes. In particular, triangular pin fins with the flat face on the windward side resulted in the highest heat transfer observed in the body of literature on pin fins, with a Nusselt augmentation of up to eight relative to a smooth channel. Six–pointed star shapes were significantly less effective, with a similar friction factor augmentation (14 < f/f_0 < 18) but a Nusselt augmentation of less than 2. This highlights the need for continued study of optimal augmentation shapes, which may be possible to construct via AM but may be poor performing.

Learning from coupon-scale studies have started to make their way into more complex applications of metal AM. Krewinkel et al. [124] applied the findings from LPBF microchannel and pin fin studies to construct a nozzle guide vane for a gas turbine engine, which was operated for over 70h in a plant. Despite fabrication challenges with the thin–walled internal baffle system common to vanes, the authors successfully constructed a vane that had adequately resolved internal pin fins, and a cooling air flow test indicated no
deviation relative to vanes constructed via conventional casting processes. The maximum experienced metal temperature was measured using a thermal history paint and indicated overall good agreement between the conventional and AM vanes. Hossain et al. [125] also constructed a vane using metal additive manufacturing, which incorporated novel cooling designs such as triangular pin fins and sweeping jet film cooling holes [126]. The trailing edge cooling with the triangular pins was increased relative to a conventional cylindrical pin fin, resulting in lower metal temperatures in their heated facility.

Ligament-based lattice and triply periodic minimal surface (TPMS) structures are a class of geometrically complex geometries that can be challenging to fabricate, and thus can benefit from AM. They can have high surface area to volume ratios, which is generally advantageous for heat transfer. Kaur and Singh [127] provide an extensive review of additive manufacturing of lattice structures, concluding that lattices can have better thermal characteristics than stochastic metal foams but suffer from minimum

![Fig. 36 Heat transfer vs friction factor augmentation for various uniquely shaped pins made by additive manufacturing, compared to conventional designs [30,79,104,107,123].](image)
size limitations, geometric accuracy, and unknown impact of surface roughness. McGlen [128] also provides a review of the application of AM-fabricated lattice structures for heat pipe wicks.

A few experimental studies have been conducted for thermal performance of metal AM lattices. Parbat et al. [129] investigated lattice structures made in IN718 via metal AM (Fig. 37), where the lattice consisted of ligaments of 0.5 mm diameter spread in multiple directions, and at two different porosities. A concurrent conjugate CFD analysis indicated that the ligaments promoted mixing in the channel, resulting in increased heat transfer and pressure drop relative to a smooth channel. Low porosity (high metal volume) increased heat transfer by about 10% but almost doubled the pressure drop. Fig. 37 shows the thermal performance (ratio of Nusselt augmentation to

\[
\frac{\frac{\text{Nu}}{\text{Nu}_0}}{f/f_0}^{1/3}
\]

(A)

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{lattice.png}
\caption{Lattice structures fabricated via AM (A), and ranking of their thermal performance (B) [129].}
\end{figure}
friction factor augmentation) for the two lattices studied (L1 = high porosity, L2 = low porosity), relative to other types of heat transfer augmentation approaches.

Liang et al. [130] compared bottom surface heat transfer around pin fins and various lattice types, including the Kagome lattice (not symmetric about the horizontal channel centerline) and body-centered cubic (BCC, symmetric) lattice. Both lattice structures resulted in higher $Nu$ on the surface of the channel relative to pin fins, with the BCC lattice slightly outperforming the average result of the Kagome lattice on the top and bottom surfaces. Again, the localized mixing caused by the lattice ligaments results in high heat transfer but also high pressure drop. Recognizing that AM allows for design freedom even within lattice geometry, Yun et al. [131] studied the effect of varying the lattice porosity along the flow length direction (Fig. 38), for both thermo-fluid and structural performance. Although none of the varying porosity designs were an improvement on pressure drop or heat transfer relative to the baseline, they observed a structural benefit from the W-patterned porosity profile due to reduced stress concentration along the channel.

Few experimental studies have been conducted on the thermohydraulic performance of TPMS surfaces, primarily because of the complexity of fabricating these structures except via AM. Clark et al. [132] performed magnetic resonance velocimetry measurements in a Schwarz-D (diamond basis pattern) TPMS at very low Reynolds numbers, concluding that the interconnected pores were good for mixing and the mixing varied somewhat with $Re$. Peng et al. [133] attempted to print a TPMS heat exchanger but encountered porosity issues in the thin walls comprising the TPMS surface causing it to fail a leak check. Cheng et al. [134] numerically investigated the flow and heat transfer in various TPMS morphologies (Fig. 39), and found that the P-type (also known as Schwarz P, or primitive TPMS) surface morphology had the highest heat transfer to friction factor ratio of all of the surfaces studied. Kaur and Singh [135] also studied TPMS surfaces in a conjugate CFD analysis and concluded that the gyroid surface had the highest convective heat transfer augmentation (including fin efficiency) relative to other TPMS types or stochastic metal foams. Gyroid and Schwarz-D TPMS surfaces were estimated to increase $Nu$ by up to 80% and 65%, respectively, relative to a printed circuit heat exchanger in the study of Li et al. [136]. No studies to date have experimentally tested the convective heat transfer performance of TPMS surfaces, so the impact of AM relevant surface roughness is unclear and more work is necessary.
5.3 Thin fins

Heat exchangers that process gas flows and their associated low conductivity can employ extended surfaces such as thin fins to provide additional surface area and convective heat transfer enhancement while balancing pressure drop. A significant amount of development has gone into optimal fin design.

Fig. 38 Variable porosity lattice structures (A), with thermal performance indicating no benefit relative to a uniform lattice at the velocities tested (B) [131].
for conventional manufacturing approaches, which generally consist of stamping of sheet metal into fin shapes that is then brazed to the parting plates. Additive manufacturing can enable fins of novel shapes [137–139], as well as fins coupled with other novel heat exchange surfaces, but a challenge of AM is limitations on thin feature sizes [140,141] and the inherent roughness. Recent work by the authors have investigated the impact of roughness on a common fin configuration known as an offset strip fin (OSF). OSF are attractive due to their interrupted nature, which results in boundary layer restarting and high convection coefficients. Also, for many

![Fig. 39 Triply periodic minimal surface types (A) and corresponding heat transfer to friction factor ratio (j/f, B) [134].](image)
orientations they can be fully unsupported, unlike louvered fins which may have surface angles on the louvers that are too low for AM fabrication even if the main fin orientation is acceptable. For conventionally manufactured OSF fins, performance has been well correlated by Manglik and Bergles [142] and Mochizuki et al. [143]. Since OSF are geometrically relatively simple, correlations typically involve not only Reynolds number but also geometric parameters like fin spacing to height ratio ($s/h$), thickness to length ratio ($t/l$), and thickness to fin spacing ratio ($t/s$).

To understand the effects of additive manufacturing roughness on fin performance, Saltzman and Lynch [144] generated offset strip fins at nominally true scale in AlSi10Mg aluminum alloy using an EOS M280, as well as an exact replica in plastic using a stereolithography process (Photocentric Liquid Crystal Pro). Table 4 gives the geometric dimensions and measured surface roughnesses (area-based arithmetic mean roughness, $S_a$) for the two geometries. Fig. 40 shows the orientation of the build as well as a part fabricated without an upper wall, which was replaced with glass for laser Doppler velocimetry flowfield measurements in the fin array (Fig. 41).

The laser Doppler velocimeter, with a measurement diameter of 117 μm, was able to acquire several measurement points within the wake of a fin for two rows. Fig. 42 shows the resulting mean streamwise velocity profiles and turbulence intensities for rows 3 and 17, as a function of distance between fins ($y/(s/2)$), such that $-1$ is the lower fin surface and $+1$ is the upper fin surface. As expected, the mean velocity is low in the wake of the upstream fin ($y/(s/2) = 0$), as well as near the upper and lower fin surfaces. Similarly, the turbulence intensity is higher in the wake and near the fins. However, when comparing the smooth and additive fins, the mean velocity is lower, and local turbulence intensity is higher, near downward facing surfaces for the metal additive fin particularly in the forward part of the array (Row 3). This is due to the larger boundary layer that is developing on the rougher downward-facing surface. Note also that the peak velocity is higher in Row 3 for the metal additive fin array, likely due to the overall larger boundary layers squeezing the core flow in the developing region of the array.

Fig. 43 illustrates the development of turbulence intensity in the center of the fin wake ($y/(s/2) = 0$) with distance into the fin array, for three Reynolds numbers. At low $Re = 300$ corresponding to purely laminar flow, the turbulence intensity is low and there is no significant difference between the smooth or additive fins. However, as Reynolds number increases to 600 (transitional), the rate of turbulence development dramatically increases after
<table>
<thead>
<tr>
<th>Value ± standard deviation</th>
<th>Fin thickness $t$ [mm]</th>
<th>Fin spacing $s$ [mm]</th>
<th>Fin length $l$ [mm]</th>
<th>Fin height $h$ [mm]</th>
<th>Upward facing roughness ($S_u$) [$\mu$m]</th>
<th>Downward facing roughness ($S_d$) [$\mu$m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design value</td>
<td>0.50</td>
<td>3.30</td>
<td>5.00</td>
<td>5.00</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>Smooth (Plastic)</td>
<td>0.51 ± 0.01</td>
<td>3.29 ± 0.01</td>
<td>4.88 ± 0.03</td>
<td>5.01 ± 0.03</td>
<td>2.4 ± 0.4</td>
<td>2.4 ± 0.4</td>
</tr>
<tr>
<td>Additive (Metal, AlSi10Mg)</td>
<td>0.63 ± 0.01</td>
<td>3.18 ± 0.01</td>
<td>5.00 ± 0.02</td>
<td>4.95 ± 0.02</td>
<td>14.6 ± 1.8</td>
<td>54.1 ± 4.2</td>
</tr>
</tbody>
</table>
row 6 for the metal additive fins relative to the smooth fins, and at the highest Reynolds number of 900, the turbulence intensity in the array is nearly at its saturation level of approximately 10% immediately after row 3. This suggests that the metal additive fins are causing earlier transition to turbulence in the array.

The implications of this behavior on the overall array friction factor is indicated in Fig. 44. Note that the slight differences in fin geometry indicated in Table 4 were accounted for in the definition of the Reynolds number and friction factor. Also shown is a correlation for smooth staggered
plates (essentially offset strip fins but without the bend features that would be present in stamped fins) by Mochizuki et al. [143]. At low Reynolds numbers, the rough additive fins are about 20% higher in $f$ than the smooth plastic fins. However, above a Reynolds number of 600, the additive fins exhibit increasing friction factor, with over 50% higher $f$ at the highest $Re$ tested.

Just as for the microchannel studies described earlier, it was expected that relative roughness might be critical for fin behavior. A separate study was conducted to determine this effect, in which OSF fins were printed in

Fig. 42 Mean streamwise velocity normalized by mass average velocity (A), and streamwise turbulence intensity (B), in the wake of AM fins at different row numbers.

Fig. 43 Turbulence intensity in the center of the fin wake ($y/(s/2)=0$) as a function of row number along the array, for different array Reynolds numbers.
316L stainless steel using an EOS M280 printer at the same build orientation as in Fig. 40. The fins were printed with two spacings: a tight spacing of $s = 1.40\text{mm}$, and a wide spacing of $s = 3.15\text{mm}$. Fin thickness was nominally $t = 0.35\text{mm}$, fin height $h = 5\text{mm}$, and fin length $l = 5\text{mm}$. A smooth replica with the same geometry was printed using a Photocentric Liquid Crystal Pro. Arithmetic mean roughness of the smooth replica was the same as in Table 4, while the stainless fins had an upskin roughness of $15.5\,\mu\text{m}$ and a downskin roughness of $26.4\,\mu\text{m}$. Note that the upskin roughness was similar to the AlSi10Mg fins, while the downskin roughness is almost half that of the value in Table 4.

Fig. 45 shows the overall friction factor vs Reynolds number for both additive and smooth materials, as well as the ratio of friction factor for additive vs smooth, for the two fin spacings considered. As was observed in the earlier fin flowfield study (Fig. 44), the wide fin spacing here generally agrees with smooth fin behavior for an extended range of Reynolds number. In this case, the $s = 3.15\text{mm}$ spacing only starts to deviate from smooth fin behavior after $Re = 2000$; this is likely because of the moderate downskin surface roughness for these stainless steel fins. However, when the fin spacing is reduced to $s = 1.4\text{mm}$, the friction factor is significantly increased, even in the laminar regime. Interestingly, this is a similar phenomenon to that shown in Fig. 13 for additive microchannels. For these fins, the friction factor augmentation relative to the smooth geometry is approximately 60% in the laminar regime but increases to more than 220% in the turbulent regime, with the increase occurring at a lower Reynolds number than for the wider spaced fins.
The Colburn $j$-factor was also measured for the stainless steel fins, using a constant surface temperature condition established by the test rig (fin efficiency was also accounted for). The results are shown in Fig. 46 for the two fin spacings. In the laminar region ($Re < 2000$), the tight fin spacing of $s = 1.40\, \text{mm}$ exhibits over 40% higher heat transfer than the wider spaced fin array, which is nominally the same as a smooth fin. This suggests that the relative roughness to fin spacing positively benefits the array heat transfer, likely by increasing turbulent motions in the array.

Fig. 45 Array friction factor for two fin spacings as a function of Reynolds number (A) and ratio of additive to smooth friction factor (B), for the metal additive (316L stainless) and smooth plastic fins.

The Colburn $j$-factor was also measured for the stainless steel fins, using a constant surface temperature condition established by the test rig (fin efficiency was also accounted for). The results are shown in Fig. 46 for the two fin spacings. In the laminar region ($Re < 2000$), the tight fin spacing of $s = 1.40\, \text{mm}$ exhibits over 40% higher heat transfer than the wider spaced fin array, which is nominally the same as a smooth fin. This suggests that the relative roughness to fin spacing positively benefits the array heat transfer, likely by increasing turbulent motions in the array.

The results in Figs. 45 and 46 include a correlation for each of the fin spacings using the matched asymptotes method [145]. Since there are only
two fin spacings available to date, it is not yet possible to develop a robust correlation to capture the effect of relative roughness on fin spacing, but this is ongoing work. The correlations for each of the spacings are:

\[
f_{s=3.15} = \left[ (14.963Re^{-0.813})^{2.86} + (0.05Re^{-0.003})^{2.86} \right]^{0.35} \tag{9}
\]

\[
f_{s=1.40} = \left[ (15.223Re^{-0.767})^{10} + (0.0969Re^{-0.005})^{10} \right]^{0.10} \tag{10}
\]

\[
\frac{j_{\text{additive}}}{j_{\text{smooth}}} = 0.494Re^{-0.525} \tag{11}
\]

\[
\frac{j_{s=1.40}}{j_{s=3.15}} = 1.9244Re^{-0.660} \tag{12}
\]

Finally, the combined performance of the fins is considered in Fig. 47. This plot shows the ratio of \( j \) to \( f \) to the 1/3 power, which is also commonly referred to as a goodness factor. Due to the somewhat higher friction factor...
but similar heat transfer performance of the $s = 3.15 \text{ mm}$ fin spacing, it has a goodness factor lower than the smooth-fin correlation of Mochizuki et al. \cite{143} for all Reynolds numbers. However, the tight fin spacing of $s = 1.40 \text{ mm}$ has a slightly higher goodness factor at low Reynolds numbers relative to smooth fins due to the improved heat transfer. This behavior converges to the wide-spacing additive fin behavior at higher $Re$, which falls increasingly below that for smooth fins. Thus, additive fins can provide some benefit in a heat transfer application but primarily at low Reynolds numbers and with tight fin spacings. Further investigation of the correlation between fin geometry and roughness is needed.

An interesting opportunity for AM is the development of novel fin shapes. Unger et al. \cite{146} generated circular fins around an oval pipe in 316L stainless steel, with novel features such as integrated pins, or serrations with integrated pins. The Nusselt number of the serrated and circular fins with integrated pins was improved relative to plain circular fins, although friction factor was increased. The most effective design for compactness of a heat exchanger was the circular fin with integrated pins.

### 6. Heat exchangers

Heat exchanger (HEX) design is a complex multiobjective problem strongly defined by the application, but also by the limitations of fabrication technologies. The potential for AM to generate complex shapes, especially
in high temperature tolerant alloys that are typically difficult to work [147], has generated a lot of interest in the community.

The largest number of studies have investigated a class of heat exchangers known as manifold microchannel heat exchangers (MMHEX) [148–152]. As depicted in Fig. 48, these designs consist of many small-length micro-channels which are fed via large manifolds, so that the flow in the microchannel is in a thermally developing state with high convection coefficients but the pressure loss is minimized by the short channel length. AM is advantageous for this design since conventional joining processes that are used to bond together the manifold-microchannel stack (such as brazing, welding, or
diffusion bonding) can block the microchannels. Zhang et al. [152] demonstrated a metal AM MMHEX made from Inconel 718, which was able to process incoming nitrogen gas at 600°C and 450 kPa and incoming air at 38°C. Compared to a plate-fin heat exchanger at matched flowrates, pressure drop, and heat exchanger effectiveness, the manifold microchannel HEX had a 25% improvement in core heat transfer density (kW/kg), implying that the heat exchanger provides more intense heat transfer per unit mass. This was even taking into account the minimum wall thickness of 0.5 mm in the manifold microchannel design, relative to a typical plate-fin thickness of 0.3 mm.

An optimization study by Arie et al. [149] for an air-liquid application relevant to dry cooling of power plants suggested that the manifold microchannel design could increase heat transfer density by more than 60% compared to a conventional wavy plate-fin design, with larger increases possible if minimum feature sizes could be accomplished. Keramati et al. [153] focused on this aspect by examining the possible printing resolution from various LPBF machines for a MMHEX core. The thinnest microchannel walls created were 0.133 mm made from a maraging steel alloy (carbon free iron-nickel alloy, 3D Systems specification), although Inconel 718 was only able to reliably print sizes to 0.22 mm. Other studies by the same group indicated strong performance of MMHEX in liquid-liquid heat exchange applications [151,154]. Alsuilami et al. [150] investigated complex features enabled by AM embedded into the manifolds of MMHEX as well as reorientation of the microchannels and found that there was not really a significant improvement in performance, likely because the MMHEX were already highly efficient.

Another series of studies have examined oil coolers for land based or aerospace applications, where the air-side geometry employs fins. Saltzman et al. [155] replicated a commercially available plate-fin aircraft oil cooler in AlSi10Mg material using PBF (Fig. 49). Their AM design was based on a CT scan of a traditionally manufactured stamped aluminum version, with the same fin spacing and plate dimensions but a few changes for printability in PBF. A second model was also developed with multiple internal vortex generators (VG), which could be added with no additional complexity in the AM part but would be difficult or impossible in the stamped aluminum part. Manifolds for the liquid plates in the AM model were simplified relative to the stamped aluminum also. Characterization of the AM build indicated surface average roughness that was about 75 times larger than the stamped aluminum (24 μm for PBF vs 0.31 μm for stamped aluminum). Fig. 50 shows the measured air-side friction factor, and the nondimensional overall heat transfer coefficient (presented as number of transfer units, NTU), as a function of
Four cases are shown: the baseline geometry with 11 fins per inch (FPI), a higher FPI model but also traditionally manufactured, the 11 FPI AM geometry, and the 11 FPI geometry with embedded VGs. For the friction factor, two correlations for louvered fins are shown [156,157]. The figure indicates similar behavior of $f$ to that observed in the fin coupon study, with high $f$ at all Reynolds numbers compared to correlations and a leveling off of $f$ with $Re$, due to the high surface roughness. The overall heat transfer coefficient is also higher for the AM HEX relative to the traditional geometry of the same FPI, but interestingly is the same as a higher FPI traditional geometry, indicating a potential to achieve similar heat transfer at reduced weight (fewer FPI).
A later study was performed by Bichnevicius et al. [158] on the same geometry also manufactured in AlSi10Mg via PBF, but with more aggressive laser speed settings (for reduced build time). Performance testing indicated further increases in heat transfer but also pressure drop. Sectioning of the AM heat exchanger indicated that the thin-walled internal geometry features, particularly fin louvers and plate internal walls, were extremely rough and in some cases had large defects indicative of significant melting. This resulted in a pressure drop of over 200% relative to the stamped aluminum baseline, although heat transfer was increased by 38% relative to the baseline.

As might be expected, an increasing number of studies generate novel AM HEX designs with nontraditional heat exchange shapes. These can range from relatively modest shape changes, to the very complex surfaces described earlier; Jafari and Wits [159] and Dbouk [160] review several studies. Hathaway et al. [161] and Garde et al. [162] improved on an existing oil cooler design used for a construction vehicle by additively manufacturing lenticular tube shapes 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 reduced the effective heat transfer area. Kong et al. [163] fabricated a counterflow rectangular minichannel heat exchanger core (termed honeycomb core) in 316L stainless steel with varying channel size along the flow direction, and found that their design increased heat transfer by about 17% but also increased pressure drop by 25%. Sabau et al. [164] designed a relatively high pressure (750 psia) heat exchanger architecture that transitioned from tube flow to shell flow for a given fluid, where the tubes were triangular rather than circular (as in a conventional shell and tube HEX). There were some challenges identified for the PBF fabrication method, but two different designs were ultimately fabricated and tested against a commercial 5kW shell and tube heat exchanger. Although the pressure drop was somewhat higher for the AM heat exchangers, the overall heat transfer coefficient was approximately 16–32% higher, indicating potential for compactness.

Regarding more complicated shapes, Gerstler and Erno [165] developed a fuel-cooled oil cooler (liquid-liquid HEX) using a multi-furcating geometry, similar in concept to TPMS surfaces. They successfully fabricated the design in four different materials: AlSi10Mg, Titanium 6–4, cobalt chrome, and Inconel 718. Relative to the shell and tube baseline design, the multifurcating HEX demonstrated the same heat exchange and pressure drop
performance with 66% less weight and 50% lower volume. Greiciunas et al. [166] developed and tested an interlayer plate HEX using commercially pure titanium, where plate layers have multiple crossing tubes to increase surface area and promote better flow distribution in the core. Although there was no comparison to a conventional geometry, the computational model of the HEX matched the measured performance well. Saviers et al. [167] presented a topology optimization approach for a supercritical CO$_2$ counterflow heat exchanger, which reduced pressure drop to half of the baseline value at a given amount of heat transfer.

7. Conclusions and future needs

Throughout this review, we have shown the importance of considering the opportunities and the impacts of using additive manufacturing for heat exchange surfaces. The inherent roughness of AM is dependent upon process parameters and on build direction, and with each comes important design considerations. As the build direction changes, both channel shape and roughness levels can be altered. An improved relation between measured surface roughness (such as from a profilometry technique) and sandgrain roughness (impact on microchannel pressure drop) is presented based on measurements from a large number of AM microchannels. We have also shown how the expected heat transfer is affected by these roughness levels, which was found to correlate well with the sandgrain roughness approach. For novel microchannels, or for thin-walled features such as fins, the roughness and feature resolution can lead to unique friction and heat transfer behavior relative to conventional manufacturing of those shapes. We have reviewed progress in fabricating functional heat exchangers using AM, including novel shapes that are not possible through other manufacturing approaches.

Despite the impressive progress achieved to date for the application of AM toward heat transfer applications, there is more work needed to fully realize its potential. For the PBF process described in this review, improved feature resolution, capability for thinner features, faster fabrication speeds, and tailored process control could open up higher fidelity cost-effective fabrication. Improved or new post-processing techniques to control or tailor surface roughness could help the heat transfer designer take advantage of roughness when appropriate. Regarding materials, there is a need to continue to develop AM-ready alloys with advantageous properties. An example is copper, which is extremely challenging to build using PBF due to its
high thermal conductivity the same property that makes it an excellent material for heat transfer applications. Another example are refractory alloys which are capable of extreme temperature tolerance but suffer from severe cracking issues when fabricated using LPBF. Finally, design tools need to advance to take advantage of complexity enabled by AM while properly accounting for roughness or feature resolution impacts, perhaps even including the manufacturing process in the optimization of a part.

Based on our review, there seems to be no shortage of interesting research and development problems in the field of additive manufacturing for heat exchange applications. It will become increasingly necessary for the practitioner to have a multidisciplinary view of the field, with expertise in materials, manufacturing, and of course thermofluids.

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