Numerical Optimization, Characterization, and Experimental Investigation of Additively Manufactured Communicating Microchannels

The degree of complexity in internal cooling designs is tied to the capabilities of the manufacturing process. Additive manufacturing (AM) grants designers increased freedom while offering adequate reproducibility of microsized, unconventional features that can be used to cool the skin of gas turbine components. One such desirable feature can be sourced from nature; a common characteristic of natural transport systems is a network of communicating channels. In an effort to create an engineered design that utilizes the benefits of those natural systems, the current study presents wavy microchannels that were connected using branches. Two different wavelength baseline configurations were designed; then each was numerically optimized using a commercial adjoint-based method. Three objective functions were posed to (1) minimize pressure loss, (2) maximize heat transfer, and (3) maximize the ratio of heat transfer to pressure loss. All baseline and optimized microchannels were manufactured using laser powder bed fusion (L-PBF) for experimental investigation; pressure loss and heat transfer data were collected over a range of Reynolds numbers. The AM process reproduced the desired optimized geometries faithfully. Surface roughness, however, strongly influenced the experimental results; successful replication of the intended flow and heat transfer performance was tied to the optimized design intent. Even still, certain test coupons yielded performances that correlated well with the simulation results. [DOI: 10.1115/1.4044194]

Introduction

As the maturity of advanced manufacturing techniques grows, the heat transfer community is offered an opportunity to develop new, unique cooling designs. In hot section components of a gas turbine engine, those unique designs can find an end use in skin cooling, an inherently challenging technology to produce due to its necessarily small size.

Additive manufacturing (AM) represents a potentially viable method for achieving complex skin cooling schemes. Various metal AM processes exist, all of which are capable of using aerospace grade materials suitable for high temperature applications. Laser powder bed fusion (L-PBF) processes, as one example, can achieve the precision required to produce microsized features. Additionally, using techniques such as L-PBF widens the design possibilities.

One source of inspiration for new designs can be found in nature; intricate networks of communicating channels are ubiquitous in biological systems and provide an efficient means of transport. In the current study, a previously conceived wavy microchannel design [1] was modified to included branches between neighboring channels, creating a complex array through which the cooling air would travel. The implementation of the branches was meant to augment heat transfer by disrupting the boundary layer formation on the primary channels’ walls; the angle of the branches was chosen such that the penalty to the pressure loss would be minimized.

Two different variations of the communicating wavy channel design were conceived. Each design was then optimized for a given objective through an adjoint-based shape optimization method [2]. The three objective functions sought to (1) minimize the pressure loss between the channel inlet and exit, (2) maximize the heat transfer on the channel top and bottom end walls, and (3) maximize the ratio of heat transfer to pressure drop. The baseline and optimized channels were then manufactured using L-PBF and nondestructively evaluated to determine the build success. Finally, the test coupons were experimentally tested for pressure loss and heat transfer at a range of Reynolds numbers between 2000 and 15,000.

Literature Review

The origin of several aspects of the current study stems from the electronics cooling industry. Wavy channel studies from the literature have widely focused on low Reynolds number applications [3–7]; wavy channels promote mixing in the flow due to the vortical structures they generate, which is beneficial to the heat transfer. However, the pressure drop through wavy channels has been shown to be only marginally higher than that through straight channels [3].

A series of communicating channel studies has also been performed for use in electronics cooling [8–11]. In these studies, straight channels were connected via branches oriented perpendicular to the flow direction. A notable result from one such study by Herman et al. [8] was that an increase in both heat transfer and pressure drop was seen at a low Reynolds number (Re = 366), when compared to a straight, noncommunicating channel. With an increase in Reynolds number (Re = 593), however, the authors cited higher heat transfer with no further penalty to pressure loss. Singh et al. [3] investigated communicating wavy microchannels in a numerical study and found that the branches increased the heat transfer; the branches were angled at 45 deg from the flow direction, making them a more aerodynamic feature than the studies in Refs. [8–11]. Peng et al. [12] mimicked the branching...
structures found in leaf veins, including the porous walls that make up those veins. The porosity in their heat sink proved beneficial to the heat transfer, while exerting minimal influence on the friction factor augmentation.

Weaver et al. [13] studied wavy channels at engine-relevant Reynolds numbers; their concept incorporated a layered design, with both the channel waves were derived from the layered weave of the channels. The design achieved uniform distribution of the cooling flow while minimizing both the pressure loss and the required mass flow through the weave. Kirsch and Thole [1,14] also investigated wavy channels at more realistic Reynolds numbers for gas turbine applications and manufactured the channels using L-PBF; in Ref. [14], the authors optimized the shape of the wavy microchannels based on the same objectives used in this current study. Experimental results showed that the wavy channels exhibited higher heat transfer than straight channels of the same aspect ratio and spacing [1]. Additionally, the as-built optimized shapes were reproduced fairly well, with the objective to maximize the ratio of heat transfer to pressure loss having yielded a performance akin to the simulations.

Other studies that have used optimization in conjunction with L-PBF heat exchangers include Dede et al. [15], who used topology optimization, and Arie et al. [16], who performed a multi-objective optimization study. In both cases, the optimized design outperformed the conventional. Several numerical studies exist as well, with the goal of furthering design tools available to heat exchanger designers [17–20].

As a consequence of any powder bed fusion AM process, large, irregular surface roughness features form on all surfaces [21,22]. Bacchewar [23] identified laser power as the strongest contributor to surface roughness on downward facing surfaces, or surfaces that are unsupported by solid material. The build direction also has a heavy influence on the roughness features that form inside of a part [21,24,25], as well as the ability to reproduce a certain shape accurately [21,24]. Part placement and inert gas flow have also been found to influence the roughness [26,27], as have the laser scan speed and the hatch distance [28,29].

In the case of heat exchangers, surface finish has a large impact on the pressure loss and heat transfer. Wong et al. [30] and Ventola et al. [31] produced AM heat exchangers and found that the surface roughness positively influenced the heat transfer; the AM surfaces at all external flow features. Stimpson et al. [22] and Snyder et al. [21] found that the surface roughness in their internal L-PBF channels augmented the heat transfer only to a certain point. Relative roughness, or the size of the roughness features compared to the hydraulic diameter, played a large role in the performance of these microchannels; higher relative roughness more strongly affected the friction factor than the heat transfer. In fact, beyond a friction factor augmentation of four, surface roughness no longer served to increase the heat transfer, a finding that confirmed results from Norris [32].

Part shrinkage or warpage can distort AM part features or cause the final dimensions to vary from the design intent. Ning et al. [33] found that parts less than 3 mm in size were susceptible to shrinkage up to 10% of the part’s design dimensions. For this reason, properly characterizing the final part is a necessary step in analyzing the performance of AM parts. Computed X-ray tomography (CT) scans are a common method for determining internal structures found in leaf veins, including the porous walls that make up those veins. The porosity in their heat sink proved beneficial to the heat transfer, while exerting minimal influence on the friction factor augmentation.

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Effectively taking advantage of the open design space offered by AM requires thinking outside the conventional designs for heat exchangers [34,36]. The current study is unique in that it combines a conventional cooling channel design with an idea derived from transport systems in nature. Further, these baseline designs have been numerically optimized, thereby taking advantage of advanced, commercially available design tools. The branches between the wavy channels create an intricate array of communicating cooling channels aimed to increase the heat transfer without substantially increasing the pressure loss.

Channel Design

The wavy channels were created by sweeping a rectangle along a path created using 45 deg arcs; each period in the channel contained four such arcs, whose edge tangents were equal. Figure 1 shows the design of the communicating channels. The white space is the channel area, while the gray represents the walls. The four 45 deg arcs in one wave period are shown, along with the branch angle, which was also at 45 deg to the tangent at the peak or trough of the channel wave.

The width of the channel was twice the width of the branches. The inlet to the branches was filleted to minimize losses while the exit of the branches was mean to encourage the flow to penetrate into the neighboring channel. The baseline designs were characterized by the ratio of their wavelength, \( \lambda \), to the entire test coupon length, \( L \). Two ratios were chosen for this study: \( \lambda = 0.1 \) and \( \lambda = 0.4 \). The length of the coupon was such that ten periods of the \( \lambda = 0.1 \) case fit in the streamwise dimension of the coupon, while 2.5 periods were present in the \( \lambda = 0.4 \) case.

Figure 2 shows four channels for each case; white space represents the channel, while the gray color illustrates the walls. The branches repeated every other period in the streamwise direction and every other channel in the spanwise direction; this alternating pattern can be seen in Fig. 2. A total of twenty channels fit in the spanwise dimension of the test coupon for the \( \lambda = 0.1 \) case, while the test coupon for the \( \lambda = 0.4 \) case contained eighteen channels. Spacing between the channels was two channel hydraulic diameters, \( D_h \).

Numerical Setup

The optimization portion of this study was reliant upon the results from an initial simulation of each baseline configuration. A high quality, wall-resolved, structured mesh of the fluid domain was created, which contained two neighboring channels with their corresponding branches. Only the fluid domains were modeled, meaning the conjugate problem was not solved. The side walls of the channels were modeled as adiabatic. To solve the steady Reynolds-averaged Navier–Stokes and energy equations, the realizable \( k-\epsilon \) turbulence model was chosen [2]; the boundary conditions in the model were set for a Reynolds number of 5000, and mimicked those from the experimental setup. Both the mesh and select boundary conditions are shown in Fig. 3. Heat was introduced into the system from a constant temperature boundary condition imposed on the top and bottom end walls (relative to the \( z \)-dimension, and not shown in Fig. 3).

Each mesh contained 2.1 million cells, and the \( y^+ \) values remained near or below one over the entire length of the domain. A grid-sensitivity study was performed on a similar geometry in a previous study [1]; doubling the number of cells in the mesh.
Adjoint-Based Shape Optimization. The converged solutions to the steady Reynolds-averaged Navier–Stokes and energy equations provided input for the subsequent optimization analysis. Embedded within the flow solver [2] was a tool to perform a sensitivity analysis, which was equipped to inform shape change. The sensitivity analysis was accomplished using an adjoint method, whose computational efficiency is epitomized by its handling of numerous degrees-of-freedom. A full mathematical description of the adjoint method is given in Kirsch and Thole [14], but a brief description will be given here.

Three different objective functions were imposed on each of the baseline designs. The objective functions are denoted as $J$, and are shown in Eqs. (1a)-(4c). The third objective function reflects a commonly used performance factor in internal cooling studies [37]

\[
J_1 = \text{min}(\Delta P)
\]  
\[
J_2 = \text{max}(Q)
\]  
\[
J_3 = \text{max}(Q/\Delta P^{1/3})
\]

The adjoint approach introduces a set of equations, known as the adjoint equations, to be solved to convergence, much like the governing equations for a flow simulation. Contained within the adjoint equations is the relation between the governing equations and the chosen objective function for a given flow field. The results from the converged adjoint equations can be used to determine the sensitivity of the design variables on the objective function. In the current study, each one of the nodes in the computational mesh represented a design variable.

Based on the results of the sensitivity analysis, the shape of the geometry at hand can be changed such that the most sensitive nodes are moved in an advantageous direction. The user defines the degree to which the shape is changed. The software [2] then morphs the mesh to reflect this shape change and ensures that the quality of the mesh remains the same as the original; additionally, $y^+$ values from subsequent flow solutions are generally unaffected.

One design iteration encompasses running the flow solver, running the adjoint solver, and morphing the mesh. Design iterations continue until either no further changes to the shape are suggested, or the flow solution no longer converges given the latest shape change. In the current study, between five and ten design iterations were needed to reach the optimized result in the simulations. Constraints were imposed on the overall length of the channels such that they would fit into the same coupon dimensions. Additionally, the periodic boundary conditions for the branches were required to change in the same manner. However, beyond these constraints, all mesh nodes represented a degree-of-freedom, and the resulting shape changes were complex and aperiodic.

Numerical Results

All three objective functions for both wavelengths were achieved. Tables 1 and 2 show the change in the observables of interest relative to the respective baseline configurations for the $\lambda = 0.1$ L and $\lambda = 0.4$ L cases, respectively. Bolded values in the two tables showcase the end result of the target observable, while the other two observables are shown for comparison.

The majority of shape changes occurred in areas surrounding the branches. Figure 4 shows three zoomed-in locations from a top-down view of the channel outlines at 50% channel height. At the top of the figure is a top-down view of the entire channel length; the three locations of the zoomed-in images are outlined in bold rectangles. The walls are colored gray, and are colored based on the outline of the baseline geometry. Outlines of the optimized channels cutting through gray color, therefore, show that the channel area has expanded relative to the baseline, while any outlines in the white area (fluid domain) represent a contraction relative to the baseline. Additionally, two deviations from the baseline are quantified for the optimized designs for reference.

A cursory glance at Fig. 4 reveals a marked difference between the outlines of the $J_1$ objective to minimize pressure loss and the two objectives with heat transfer as an observable, $J_2$ and $J_3$. In general, where the walls of the objective to minimize pressure loss moved outward, the walls of the other two objectives moved inward, and vice versa. In specifically looking at the outline of the $J_1$ objective in Fig. 4(c), the angle of the branch exit shifted closer to the direction of flow, and the opposite wall bowed outward slightly. This shape change served to decrease fluid momentum, and thus, reduced the extent to which fluid from the branches penetrated the neighboring channels.

Evidence for this claim can be seen in Fig. 5, which shows normalized axial velocity contours from each of the baseline and optimized configurations; the slice location is immediately following the branch exit highlighted in Fig. 4(c). The $J_1$ objective shows the lowest normalized velocity across the entire width of the channel. Perhaps most notably, the normalized velocity on the leeward wall (left wall, from the reader’s point of view) is only marginally higher than the mean velocity, $U_{mean}$. This result suggests that fluid from the branches hardly penetrated to the far wall, an assuredly intended consequence of the shape changes in the

Table 1 Changes in $\Delta P$, $Q$, and $Q/\Delta P^{1/3}$ relative to $J_1$ = 0.1 L baseline

<table>
<thead>
<tr>
<th>$J_1$ = min($\Delta P$)</th>
<th>$J_2$ = max($Q$)</th>
<th>$J_3$ = max($Q/\Delta P^{1/3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta P$ (%)</td>
<td>$Q$ (%)</td>
<td>$Q/\Delta P^{1/3}$ (%)</td>
</tr>
<tr>
<td>-4.4</td>
<td>3.9</td>
<td>-2.5</td>
</tr>
<tr>
<td>+6.6</td>
<td>+6.6</td>
<td>+4.3</td>
</tr>
<tr>
<td>+0.4</td>
<td>+1.92</td>
<td>+1.8</td>
</tr>
</tbody>
</table>

Note: The bold text refers to the end result of the target observable.
channels and branches. By contrast, the baseline, $J_2$, and $J_3$ cases all showed high normalized velocity on both the windward and leeward walls, thus validating the original design intent to enable jet formation.

Moving slightly downstream from Fig. 5, Fig. 6 shows nondimensional temperature contours, with secondary velocity vectors overlaid, at the trough of the wave. Immediately visible in Fig. 6 for the baseline, $J_2$, and $J_3$ cases are the presence of Dean vortices, flow features that are commonly found in wavy channels due to the centripetal forces experienced by the fluid particles [3]. To note, Kirsch and Thole [14] found no such vortical formation in the baseline design of the noncommunicating $\lambda = 0.1$ $L$ microchannels; insufficient flow development length was posited to explain the absence of the Dean vortices. The introduction of the branches in the current study, however, appeared to have increased the fluid momentum enough to form the characteristic vortical pattern.

Given the velocity contour of the $J_1$ objective function in Fig. 5, the lack of fully formed vortical structures in the $J_1$ slice in Fig. 6 is unsurprising. The secondary flows present in the $J_1$ slice are weaker than those seen for any of the three other geometries. The shape changes that prevented jet-like flow from the branches led to the inability of Dean vortices to form. These observations in flow patterns are consistent with the overall numerical results (Table 1), which showed a 4.4% decrease in pressure loss from the $J_1$ objective relative to the baseline.

Contours of nondimensional temperature also support the findings from the objective to maximize heat transfer shown in Table 1: the highest nondimensional temperature can be seen in the $J_2$ slice in Fig. 6, especially at the channel midheight. The vortices are strongest in the $J_2$ slice, confirming that the stronger vortices are more effective at removing heat from the channel end walls.

Figure 7 presents the outlines of the baseline and optimized channels for the $\lambda = 0.4$ $L$ configuration in a similar manner to Fig. 4; the channel outlines were taken at 50% the channel height, and the gray color illustrates the walls, while the white color is the fluid domain. The walls are again colored to the edge of the baseline design. For reference, several deviations from the baseline design are quantified.

Striking differences between the optimized and baseline branch locations can be seen in Fig. 7, most notably in Fig. 7(a). To note, the outlines of the optimized branches do not reflect a translation of the branch locations; instead, the outlines show how far the branch had arched at the channel midheight. At the top and bottom end walls, all four cases’ branch locations matched. Where heat transfer was to be maximized ($J_2$ and $J_3$ objective functions), the branch shape at 50% channel height jutted forward, in the direction of the channel inlet. By contrast, the midheight branch shape for the objective to minimize pressure loss ($J_1$) extended aft. This behavior strongly affected the interaction between flow emanating from the branches and the main channel flow.

Slightly downstream of the branch location highlighted in Fig. 7(a) came further notable shape changes. The location of the slices shown in Fig. 8 was taken to be where the radius of curvature in the wave design switched signs (Fig. 1). The slices are shown...
with contours of normalized axial velocity. Figure 8 reveals stark differences in the patterns of the shape changes; objective functions $J_2$ and $J_3$ showed a sharp inward bow of the leeward wall, whereas the leeward wall of the $J_1$ objective function bowed outward. Additionally, the contour levels in each of the slices varied among the objectives. The arched nature of the branch shapes, along with the shape changes downstream of the branches, either encouraged ($J_2$ and $J_3$ objectives) or suppressed ($J_1$ objective) the intended jet formation.

The downstream effect of the jet penetration (or lack thereof) can be seen in Fig. 9, where nondimensional temperature contours are shown for a slice location at the trough of the channel; secondary velocity vectors are overlaid. For the $J_2$ and $J_3$ objectives, where a windward bend in channel shape resulted in higher normalized axial velocity (seen in Fig. 8), Fig. 9 shows characteristic Dean vortex formation. Additionally, nondimensional temperature is visibly higher on the windward walls of the $J_2$ and $J_3$ objectives than for either the baseline or $J_1$ cases.

By contrast, no vortical structures were seen in the slice for the $J_1$ objective; expanding the leeward wall at the location highlighted in Fig. 8 lowered the fluid momentum, which resulted in a lack of vortex formation in the channel trough.

The vector patterns in Fig. 9 support the results shown in Table 2. Pressure loss was decreased for the $J_1$ case relative to the baseline by 3.2%, due most likely to the lack of Dean vortices, while heat transfer was increased by 11.7% and 3.8%, respectively, in both the $J_1$ and $J_2$ cases due to increased fluid mixing.

The intent of the communicating nature of the microchannels was to increase the heat transfer while mitigating any increase in the frictional losses. As such, the communicating channel simulation results can be compared to the simulation results from the noncommunicating channels [14]. Table 3 shows the difference in $\Delta P$ and $Q$ between the communicating wavy channels relative to their respective noncommunicating counterpart. In general, the design intent was successful. For both cases, the benefit to heat transfer outweighed the detriment to pressure loss. Further exploration into the differences between communicating and noncommunicating channels will occur in the “Experimental Results” section.

**Geometric Characterization**

Once the optimization analysis was complete, each set of channels was duplicated to fit into a test coupon for experimentation. Each of the test coupons was manufactured using L-PBF [38] with Inconel 718 powder, and was built layerwise at a 45 deg angle to the build plate. The machine specifications were set to the default for Inconel 718 [39]. Figure 10 shows two views of the same coupon on the build plate, with one view looking into the flow direction and another looking at the front face of the test coupon. Pertinent dimensions are given for scale. Support structures were located on the coupon flanges, as well as the downward-facing sidewall of the test coupon itself.

Once the build was complete, the parts were heat treated to remove any residual stresses that accumulated during the build.

### Table 3 Changes in $\Delta P$ and $Q$ relative to noncommunicating channels

<table>
<thead>
<tr>
<th>$k = 0.1 L$</th>
<th>$k = 0.4 L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta P$ (%)</td>
<td>$Q$ (%)</td>
</tr>
<tr>
<td>$J_1 = \min(\Delta P)$</td>
<td>$-4$</td>
</tr>
<tr>
<td>$J_2 = \max(Q)$</td>
<td>$+7$</td>
</tr>
<tr>
<td>$J_3 = \max(Q/\Delta P^{1/3})$</td>
<td>$+0.4$</td>
</tr>
</tbody>
</table>
process. The parts were then removed from the build plate using a wire electro-discharge machine, and any other remaining support pieces were manually removed from the parts. The flange surfaces of the test coupons were smoothed to ensure good contact between the coupon and test section, but all other surfaces remained untouched.

A critical step in analyzing the experimental results lies in measuring the internal channel dimensions. Nondestructive evaluation of the test coupons was accomplished using CT scans, which allowed for a voxel size (resolution) of 35 \( \mu\)m. Analysis of the CT scan data can be performed in a similar manner to the numerical results analysis, by visualizing 2D slices of the as-built test coupon.

Figure 11 shows a top-down view at 50% channel height of the as-built baseline and optimized channels. The format of Fig. 11 mimics that of Fig. 4, with all three zoomed-in locations in Fig. 11 matching those from Fig. 4. Additionally, for reference, a dotted line showing the intended baseline design is included.

A few notable features in Fig. 11 stand out, perhaps the first of which is the large protrusion in the \( J_2 \) outline (red) seen at location (a), in addition to the smaller protrusions visible at location (c). These protrusions are roughness elements, a natural consequence of the L-PBF process. These roughness elements can extend into the flow as far as 25% of the channel width, and are highly irregular; as such, comparable protrusions are seen across all four cases, most especially on unsupported surfaces. The unsupported surfaces can be identified in Fig. 11 by any outlines that are jagged in nature. By contrast, smooth outlines indicate a supported surface.

A second notable feature in Fig. 11 can be seen in the inability of the L-PBF process to reproduce the branch feature designed to encourage jet penetration. Specifically at locations (a) and (c), the dotted line depicting the intended baseline design lies isolated from the as-built outlines, and extends farther into the channel than any of the as-built channels. Due to the chosen build direction, this particular design feature was unsupported and, because of its small size, did not build properly. Given the complicated nature of these channel designs, however, no one build direction could have fully supported all internal features.

The dimension emphasized in Fig. 11 highlights the capability of the L-PBF process to produce the intended inward bow in the \( J_2 \) and \( J_3 \) objectives. As shown in Fig. 4, the intended geometric feature was to bow inward by approximately 80 \( \mu\)m; the as-built dimension was closer to 120 \( \mu\)m. By contrast, the outward bow intended in the \( J_3 \) design was not successfully reproduced. Figure 4 shows that the difference between outlines of the baseline and \( J_3 \) objective was to be around 50 \( \mu\)m at 50% channel height. Such a deviation was not enough to be captured by the L-PBF process faithfully.

A number of design intentions, however, were successfully reproduced in the build. For example, the optimizer created a slight contraction in flow area in the branch for the \( J_2 \) and \( J_3 \) objectives (Fig. 4(a)), and indeed, the as-built branch outlines for the \( J_2 \) and \( J_3 \) objectives followed suit (Fig. 11(a)). Location (b) shows a similar trend in branch flow area. Additionally, the main channel flow area was to contract where the fluid exited the branch and entered the main channel for objectives \( J_2 \) and \( J_3 \) (Fig. 4(a)), a behavior that was also reproduced in the as-built channels (Fig. 11(a)).

Figure 12 shows select CT scan results for the \( \lambda = 0.4 \) \( L \) case, presented in the same manner as Fig. 11; all locations outlined in Fig. 12 are the same as those highlighted in Fig. 7. The outlines of the baseline intended designs are also included for reference and are shown in dotted lines in the figure; notable deviations of the optimized designs from the baseline are quantified as well.

Much like Fig. 11, Fig. 12 shows large roughness features that formed on the unsupported surfaces in the channels. Especially visible at locations (a) and (c), these roughness features are concentrated near the branch feature intended to encourage jet penetration. Just as for the \( \lambda = 0.1 \) \( L \) case, this small feature was not built successfully because it was unsupported throughout the build.

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Much like Fig. 11, Fig. 12 shows large roughness features that formed on the unsupported surfaces in the channels. Especially visible at locations (a) and (c), these roughness features are concentrated near the branch feature intended to encourage jet penetration. Just as for the \( \lambda = 0.1 \) \( L \) case, this small feature was not built successfully because it was unsupported throughout the build.
However, the general geometric trends seen in Fig. 7 were largely reproduced in the as-built test coupons, both qualitatively and quantitatively. In the case of the \( J_1 \) objective, the L-PBF channels showed the most significant deviation from the baseline L-PBF channel, as was the intent; at all three highlighted locations, the channel bowed sharply toward the inlet. Similarly, for objectives \( J_2 \) and \( J_3 \), the main channel was to contract in area where the fluid exited the branch. Such a characteristic is seen in Fig. 12 at locations (a) and (c).

The CT scan data were also used to quantify the as-built channel dimensions. An in-house code was written to analyze 2D slices of the L-PBF coupons sweeping through the coupon in the streamwise dimension; the cross-sectional area and perimeter were calculated, which were then used to calculate a hydraulic diameter. To achieve a meaningful comparison among all baseline and optimized channels, the dimensions used in the subsequent results analyses were measured using the CT scan data at the entrance to each of the channels. As such, dimensions given in Tables 4 and 5 are for the entrance to each of the channels. For comparison, data from Kirsch and Thole [14] are included as well, denoted as “isolated.” In general, the communicating channels built to within 10% of the intended dimensions.

**Experimental Setup**

All pressure loss and heat transfer experiments were performed on a bench-top rig, using air as the working fluid; Fig. 13 shows a cross section of the facility, which has been used in numerous previous studies [1,14,21,22,40]. Flow into the rig was controlled using a commercial mass flow controller [41]. The inlet to test section was pressurized, and was held at a constant pressure over the range of Reynolds numbers of interest in this study. A needle valve downstream of the test coupon was adjusted to achieve the different Reynolds numbers, while maintaining the inlet pressure.

Static pressure taps were located upstream of the nylon inlet contraction and downstream of the nylon exit contraction to measure the pressure drop across each test coupon. The fluid velocity was calculated by using the inlet cross-sectional area calculated from the CT scans, along with the calculated fluid density and measured mass flow rate. Channel length was taken to be the length that the fluid traveled, taking into account the wavy nature of the channels.

![Diagram of flow and channel dimensions](image)

**Results and Discussion**

**Experimental Uncertainty.** Experimental uncertainty was evaluated using the methods proposed by Kline and McClintock [42]. In the flow tests, the pressure transducers used to measure the pressure drop across the coupons governed the uncertainty. For Reynolds numbers above 4000, the overall uncertainty was below 7%, while for Reynolds numbers below 4000, the overall uncertainty was below 9%. Repeatability among tests, however, was between 1.5% and 2% for all Reynolds numbers.

Calculation of the test coupon surface temperature, or the temperature of the test coupon at the solid-fluid interface, introduced the largest source of uncertainty in the heat transfer tests. The one-dimensional conduction analysis used to calculate the surface temperature takes into consideration the thickness of the copper block, the paste adhering the block to the coupon, and the coupon wall; the measurement of the thin layer of paste contributed the most to the heat transfer calculations. Nusselt number overall uncertainty, therefore, was below 6% for all coupons, with precision uncertainty below 3%.

**Pressure Loss Test Results.** Figure 14 shows friction factor augmentation for the communicating \( \lambda = 0.1 \) L case. Markers for the experimental results are labeled “AM,” while the numerical results are labeled as “k-ε.” The trend in experimental data is upward because flow through the microchannels reaches the fully rough flow regime [1,14], where friction factor no longer changes as Reynolds number increases; the baseline friction factor, \( f_0 \), however, continues to decrease with increases in Reynolds number.

Immediately apparent in Fig. 14 is the difference in magnitude in friction factor augmentation between the numerical and experimental results. This difference can be attributed to the high surface roughness in the channels, which was not modeled in the simulation. Additionally, the spread in data across the experimental results is wider than that for the numerical results.

The objectives to minimize pressure loss (\( J_1 \) objective) and to maximize heat transfer (\( J_2 \) objective) exhibited analogous performance to the numerical predictions. In the \( J_1 \) case, the shape changes sought to decrease the fluid momentum and thus, decreased the propensity for jet-like flow from the branches into the channels. Such shape changes translated well to the physical domain; the L-PBF process was able to reproduce these nuanced shape changes successfully, as evidenced by the results in Fig. 14.
Additionally, as was seen in Fig. 11, the branch feature that was designed to encourage jet penetration did not build exactly as intended. The lack of jet penetration, therefore, may have furthered the optimizer’s intent to decrease pressure loss.

Similarly in the $J_2$ case, the goal of the geometric changes was to increase the fluid velocity in the branches, thereby creating stronger jets to enter the main channels. Despite not closely replicating the intended branch geometry, the build did replicate other shape changes that sought to encourage Dean vortex formation in the main channels. Therefore, as expected, the friction factor augmentation from the $J_2$ case was higher than that from the baseline.

On the other hand, the shape changes that occurred for the $J_3$ case created weaker vortical structures than in the $J_2$ case (Fig. 6). Such a design intent, coupled with the as-built branch shape, may explain the results from the $J_3$ objective, which deviated from the expected; the friction factor augmentation for the $J_3$ objective was the lowest among all cases.

Figure 15 shows friction factor augmentation for the $\lambda = 0.4$ L case, and also includes both experimental and numerical results. A similar disparity existed between the magnitude of friction factor augmentation in the experimental and numerical results for the $\lambda = 0.4$ L case as did for the $\lambda = 0.1$ L case. Additionally, the augmentation values varied greatly across the experimental results, whereas limited variation was seen for the numerical results. These differences can be explained by the high surface roughness in the L-PBF microchannels, which was not accounted for in the simulation.

The lowest friction factor augmentation values were seen for the baseline case in Fig. 15. Even where pressure drop was to be minimized, in the $J_1$ objective, the friction factor augmentation exceeded its nonoptimized counterpart. As was seen in the “Numerical Results” section, the shape changes for the $\lambda = 0.4$ L design were considerable, especially in the branch itself (Fig. 7). As previously mentioned, in the case of the $J_1$ objective, the arch of the branch shape in the aft direction lowered the fluid momentum, thereby causing lower fluid velocity downstream (Fig. 8); the arch was reproduced fairly well in the as-built channels (Fig. 12).

![Table 4 Inlet dimensions of the CAD and as-built channels for $\lambda = 0.1$ L](image)

<table>
<thead>
<tr>
<th></th>
<th>Communicating</th>
<th>Isolated [14]</th>
<th>CAD</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_c$ (mm$^2$)</td>
<td>$A_c$ (mm$^2$)</td>
<td>$A_c$ (mm$^2$)</td>
<td>$A_c$ (mm$^2$)</td>
</tr>
<tr>
<td>$D_h$ (mm)</td>
<td>$D_h$ (mm)</td>
<td>$D_h$ (mm)</td>
<td>$D_h$ (mm)</td>
</tr>
<tr>
<td>Baseline</td>
<td>0.5</td>
<td>0.63</td>
<td>0.52</td>
</tr>
<tr>
<td>$J_1$ [min($\Delta P$)]</td>
<td>0.48</td>
<td>0.63</td>
<td>0.43</td>
</tr>
<tr>
<td>$J_2$ [max($Q$)]</td>
<td>0.5</td>
<td>0.64</td>
<td>0.44</td>
</tr>
<tr>
<td>$J_3$ [max($Q/\Delta P^{1/3}$)]</td>
<td>0.46</td>
<td>0.62</td>
<td>0.44</td>
</tr>
</tbody>
</table>

![Table 5 Inlet dimensions of the CAD and as-built channels for $\lambda = 0.4$ L](image)

<table>
<thead>
<tr>
<th></th>
<th>Communicating</th>
<th>Isolated [14]</th>
<th>CAD</th>
</tr>
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<td>0.52</td>
</tr>
<tr>
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<td>0.69</td>
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</tr>
<tr>
<td>$J_2$ [max($Q$)]</td>
<td>0.55</td>
<td>0.69</td>
<td>0.44</td>
</tr>
<tr>
<td>$J_3$ [max($Q/\Delta P^{1/3}$)]</td>
<td>0.52</td>
<td>0.68</td>
<td>0.44</td>
</tr>
</tbody>
</table>

![Fig. 13 Experimental test rig for flow and heat transfer measurements](image)

Additionally, as was seen in Fig. 11, the branch feature that was designed to encourage jet penetration did not build exactly as intended. The lack of jet penetration, therefore, may have furthered the optimizer’s intent to decrease pressure loss.

Similarly in the $J_2$ case, the goal of the geometric changes was to increase the fluid velocity in the branches, thereby creating stronger jets to enter the main channels. Despite not closely replicating the intended branch geometry, the build did replicate other shape changes that sought to encourage Dean vortex formation in the main channels. Therefore, as expected, the friction factor augmentation from the $J_2$ case was higher than that from the baseline.
However, such a shape change also resulted in stronger recirculation zones in the branch itself, specifically near the top and bottom of the channel. Figure 16 shows a top-down view at 25% channel height of the baseline and $J_1$ configurations. The view location is the same as in Fig. 7(a). Secondary velocity vectors are laid atop normalized axial velocity contours to highlight the direction of the flow. For the $J_1$ objective, the direction of the vectors moving through the branch is noticeably different than the direction of the vectors in the baseline design. Given the friction factor results in Fig. 15, this recirculation zone is hypothesized to have negatively affected the frictional losses through the coupon. In L-PBF microchannels, where roughness plays a large role in the flow development, the negative effects of this recirculation zone may have been amplified and thus, yielded a higher friction factor augmentation than the baseline.

As predicted, both the $J_2$ and $J_3$ objectives also exhibited a higher friction factor augmentation than the baseline design, as seen in Fig. 15. The shape changes that generated higher velocity in the branches were successfully reproduced in the L-PBF test coupon. The branch shapes, especially, were replicated faithfully (Fig. 12) and the roughness features most likely enhanced the optimizer’s intent.

Heat Transfer Results. Much like the friction factor results, heat transfer results will be presented in the form of an augmentation over a smooth baseline channel. Experimental and numerical heat transfer results for the $\lambda = 0.1$ L case are shown in Fig. 17. The trend in Fig. 17 is a downward slope, indicating that the wavy channels exhibit the best performance at lower Reynolds numbers ($\text{Re} < 5000$), which is consistent with the results from Ref. [1].

Heat transfer performance mimics the friction factor results; the objective to maximize heat transfer ($J_2$) yielded the highest heat transfer augmentation, as was predicted by the simulation. Similarly, the objective to minimize pressure loss ($J_1$) yielded the lowest augmentation values.

Over the Reynolds number range tested, the $J_2$ objective showed about a 10% increase in heat transfer augmentation over the baseline design, which was slightly higher than expected based on the simulation’s prediction (Table 1). Given that the shape of the $J_2$ microchannels was meant to promote jet formation and encourage Dean vortex development, the addition of surface roughness most likely augmented these goals.

A notable result from Fig. 17 is the heat transfer performance for the $J_3$ case relative to the $J_1$ case. Figure 14 showed that the $J_3$ case exhibited lower friction factor augmentation than the $J_1$ case, which deviated from the expectation, but Fig. 17 shows similar heat transfer performance between the two geometries. These results suggest that the optimized geometry achieved its goals, at least partially: an attempt to maintain good heat transfer performance did so without a substantial increase in pressure loss.

Figure 18 presents heat transfer augmentation values for the $\lambda = 0.4$ L case, and contains both experimental and numerical results. The heat transfer augmentation for all four L-PBF test coupons was similar; the $J_1$ objective yielded the lowest heat transfer augmentation, which was to be expected, but the two objectives with heat transfer as the observable yielded results nearly on top of the baseline results. Despite the higher friction factor augmentation seen for the $J_2$ and $J_3$ cases in Fig. 15, the heat transfer performance appeared to be unaffected. The features that were successfully reproduced in the build served less to increase the heat transfer than they did to augment the frictional losses.

Communicating Versus Noncommunicating Channel Performance. To gain a better understanding of how the addition of the branches in wavy microchannels affected performance, we can directly compare the friction factor and heat transfer augmentation between the communicating and noncommunicating (isolated) channels from Kirsch and Thole [14]. Figure 19 shows the friction factor augmentation versus Nusselt number augmentation at a Reynolds number of 5000 for the communicating and non-communicating channels.

The first element of Fig. 19 that stands out is the notably high friction factor augmentation shown by the $\lambda = 0.1$ L wavy channel with the objective to maximize heat transfer. By contrast, the $\lambda = 0.1$ L communicating case to maximize heat transfer shows a much decreased friction factor augmentation, with a comparable heat transfer augmentation. In the case of the noncommunicating channels, the objective to maximize heat transfer was achieved in the simulation by creating large vortical structures in the channels [14]. Given the high surface roughness in the as-manufactured channels, this intended flow feature was enhanced, and, as a consequence, a significant increase in the frictional losses was measured. On the other hand, the objective to maximize heat transfer in the communicating case revolved around encouraging jet penetration from the branches into neighboring channels. While such jet penetration may have been tempered given the results of the as-built channels (Fig. 11), this design intent incurred less of a penalty to the friction factor, and was less affected by the roughness in channels.

In a similar vein, the objective to minimize pressure loss for the communicating and noncommunicating channels yielded...
The current study was created to push the limits of the additive manufacturing process. The baseline microchannel configurations borrowed a key element from many transport systems found in nature, namely secondary branches that connect primary channels; two different wavelengths of a communicating wavy channel design were chosen for a baseline configuration. Each was numerically optimized with three different goals to (1) minimize pressure loss, (2) maximize the heat transfer, and (3) maximize the ratio of heat transfer to pressure loss. The resultant geometries were complex, with cross-sectional shapes that were far from rectangular.

Each of the objectives functions was achieved numerically, with the majority of shape changes occurring near the branches themselves. Where pressure loss was to be minimized, the branches morphed so that the fluid exited the branches at a shallower angle into its neighboring main channel. Other shape changes in the objective to minimize pressure loss sought to decrease the fluid momentum, which lessened the formation of Dean vortices in the channel. On the other hand, where heat transfer was to be maximized, the shape of the branches encouraged jet penetration into neighboring channels, thereby enhancing the formation of Dean vortices and increasing the fluid mixing.

The numerically optimized channels were duplicated to fit in test coupons, and were additively manufactured using a laser powder bed fusion process. The as-built test coupons stayed relatively true to their design intents. However, a natural consequence of the additive manufacturing process is the surface roughness; the presence of large roughness features, especially on any unsupported surfaces, most assuredly affected the flow through the channels, and also influenced how well the design of the branch exit was replicated.

Experimental results showed that, in the shorter wavelength configuration, the objective to minimize pressure loss was achieved. The optimizer’s goal to decrease the fluid momentum translated well to the as-built microchannels, with the friction factor augmentation of the optimized channels around 6% less than the baseline. Similarly, the objective to maximize heat transfer was achieved as well, with an increase in the heat transfer augmentation from the optimized channels at around 9% over the baseline.

In the longer wavelength channels, the presence of the branches did not strongly influence the channels’ heat transfer performance. While objectives to maximize heat transfer resulted in an increase in the pressure loss, heat transfer remained unaffected. The effects of surface roughness may have outweighed the effects of the shape changes surrounding the branches due to the fact that the branch additions were sparse.

This study represents only an initial foray into the relationship between design tools and additive manufacturing. While additive manufacturing lifts many design constraints imposed by conventional means, the manufacturing process is less than perfect. Different concerns, such as build direction and surface roughness, must be taken into consideration. Until a tool that more closely integrates design decisions and build consequences exists, the numerical tools currently available for optimization methods should be explored. Additionally, detailed investigation into the current capabilities of the additive manufacturing process is critical in furthering the technology. With continued research into both the design tools and the manufacturing technology, the full potential of additive manufacturing can be realized, and more effective microchannel cooling configurations can be created.
Acknowledgment

The authors would like to thank Corey Dickman and Griffin Jones at Penn State’s CIMP-3D laboratory for their efforts in manufacturing all test coupons, as well as Jacob Snyder in the Penn State START laboratory for CT scanning all test pieces.

Funding Data

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Nomenclature

$D_h$ = hydraulic diameter, $A \cdot \rho \cdot f^{-1}$
$f$ = Darcy friction factor
$h$ = convective heat transfer coefficient
$\delta$ = channel height
$I$ = objective function
$k$ = thermal conductivity
$L$ = coupon length
$Nu$ = Nusselt number, $h \cdot D_h \cdot k^{-1}$
$P$ = static pressure
$Q$ = heat transfer rate
$R$ = governing equations
$Ra$ = surface roughness, $\sum_{i=1}^{N} |z_{surf} - z_{mean}|$
$Re$ = Reynolds number, $U \cdot D_h \cdot \nu^{-1}$
$T$ = static temperature
$T_i$ = inlet temperature
$T_s$ = surface temperature
$U$ = axial velocity
$\bar{u}$ = flow field state vector
$\bar{u}_{in}$ = mean axial velocity
$y^+$ = inner wall coordinate, $y^+ = y \cdot u_{\tau} / \nu$

Greek Symbols

$\Delta$ = differential
$\lambda$ = wavelength
$\theta$ = nondimensional temperature $(T - T_s)/(T_i - T_s)$

References


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