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A Permeable-Membrane Microchannel Heat Sink made by Additive Manufacturing

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Abstract

Microchannel heat sinks are capable of removing dense heat loads from high-power electronic devices with low thermal resistance, but suffer from high pressure drops due to the small channel dimensions. Features that reduce the pressure drop, such as manifolds, increase fabrication complexity and are constrained by traditional subtractive manufacturing approaches. Additive manufacturing technologies offer improved design freedom and reduced geometric restrictions, expanding the types of features that can be produced and integrated into a heat sink. In this work, a novel permeable membrane microchannel (PMM) heat sink geometry is proposed and fabricated using direct metal laser sintering (DMLS) of an aluminum alloy (AlSi10Mg). In this PMM design, the cooling fluid is forced through thin, porous walls that act as both conducting fins and membranes that allow flow through their fine internal flow features for efficient heat exchange. The design leverages the ability of this fabrication process to incorporate complex, arbitrarily curved structures having internal porosity to enhance heat transfer and reduce pressure drop across the heat sink. The PMM heat sink geometry is benchmarked against a low-pressure-drop manifold microchannel (MMC) heat sink. A reduced-order model is used to explore the relative performance trends between the designs. Both heat sinks are experimentally characterized at flow rates of 50-500 mL/min using deionized water as the working fluid. At a constant pumping power of 0.018 W, the permeable membrane microchannel design offers both lower thermal resistance (17% reduction) and lower pressure drop (28% reduction) compared to the manifold microchannel heat sink.

Keywords: additive manufacturing, electronics cooling, microchannel heat sink, permeable membrane, porous media

Nomenclature

A	area	u_{sup}	superficial velocity
C	coefficient in Eq. (2)	x^+	dimensionless entry length
c_p	specific heat capacity	<i>Greek symbols</i>	
D_H	hydraulic diameter	K	permeability
D_p	particle diameter	A	aspect ratio
f_F	Fanning friction factor	μ	dynamic viscosity
G	mass flux	ρ	density
H	heat transfer coefficient	Φ	porosity
K_∞	incremental pressure drop number	α	aspect ratio
k	thermal conductivity	<i>Subscripts</i>	
L	length	app	apparent, accounting for developing flow
Nu	Nusselt number, hD/k_l	$base$	evaluated at the heat sink base
Pr	Prandtl number, $c_p\mu/k_l$	ch	channel
ΔP	pressure drop	dev	developing flow
Q_{in}	power input	eff	effective
R	thermal resistance	f	evaluated for the fluid
Re	Reynolds number, GD/μ	s	evaluated for the solid material
T	temperature	sf	solid-fluid interface
t_{wall}	wall/membrane thickness	tot	total

1. Introduction

The pursuit of higher power and more compact electronics in aerospace, automotive, and other applications requires complementary thermal management technologies that can effectively remove large amounts of heat within a small envelope. Microchannel heat sinks are one option for high-heat-flux cooling, and have been extensively studied for a range of working fluids in both single- and two-phase operation [1–3]. While straight channels of rectangular cross section are the

most common, other cross-sectional shapes (*e.g.*, circular, triangular, trapezoidal) have also been investigated [4,5]. Some studies have considered channels with a non-linear wall profile both numerically [6] and experimentally [7]; wavy channels can increase heat transfer at the cost of a higher pressure drop. Indeed, the high pressure drop associated with flow through the microchannel heat sinks is a primary drawback, and numerous design concepts have been proposed to address this issue.

One of the more effective methods of reducing the pressure drop across a microchannel heat sinks is through the addition of a manifold layer that shortens the flow length through the microchannels [8]. Experimental studies of such manifold microchannel (MMC) heat sinks have shown heat flux dissipation in excess of 1 kW/cm^2 at moderate pressure drops [9,10]. The geometry of MMC heat sinks has been optimized for different performance objectives [11,12], with the optimized designs improving surface temperature uniformity and reducing the thermal resistance of the heat sink at a fixed pumping power compared to a standard microchannel heat sink without a manifold [13].

Aside from the addition of a manifold, attempts to improve the performance of microchannel heat sinks have included the incorporation of porous features. The introduction of a porous medium that occupies the entire microchannel cross-section has been shown to provide excellent heat transfer performance [14], albeit at the cost of a drastically increased pumping power [15]. Hung *et al.* [16] numerically investigated several alternative arrangements of the porous medium within microchannels and found that a porous layer on the walls of the channel offered the best balance between increased thermal performance and higher pressure drop. Chuan *et al.* [17] simulated the performance of a straight microchannel design utilizing porous fins between the channels instead of the standard solid walls; a slight increase in thermal resistance was offset by a significant reduction in the pressure drop. This decrease in pressure drop was attributed to the effectively non-zero ‘slip’ velocity at the wall of the porous fin. Lu *et al.* [18] extended this idea to consider wavy channels; in addition to the pressure drop reduction offered by the porous fins, the wavy channels reduced the thermal resistance as a result of a longer effective flow length, mixing due to vortices, and forced permeation of a portion of the fluid through the fins. However, while these numerical modeling efforts indicate the potential improvement from these increasingly complex designs, fabrication of such heat sinks via conventional subtractive techniques (*e.g.*, micromachining, anisotropic chemical etching) is difficult, if not impossible. The complexity of

structures with internal porosity has been limited to features that can be produced by sintering particles in a mold.

Advances in additive manufacturing (AM) technologies have recently made it possible to more readily fabricate complex geometries, though there has been little focus to date on leveraging these fabrication capabilities to enhance the performance of microchannel heat sinks for electronics cooling. Studies of microscale heat exchangers made by additive manufacturing, specifically powder bed fusion processes, frequently highlight issues associated with material properties and high surface roughness. The authors' previous work [19] demonstrated AM fabrication of straight microchannel and manifold microchannel heat sinks in an aluminum alloy having channel hydraulic diameters of 500 μm and monolithic construction. The pressure drop was well-predicted by conventional hydrodynamic theory, with a roughness-induced early transition to turbulence at low Reynolds numbers ($\text{Re} < 800$). The thermal performance was overpredicted, which was attributed to uncertainty in the thermal conductivity of the material. Kirsch and Thole [20,21] experimentally tested additively manufactured wavy microchannels based on numerically optimized designs. Their results showed that the wall roughness introduced by AM processes assisted in augmenting the heat transfer, while also contributing to an increase in pressure drop. Designs optimized for reduced pressure drop did not meet the performance expectations due to the added roughness, but design optimization targeted at both pressure drop reduction and heat transfer augmentation yielded improved performance compared to the baseline wavy channels of rectangular cross-sections. Pin fin heat exchangers have also been studied experimentally [22–24], and the geometric print fidelity and surface roughness were shown to have a large effect on performance; accurate production of sharp-edged solid features below 0.5 mm could not be achieved. Arie *et al.* [25] investigated an air-water heat exchanger fabricated with multiple metal alloys using DMLS; despite significant fabrication inaccuracies, the devices led to increases in gravimetric heat transfer density. Polymer heat exchangers fabricated using additive manufacturing technologies have also been studied [26,27], due to their favorable chemical and corrosion resistance, as well as light weight.

The fabrication of additively produced porous media, primarily non-stochastic lattice structures, has been studied but the focus has not been heat exchange applications. These structures can be produced with powder bed fusion processes with different metals at porosities between 30–90% [24,28]. In addition to heat exchangers, these structures are desirable in filtration applications

[29]. The literature regarding the intentional introduction of stochastic porosity within parts fabricated with powder bed fusion processes is sparse, as this is generally an undesired result and significant effort has been directed at eliminating porosity in nominally solid parts. Nevertheless, stochastic porosities of up to 45% have been reported in aluminum and titanium alloys [30]. The porosity is generally induced by varying the process parameters, including hatch spacing (the distance between adjacent laser passes) and scanning speed [31].

While the study of additively manufactured microscale heat exchangers is relatively new, there have been a few demonstrations of the novel and complex heat exchanger designs that can be enabled. Dede *et al.* [32] used topological optimization to generate a heat sink geometry for an air-cooled jet impingement application that was then produced using powder bed fusion in an aluminum alloy. The additively produced design was compared to several conventional designs, achieving an improved coefficient of performance even when compared to heat sinks made of a higher thermal conductivity material. Robinson *et al.* [33] utilized an electrochemical fabrication additive process to demonstrate the fabrication of a hybrid heat sink geometry that incorporates both jet impingement and microchannel flows; the designed geometry addressed several concerns normally associated with jet arrays such as wall jet formation, cross-flow, and flow distribution. The design was studied numerically, and the performance was shown to compare favorably against other selected compact, high-performance heat exchangers.

The current study proposes and experimentally evaluates a novel permeable membrane microchannel (PMM) heat sink design concept. In this design, all of the fluid entering the heat sink is forced through thin, porous walls; the porous walls act as both conducting fins and membranes with fine internal flow features that allow throughflow for efficient heat exchange. The design exploits the capabilities of direct metal laser sintering to produce complex and thin porous features to overcome the pressure drop challenges associated with using porous materials for heat exchange. A reduced-order model is developed to compare the performance trends against a benchmark manifold microchannel design. The heat sinks are fabricated and experimentally characterized to quantify the extent of the hydraulic and thermal performance improvement achieved.

2. Permeable Membrane Microchannel Heat Sink

The convective heat transfer coefficient in a channel increases as its hydraulic diameter decreases; this inverse scaling is the fundamental driver for using microscale channels in a heat sink. This effect can also be achieved in flow through an open-celled microporous medium in which the effective hydraulic diameter is just the pore size. Porous media can generally achieve smaller hydraulic diameters and higher internal surface area-to-volume ratios than straight microchannels in a heat sink, at the cost of a significantly higher pressure drop. To minimize this pressure drop penalty, the porous layer thickness should be as small as possible to reduce the flow length through the narrow pore paths. The frontal area should also be maximized to reduce the flow rate through any one pore path. These principles guide the concept development of our permeable membrane microchannel (PMM) heat sink design.

As discussed in Section 1, reduction of the pressure drop in these microchannel heat sinks is a central goal. Manifold microchannel (MMC) heat sinks, which use a manifold design with multiple flow inlets and outlets to reduce the effective flow length through the microchannels, serve as a good benchmark. Figure 1(a,c,e) shows a top-down unit cell schematic and isometric views of a manifold microchannel heat sink design. This MMC design consists of a layer of straight microchannels that are capped with a manifold layer containing inlet-outlet pairs that distribute the flow across the entire bank of microchannels. The working fluid travels along the manifold inlet, down into the microchannel layer below, across a short flow length in the microchannels, and then exits via the outlet channel.

The permeable membrane microchannel heat sink design comprises a bank of thin porous ‘membranes’ separated by small channels (see the top-down unit cell schematic and isometric views in Figure 1(b,d,f)). These permeable membranes act both as fins that conduct heat up from the heat sink base and as the primary heat exchange surface (within the membrane pores). In the PMM design, the fluid enters the heat sink and travels along an inlet channel. Solid endcaps force the fluid across the thin permeable membrane before it collects in the neighboring outlet channel. Non-linear horizontal and vertical membrane profiles are incorporated that increase the area of the membrane front face compared to a flat design, so as to reduce the pressure drop and increase the heat exchange area.

Subtractive and other conventional machining processes would not be able to readily produce the complex geometry with locally porous features shown in Figure 1d, necessitating

additive manufacturing approaches for fabricating the PMM design. For production of microchannel heat sinks, accurate fabrication of sub-millimeter features in a high-thermal-conductivity metal is necessary. Direct metal laser sintering (DMLS) is a commercially mature, widely available technology that is suitable for producing microchannel heat sinks with features on the order of 100s of μm . DMLS is a powder bed fusion technology that uses a laser to selectively fuse a thin layer of metal powder to create a cross-section of the desired part. After fusion, another thin layer of powder is deposited on top and the process repeats; in this way, parts build up layer by layer. Based on our previous work [19] and the literature [24], a microchannel width of 500 μm is near the lower limit of what can be commercially fabricated; even though laser spot sizes of 50-100 μm are common, the significant heat-affected zone prevents finer widths from being fabricated. Therefore, the membrane pores in the PMM cannot be directly printed; rather, the material must be rendered porous during the powder fusion process to achieve these small pore features. A number of alloys are available for use with DMLS; the current work considers AlSi10Mg, an aluminum alloy with a nominal conductivity of $k_s \approx 110 \text{ W/m-K}$ [34].

3. Design and Characterization Methods

3.1 Reduced-Order Heat Sink Modeling

To evaluate the relative performance of the PMM and MMC designs, a reduced-order model is developed. The model is used to study performance trends of the PMM design as a function of the membrane characteristics and provide an assessment of this new heat sink design relative to an MMC design.

The pressure drop across the heat sinks is assumed to occur primarily across the smallest hydraulic diameter features used for heat exchange (*viz.*, the microchannels in the MMC design and the porous membrane in the PMM design) and along the length of the outlet channel. The total pressure drop across the heat sinks is the sum of the individual pressure drops across these two features. The pressure drop in the inlet channel will be lower than in the outlet due to pressure recovery by fluid discharge from the inlet and, in the manifold design, the smaller hydraulic diameter of the outlet. When the fluid passes through the outlet channel, there can be the opposite effect, as the acceleration of the fluid exacerbates the pressure drop. For the outlet channel pressure

drop, in both the MMC and PMM designs, a conservative estimate is to assume that all of the flow travels along the entire outlet channel length:

$$\Delta P_{ch} = \frac{2f_F L_{ch} G_{ch}^2}{\rho_f D_H} \quad (1)$$

where f_F is the Fanning friction factor for fully developed laminar flow in rectangular channels [35] and is given by

$$f_F = \left(\frac{24}{Re}\right) (1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5). \quad (2)$$

The pressure drop in the microchannels for the MMC design can be calculated using Equation (1) using the fractional flow rate that goes through any one microchannel and the effective flow length between the inlet and outlet. For simplicity, the flow is assumed to be equally distributed among the channels. The pressure drop across the membrane in the PMM design can be calculated using Darcy's Law

$$\Delta P_{wall} = \frac{u_{sup} \mu_f t_{wall}}{K} \quad (3)$$

where $u_{sup} = \dot{V}/A$ is the superficial velocity within the porous medium. The permeability K of the membrane is estimated using the Carman-Kozeny equation for flow in packed beds of spherical particles

$$K = \frac{D_p^2 \phi^3}{150(1-\phi)^2} \quad (4)$$

It is assumed that all heat transfer to the fluid occurs within the microchannel layer in the MMC design and within the membrane in the PMM design. For the microchannels, the Nusselt number is calculated assuming thermally developing flow as [36]

$$Nu_{dev} = 1.86 \left(\frac{Re Pr D_H}{L} \right)^{\frac{1}{3}} \left(\frac{\mu_f}{\mu_{sf}} \right)^{0.14} \quad (5)$$

The heat transfer surface area of the microchannels is trivially calculated from the given channel geometry. For the membrane, the pore Nusselt number is obtained from a particle-diameter-dependent correlation [37]:

$$Nu_{sf} = 2.081 + 0.296 \left(Re_{D_p}^{0.6} Pr^{\frac{1}{3}} \right)^{1.2} \quad (6)$$

where the particle diameter is taken as the powder clump size in the fabricated membrane. The internal solid-fluid interfacial area of the membrane is estimated for cubic close-packed spherical particles as

$$A_{sf} = \frac{6(1-\phi)V_{tot}}{D_p} \quad (7)$$

Since there is convection both within the pore space and on the exterior surface of the membrane, a Nusselt number for the channel must also be considered. Similar to the friction factor, this value is calculated assuming fully developed laminar flow as [38]

$$Nu_{ch} = 8.235 \left(1 - \frac{1.883}{\alpha} + \frac{3.767}{\alpha^2} - \frac{5.814}{\alpha^3} + \frac{5.361}{\alpha^4} - \frac{2}{\alpha^5} \right). \quad (8)$$

Due to the height of the microchannel fins and the relatively low effective conductivity of the permeable membranes, it is important to account for the fin efficiency of these features. The fin efficiency of the microchannel walls can be calculated using the nominal thermal conductivity of the solid printed aluminum. For the porous membrane, the effective thermal conductivity k_{eff} is calculated in accordance with the effective medium theory model [39], and is given as

$$k_{eff} = \frac{1}{4} \left((3\phi - 1)k_s + (3(1 - \phi) - 1)k_l + \sqrt{((3\phi - 1)k_s + (3(1 - \phi) - 1)k_l)^2 + 8k_l k_s} \right) \quad (9)$$

The efficiency for straight rectangular fins with an adiabatic tip is

$$\eta_f = \frac{\tanh(mL)}{mL} \quad (10)$$

where $m = \sqrt{\frac{h_{dev}P}{k_s A_c}}$ for the solid fins in the manifold heat sink design and $m = \sqrt{\frac{h_{sf}}{k_{eff}} \left(\frac{6(1-\phi)}{D_p} \right)}$ in the permeable membrane design. The thermal resistance of the manifold microchannels and permeable fin arrays can be calculated respectively as

$$R_{MMC} = \frac{1}{h_{dev}(A_{base} + (N\eta A)_{fin})} \quad (11)$$

$$R_{PMM} = \frac{1}{h_{ch}A_{base} + h_{sf}A_{sf}N_{fin}\eta_{fin}} \quad (12)$$

where Eq. (12) accounts for the different heat transfer coefficients within the pore space and on the exterior of the membrane.

3.2 Experimental Facility and Procedures

A flow loop identical to that described in Ref. [19] was used to experimentally characterize the thermal and hydraulic performance of the heat sinks. The flow loop uses deionized water as the working fluid, imposes controlled, constant boundary conditions on the heat sink samples, and enables measurement of the flow rate, fluid temperature, heat sink temperature, pressure drop, and power input; the key components are briefly summarized here. The system is a closed loop and a gear pump is used to circulate the working fluid. The flow rate is measured and the fluid is filtered and preheated before entering a test section that holds the heat sink. A 200 W ceramic heater provides adjustable heat input to the base of the heat sink being tested. After exiting the heat sink, the fluid is cooled back to ambient temperature and returned to a flexible reservoir that is maintained at atmospheric pressure.

The test section presses the heat sink onto the heater, positions thermocouples for temperature measurements, and contains pressure taps to measure the pressure drop; slight modifications were made compared to Ref. [19]. Due to the lack of an incorporated lid on both heat sinks due to a desire to visualize the heat transfer features, a silicone rubber gasket was used to seal the interface between the heat sink and a polycarbonate block that has features to route flow into and out of the heat sink. The inlet temperature of the working fluid was maintained at 30 °C.

Prior to testing, the heat sinks were cleaned with compressed air and inserted into the test section. The experimental heat loss is measured by assembling the test section and applying power in the absence of the working fluid. Upon reaching a steady temperature at each power, the base heat sink temperature is recorded. A best-fit line, assuming a zero intercept, is fitted to these measurements to yield an empirical correlation and allow for conservative estimation of the temperature-dependent heat loss based on the base temperature of the heat sink. The range of heat loss in this study is 2.8% to 4.1%.

To characterize the hydraulic performance of the heat sinks, the flow rate through the unheated test section was varied over the range from 50 mL/min to 500 mL/min in 50 mL/min increments. After achieving steady conditions at each flow rate, the pressure drop across the heat sink is measured. These tests were used to identify the flow rates at which the two heat sink designs had the same pumping power so that thermal performance comparisons could be made at constant pumping power. Two nominal pumping powers of 0.008 W and 0.018 W were chosen for the thermal performance characterization.

At each of the two pumping powers, the heat input power to the heat sink is incremented from 0 W to 200 W in steps of 20 W. At each step, the system is allowed to reach steady state and then data are recorded for 60 s; a single time-averaged value is reported for each measurement. The flow is considered steady when the time-averaged fluctuations in pressure drop are less than 50 Pa and less than 0.1 °C in temperature. The thermal performance is characterized by the total thermal resistance of the heat sinks

$$R_{tot} = \frac{T_{base} - T_{f,in}}{Q_{in}} \quad (13)$$

which can be calculated directly from the measured temperatures at the center of the heat sink base and the fluid inlet temperature, as well as the loss-adjusted heat input. To compare to the reduced-order model, the conduction resistance through the base and the caloric resistance of the fluid are subtracted from the total resistance, leaving only the resistance of the finned arrays. These conduction and caloric resistances are respectively calculated as

$$R_{cond} = \frac{t_{base}}{k_s A_c} \quad (14)$$

$$R_{cal} = \frac{T_{f,avg} - T_{f,in}}{Q_{in}} \quad (15)$$

For a given heat sink geometry and flow rate, the thermal resistance is expected to be constant with power input during single-phase operation; changes in heat flux translate to proportional changes in the streamwise temperature gradient within the fluid and the local temperature difference between the convection surface and the bulk fluid. Due to the near-constant values of thermal resistance measured across the range of power inputs, the thermal resistance is reported as an arithmetic mean of all test points from 0 W to 200 W for a given heat sink and flow rate.

The sensor uncertainties specified by the manufacturers are listed in Table 1. The uncertainty in calculated thermal resistance is also listed, and was determined using a sequential perturbation method [40]. The uncertainty in thermal resistance is highest at the lower flow rates due to the smaller temperature difference between the heat sink base and the working fluid.

4. Results

4.1 Heat Sink Geometry

A manifold microchannel heat sink (Figure 1e) is used as the benchmark for comparison against the permeable membrane microchannel heat sink design concept. The MMC design is held fixed, using the minimum possible feature sizes based on the fabrication limits for the geometry. The manifold layer is 1.5 mm tall and has 1.0 mm–thick solid wall features, with inlet and outlet channel widths of 1.5 and 0.5 mm, respectively. The inlet-to-microchannel area ratio is thus a factor of 9, supporting the reduced-order model assumption that the inlet pressure drop will be much smaller than the microchannel pressure drop. The outlet, at 3 times larger than the microchannel, cannot be neglected and its pressure drop is thus approximated in the model. These manifold channel widths are in accordance with several design optimizations performed in the literature that suggested an ideal single-phase inlet-to-outlet width ratio of 3:1 [11,12]. The effective flow length, from inlet to outlet through the microchannels, is thus 2.00 mm. The total footprint of the microchannel layer is 15.0 mm $\times$ 15.5 mm and is covered by sixteen rectangular microchannels of 0.5 mm width and 2.0 mm height, spaced by 0.5 mm-wide solid fins. The base thickness between the bottom of the heat sink and the bottom of the microchannel layer is 1.0 mm; while smaller base thicknesses are possible based on the fabrication capabilities, this thickness was chosen to eliminate any potential for porosity-induced leakage of fluid through the base. A 250 μ m deep, 1000 μ m wide groove runs from one edge of the heat sink to the center, allowing for placement of a thermocouple to measure the base temperature.

The permeable membrane microchannel heat sink design (Figure 1f) uses membranes that are 2.0 mm tall and cover a 15.0 mm $\times$ 15.5 mm footprint, identical to the microchannel layer envelope in the MMC design (note that the PMM design is more compact compared to the MMC design since it does not need a manifold layer). The permeable membranes have a curved profile in the horizontal plane, with an amplitude of 0.5 mm and a wavelength of 25% of the channel length. The vertical profile (normal to the heat sink base) is that of a triangular chevron with an amplitude equal to the width of the fin. The specific membrane profiles are heuristically chosen for this demonstration; however, the shape of the membrane offers a design variable that is only limited by the capabilities of the fabrication process. The inlet and outlet channels have widths of 0.6 mm. The solid endcaps at the ends of the channels are 0.5 mm thick. The base thickness is identical to the manifold design and also contains a thermocouple groove.

In the PMM design, the membrane pore characteristics and thickness that can be successfully fabricated with the additive process are not known *a priori*. Nevertheless, the reduced-order model assumes that the pores will be significantly smaller than the channels, and thus will dictate the pressure drop. The following sections evaluate the range of membrane characteristics that can be fabricated via additive manufacturing (Section 4.2), input a range of membrane characteristics into the reduced-order model to identify the PMM design space (Section 4.3), and experimentally evaluate one promising design (Section 4.4).

4.2 Membrane Fabrication

Fabrication processing parameters for direct metal laser sintering to achieve a specified induced porosity in AlSi10Mg are not commonly available. A set of process-tuning sample cubes was designed and fabricated to determine the membrane thickness that could be achieved at different bulk sample porosities. To this end, ten samples were fabricated in collaboration with a commercial vendor (EOS M280; GPI Prototype & Manufacturing Services), each with different laser and scanning parameters. The geometry of the sample cubes and a photograph of one fabricated part are shown in Figure 2. The sample cube has a solid base layer, a porous core layer in the center, and a solid top layer. The porous core layer is used to assess the nominal bulk porosity that is achieved at the given processing parameters. A series of six fins of differing widths ranging from 150 μm to 500 μm are built on the solid top surface. These fins have the same chevron profile as desired for the heat sink, with a fin height of 1.0 mm and a wavelength of 5.0 mm. Across the set of ten sample cubes, porosities between 12-23% were achieved, as determined based on mass and volume measurements. The thinnest fins below 300 μm in width failed to build on all samples. The 300 μm -wide fins were successfully built when the bulk porosity was low (approximately <16%) and the 400 μm and 500 μm fins were successfully constructed on all samples (as can be seen for the 23% porosity sample in Figure 2b). As a result, the fin width chosen for the experimental demonstration is nominally 400 μm thick, the thinnest width that can be successfully fabricated at the process parameter set used for the heat sink.

In addition to optical inspection, μCT scanning (Bruker Scyscan 1272) was used to non-destructively examine the morphology of the permeable membranes. Figure 3a shows a 3D reconstruction of a 3 mm-long section of the nominally 400 μm thick membrane from the sample cube shown in Figure 2b. From the reconstruction, it is found that the actual effective thickness of

the membrane, approximately 300 μm , is below that of the nominal geometry specified during printing. This is due to the fact that, when fabricating porous features, the standard laser-scanning dimensional offsets that compensate for the heat-affected zone and melt pool size during printing of solid features are not accurate (such offsets were disabled entirely during printing of these porous parts). Additionally, while the powder used to fabricate the samples has a mean particle diameter of 45 μm , the membrane exhibits larger clumps of solid material and pores. The solid clump sizes are approximately 250 μm in diameter (the membrane is only 1-2 clump diameters thick), with membrane pores ranging between 150 μm and 400 μm in diameter. Formation of these large clumps during fabrication using aluminum powder has been attributed to a large temperature gradient across the melt pool, which leads to Marangoni convection and the “balling” of the melt line to achieve equilibrium [41]. These measured clump sizes and pore diameters are used as inputs to the reduced-order model, to evaluate the viability of the permeable membrane microchannel design compared to the benchmark manifold microchannel design, for a range of membrane porosities and thicknesses.

4.3 Model Predictions

The performance of the PMM design is evaluated relative to the MMC design based on the pressure drop ratio, $\Delta P_{PMM} / \Delta P_{MMC}$, and the fin array thermal resistance ratio, R_{PMM} / R_{MMC} . The performance ratios are compared at a constant pumping power of 0.018 W. The fluid properties are evaluated assuming a bulk fluid temperature of 30 °C and a surface temperature of 60 °C, which corresponds to a moderate power input within the range of the heater. While various performance factors can be used to assess heat sink designs, comparison of the thermal resistance at a constant pumping power is common. However, for a fair comparison it is important to ensure that the two designs also have the same order of pressure drop at this pumping power, such that they would use similar pumping technologies.

Figure 4 plots contours of the pressure drop ratio (Figure 4a) and thermal resistance ratio (Figure 4b) for ranges of membrane thickness and porosity that encompass and expand upon those achieved in the fabrication of the samples cubes. Figure 4a shows that the pressure drop ratio over a majority of the viable parameter range studied is between 0.5 and 1.5, and that the pressure ratio improves (*i.e.*, reduces) as the membrane becomes thinner and more porous. Conversely, Figure 4b shows that as the membrane gets thicker and less porous, the relative

thermal resistance of the PMM design improves. Thicker and less porous membranes increase the interfacial area and the fin efficiency, leading to low thermal resistance at the cost of a higher pressure drop. These trends also apply to a comparison at the lower pumping power. For a membrane with an effective thickness of 300 μm and a porosity of approximately 23% (within the range demonstrated for the fabricated sample cubes, Section 4.2), the model predicts a 16% reduction in the pressure drop and 24% reduction in the fin array thermal resistance for the PMM heat sink compared to the MMC design. This membrane width and porosity are used to fabricate and experimentally characterize the PMM heat sink in the next section.

4.4 Hydraulic and Thermal Heat Sink Characterization

The manifold microchannel and permeable membrane microchannel heat sink designs were both fabricated by a commercial vendor using the same aluminum alloy (AlSi10Mg) and AM process (DMLS). Images of the fabricated heat sinks are shown in Figure 5.

The measured pressure drop as a function of total flow rate is shown in Figure 6 for both heat sinks. As predicted by the reduced-order model, the pressure drop of the PMM design is decreased compared to the MMC design. The pressure drop reduction is between 20-70%, with higher reductions being achieved at the higher flow rates. The magnitude of the pressure drop (<4 kPa at 500 mL/min) is very low.

The pressure drop data from the adiabatic hydraulic testing are shown (open symbols) as a function of pumping power in Figure 7 for both heat sinks. The pressure drop data measured during the thermal testing are superimposed as filled symbols; the thermal test points which were chosen to enable comparison of the MMC and PMM thermal resistance at both constant pumping power (~ 0.008 W and ~ 0.018 W) and pressure drop (~ 2.5 kPa). The measured values of total thermal resistance are annotated in the figure next to the corresponding pressure drop data point. At a pumping power of 0.008 W, the thermal resistance of the permeable membrane microchannel design is 10% lower than the manifold microchannel design, and the pressure drop is reduced by 26%. At the higher pumping power of 0.018 W, the reduction in total thermal resistance is 17% and the reduction in pressure drop is 28%. At a constant pressure drop of 2.5 kPa, the thermal resistance of the PMM heat sink is reduced by 25% compared to the MMC heat sink. From these data, it is shown that the same total thermal resistance can be achieved with the permeable membrane design at a 56% lower pressure drop. The permeable membrane microchannel heat sink

design is unequivocally demonstrated to provide improved performance over the manifold microchannel benchmark.

The reduced-order model predictions and the experimental results compare favorably at the higher nominal pumping power of 0.018 W. The model predicts a pressure drop reduction of 16% and a decrease in the fin array thermal resistance of 24%; after subtracting the conduction and caloric resistances, the experimental data show decreases of 28% in the pressure drop and 18% in the thermal resistance. The conduction resistance through the base contributes between 7.5% and 11% of the total resistance, with the caloric resistance contributing 2.4% to 4.8%. The reduced-order nature of the model, high surface roughness of the channels and the inexact value of the nominal thermal conductivity of the additively produced aluminum are all potential factors leading to the slight differences between the model predictions and experiments.

5. Conclusions

A novel permeable membrane microchannel (PMM) heat sink design is proposed, experimentally characterized, and benchmarked against a high-performance manifold microchannel (MMC) heat sink design. In the PMM design, all of the working fluid is forced to flow through a bank of thin porous membranes separated by small channels; these membranes act both as conducting fins and have fine internal flow features to allow through-flow for efficient heat exchange. A reduced-order model is used to assess the relative pressure drop and thermal resistance for the two designs at a constant pumping power for a range of membrane thicknesses and porosities. The PMM and MMC designs were fabricated in an aluminum alloy using direct metal laser sintering. Micro computed tomography scanning was used to non-destructively characterize the stochastic porous features in the fabricated membrane; the effective membrane thickness is decreased compared to the nominal design and the solid clump sizes are significantly larger than the powder particle diameter. Experimental characterization of the heat sink designs shows that the permeable membrane microchannel design can offer a reduced thermal resistance at a constant pumping power or can match the thermal resistance at a pressure drop that is 56% lower.

The permeable membrane microchannel design demonstrates the ability of additive manufacturing to produce complex geometries incorporating locally porous features, otherwise unobtainable via conventional manufacture, to achieve heat sink performance enhancement.

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Table 1. Uncertainty in measured and calculated values.

Measured Value	Uncertainty
Pressure drop	± 0.172 kPa
Volumetric flow rate	± 5 mL/min
Base temperature	± 0.5 °C
Fluid temperature	± 1.0 °C
Voltage	$\pm < 1\%$
Calculated value	Mean uncertainty (range)
R_{tot}	2.5% (0.9 – 10.3%)

Figure 1. Top view concept diagrams of the (a) manifold microchannel heat sink design and (b) the permeable membrane microchannel heat sink design. Isometric views of the (c) manifold

microchannel and (d) permeable membrane unit cell geometry. Isometric views of (e) manifold microchannel and (f) permeable membrane heat sink geometry with the unit cells highlighted.

Note for editor: Figure 1 is sized to be one column wide.

(a)

(b)

Figure 2. (a) Geometry of the sample cubes; dimensions shown in millimeters and (b) a photograph of fabricated sample #7 (approximately 23% porosity).

Note for editor: Figure 2 is sized to be one column wide.

(a)

(b)

Figure 3. (a) Side view of a μ CT-based 3D reconstruction of a nominally 400 μ m thick permeable membrane; fluid flow direction would be normal to the page. (b) A top-view image slice from the μ CT scan data through the lower half of the membrane; the footprint of the reconstruction shown in (a) and the view orientation are marked.

Note for editor: Figure 3 is sized to be one column wide.

(a)

(b)

Figure 4. Contour plots showing the ratio of the (a) pressure drop, $\Delta P_{PMM}/\Delta P_{MMC}$, and (b) fin array thermal resistance, R_{PMM}/R_{MMC} , between the permeable membrane microchannel (PMM) and manifold microchannel (MMC) designs at a fixed pumping power of 0.018 W.

Note for editor: Figure 4 is sized to be one column wide.

Figure 5. Top-down images of the additively manufactured (a,c) manifold microchannel (MMC) and (b,d) permeable membrane microchannel (PMM) heat sinks.

Note for editor: Figure 5 is sized to be two columns wide.

Figure 6. Comparison of measured pressure drop as a function of flow rate for the manifold microchannel (MMC) and permeable membrane microchannel (PMM) heat sinks (± 0.172 kPa error bars not shown).

Note for editor: Figure 6 is sized to be one column wide.

Figure 7. Comparison of the performance of the manifold microchannel and permeable membrane heat sink designs. The dashed lines and open symbols are the pressure drop data plotted as a function of pumping power; the solid symbols and text annotations are the thermal test points and corresponding total thermal resistance.

Note for editor: Figure 7 is sized to be one-and-a-half columns wide.