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Generative Designs of Lightweight Air-Cooled Heat Exchangers

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1. Introduction

1.1. Objective

The development of high-performance air-cooled heat exchangers is required to permit the rapid growth of vehicle and aircraft electrification. In electric vehicles and airliners, the motors and power electronics are integrated into a compact space, leading to unprecedently high power density. To achieve higher overall thermal efficiency, the heat exchangers must be extremely light while maintaining their heat transfer performance and mechanical robustness. Recently advances in 3D metal printing, *e.g.*, direct metal laser sintering, and selective laser melting, have enabled the manufacturing of high-performance robust heat exchangers by eliminating thermal boundary resistance and ensuring a uniform thermal expansion coefficient. Nonetheless, many of the existing 3D-printed heat exchangers are still based on traditional designs (*e.g.* fins, channels, meshes, *etc.*), which cannot fully exploit the higher degrees of freedom for manufacturing enabled by 3D printing. To help address this issue, the program nTopology has been used in an attempt to design and simulate heat sinks that have a lightweight and robust design while still delivering high performance during steady-state convection cooling operations.

1.2. Significance

The development of effective cooling algorithms for power electronics is a key bottleneck to power densification. Among the major causes of electronics majors, overheating accounts for 55% of the failures [1]. While pumped liquid cooling can provide a higher heat transfer rate, air cooling is preferred for the thermal management of power electronics in electric vehicles and airliners due to the simplicity and lower weight of the cooling system. Moreover, air cooling is also critical to dry cooling of power plants using air-cooled condensers [2] and or encapsulated phase change materials heat exchangers [3], which have become more popular due to the recent concerns regarding freshwater withdrawal by wet cooling. Ideal air-cooled heat sinks are expected to have high heat transfer performance and low-pressure drop (and thus low power consumption for pumping). However, due to the limited design space, conventional designs of heat sinks (e.g., fins, pillar arrays) are subject to the trade-off between the heat transfer performance and the pressure drop, as illustrated in the plot of the *f* vs Nu in Figure 1. Here, the two dimensionless numbers, Nusselt number (Nu) and the Fanning friction factor (f) are used to characterize the heat transfer performance and the pressure drop, respectively. Nusselt number is defined as Nu = hD/k, where h is the convective heat transfer coefficient, k is the thermal conductivity of air, and D the characteristic length of the heat sink. The friction factor is defined as f = 2τ/(ρu2), where τ is the shear stress, ρ the density of air, and u the airspeed. Ideally, the designed heat sink will have a high Nusselt number and a low friction factor.

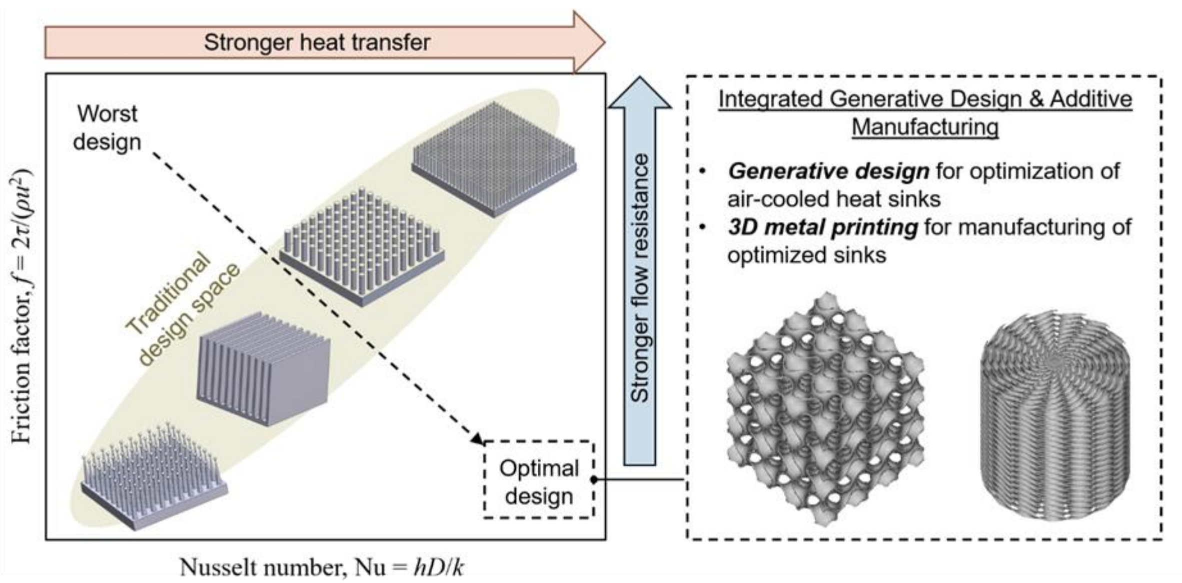


Figure 1:The trade-off between heat transfer performance and pressure drop of air-cooled heat exchangers

The governing law for all of these heat exchangers is the equation for Newton's law of cooling, $Q_{conv} = hA_s(T_s - T_\infty)$, where Q_{conv} is the rate of convection heat transfer, A_s is the surface area which heat transfer takes place, T_s is the surface temperature, and T_∞ is the ambient temperature of the working fluid. There

Heat sinks that have been designed both with bio-inspired and utilizing triply periodic minimal surface cellular structures have shown similar performance to those of conventional heat sinks[4]. With nTopology, the design space can utilize these new designs and can extend past the conventional designs.

nTopology is a program that is separate from other CAD software. Other CAD software is based on 2D drawings and making them 3 dimensional. nTopology is based on the idea of building from a 3D solid model[5]. This allows the user to utilize more complex geometries, and what nTopology calls a "field-driven approach" when it comes to designing and optimizing parts. Overall, their aim is to cut out the non-value work from engineers' workflow so that they can create the highest-performing parts in the shortest amount of time. nTopology has proven itself to be useful when trying to optimize parts or products. It has been used to design air-cooled cylinders for UAV drone engines[6] and even shoes that will be custom designed to make for a more comfortable wearing experience[7].

2. Methodology

2.1. Data Collection

All data for the heat exchangers were collected by conducting thermal simulations in nTopology. All simulation prep, material properties, and testing conditions were held constant between each analysis. Simulation prep included meshing and modeling of each heat exchanger. The material properties for the thermal analysis of the heat exchangers came from the material properties of aluminum alloy 1060 in Solidworks to compare the simulation results between Solidworks and nTopology. The other constants used for thermal simulations were heat generation, convection coefficient, and ambient temperature. Heat generation was decided on by considering what application the control heat sink would be used for. There is a wide range of wattage used by computer processors, ranging from 40 watts for average notebook computers to around 100 watts for desktop computers[8]. Since heat exchangers for desktop computers are rather large and have their own fans to force convection heat transfer, the wattage was set closer to the lower end of the range. The convection coefficient was selected from a table of ranges of convection heat transfer coefficients based on the type of convection and the working fluid in the system in table 1. This was done because the convection heat transfer coefficient is an experimentally determined parameter based on variables that impact convection heat transfer[9]. The ambient temperature was set at 295°K which is approximately 20°C or about room temperature.

Type of convection	h, W/m^2·K*
Free convection of gases	2–25
Free convection of liquids	10–1000
Forced convection of gases	25–250
Forced convection of liquids	50–20,000
Boiling and condensation	2500–100,000

*Multiply by 0.176 to convert to Btu/h·ft^2·°F.

Table 1:Typical values of convection heat transfer coeffecient[9]

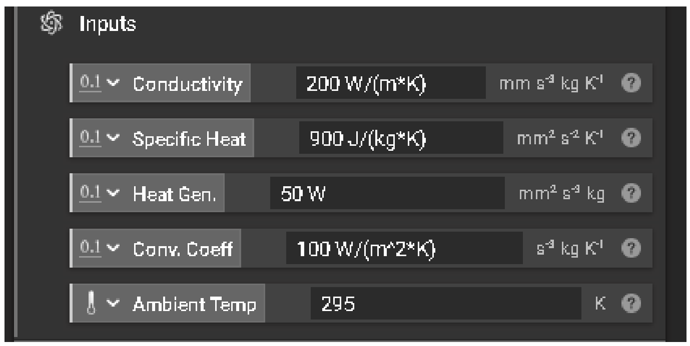


Figure 2: Inputs used for thermal simulation for all heat exchangers

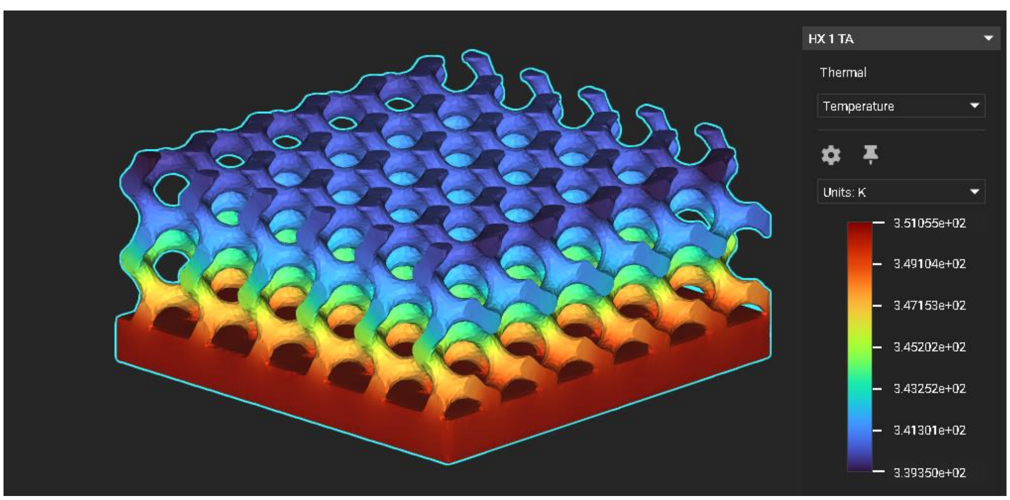


Figure 3: Heat exchanger 1 thermal analysis results

2.2. Model Architecture

All designs for the lattice heat exchangers have the same length, width, height, and base as the control heat exchanger so that the size is not a variable that would contribute to heat rejection performed by each heat exchanger. The control heat sink is 40 mm x 40 mm x 19.5 mm with a base that is 4 mm thick.

Because of how nTopology is designed, modeling objects is different than other CAD software. To make the lattice heat exchangers, start by adding a box block to your workbook for the base. In this block, the length, width, and height of the base as well as its center point will be specified. The next block to add to the workbook is walled TPMS. TPMS stands for the triply periodic minimal surface. In the walled TPMS block, a body block will need to be added to specify the space the lattice structure will be in. The body block used for these heat exchangers was a box block. In this box block, the length and width should match that of the base, and the height will be how tall the cooling structure will be, or in this case, 15.5 mm. For the center point, the x value and y value will match the base. The z value for the center point will be equal to half the height of this box block and half the height of the base body block. This z value will place the cooling structure right on top of the base. Now that the body block for the walled TPMS block is specified, the cell size, approximate thickness, and approximate bias length can be inputted. Once those values are input, the fill type can be selected, and the lattice structure will appear. The last step in making the heat exchanger is to add a boolean union block to combine the base to the lattice structure of the heat exchanger. Once the heat exchanger is completed, it is ready to be meshed and made into an FE model for thermal analysis.

The workbooks used to design and simulate the heat exchangers under thermal loads utilizes custom workbooks to streamline the iterative design process. The workbook “Complex Mesh Preset” is used to create a mesh from the implicit body of the heat exchanger to use in the thermal analysis. This preset starts by creating a surface mesh from the boolean union block, an implicit body. After the surface mesh is created from the mesh from the implicit body block, it is then input into the remesh surface block. This block takes the mesh input and refines it to better match the complex geometries used for the heat exchangers. Finally, that mesh is input into the robust tetrahedral mesh block to again remesh the input to clean up any defects that may have occurred during the last two meshes. This block produces a mesh that has valid, high-quality elements. This preset was created because the complex geometry of the lattice structures requires multiple steps to create a mesh that is free of any defects to be used for thermal analysis. The resulting meshes can be seen in figure 5

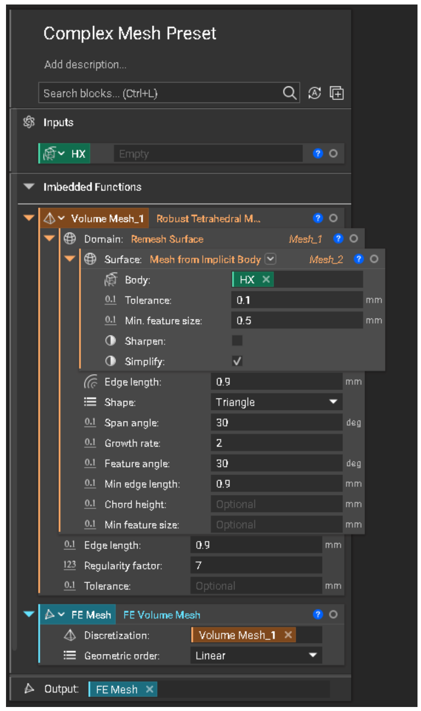


Figure 4:Internal settings for Complex Mesh Preset

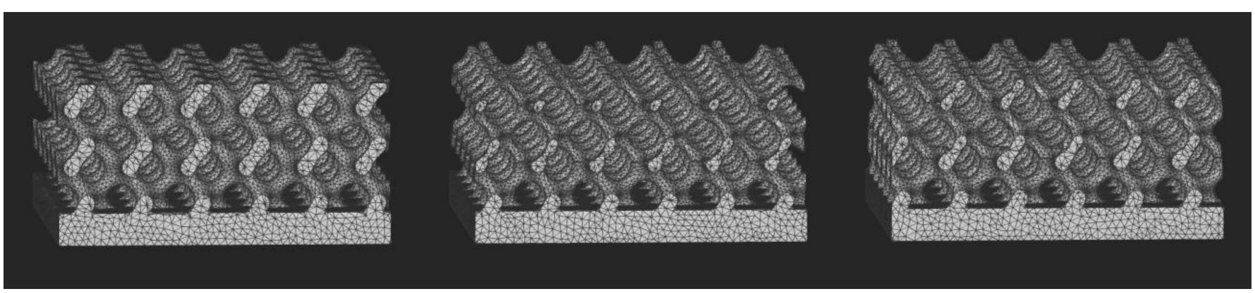


Figure 5: Resulting meshes on various heat exchangers from the complex mesh preset

The workbook "FE Model Preset" is used to add material properties to the mesh output by the "Complex Mesh Preset" to be used in the thermal analysis.

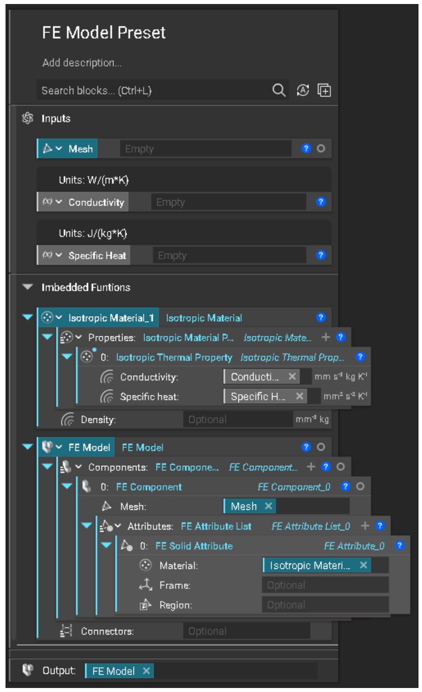


Figure 6: Internal settings for FE Model Preset

The workbook "Thermal Analysis for HX2" as shown in figure 7 is the workbook that sets up and runs the thermal simulations for each heat exchanger. It uses the mesh and model from the previous presets as well as 2 boxes to specify where the thermal loads for the analysis will be applied. These boxes are the boundary conditions used for the thermal analysis. The base plate box should be a box block that incases the entire base of heat exchanger. The fin face block should also be a box block but incases the cooling structure of the heat exchanger. This is also where the values for heat generation, convection coefficient, and ambient temperature are specified. Once everything is set up and ready, nTopology will run a steady-state heat conduction analysis from which the temperatures reported are assumed that the system has reached equilibrium based on the prescribed heat flux boundary conditions.

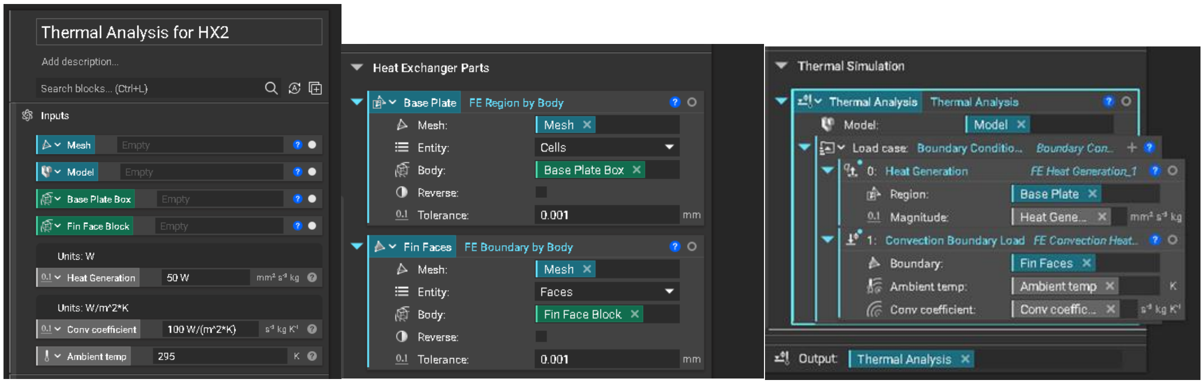


Figure 7: Internal settings for Thermal Analysis for HX2

For all variables entered into the previous workbooks, it is best practice to also enter in the units since the units in nTopology are not base SI units by default. However, this can be changed in the settings. The thermal analysis produced from this workbook for the control heat exchanger, as shown in figure 8, was compared to the thermal analysis simulated in Solidworks, as shown in figure 9, to validate the model for use with the heat exchangers that will not be able to be exported for thermal analysis in Solidworks. Based on the results from table 2, nTopology was considered to be an adequate substitute for Solidworks thermal simulations.

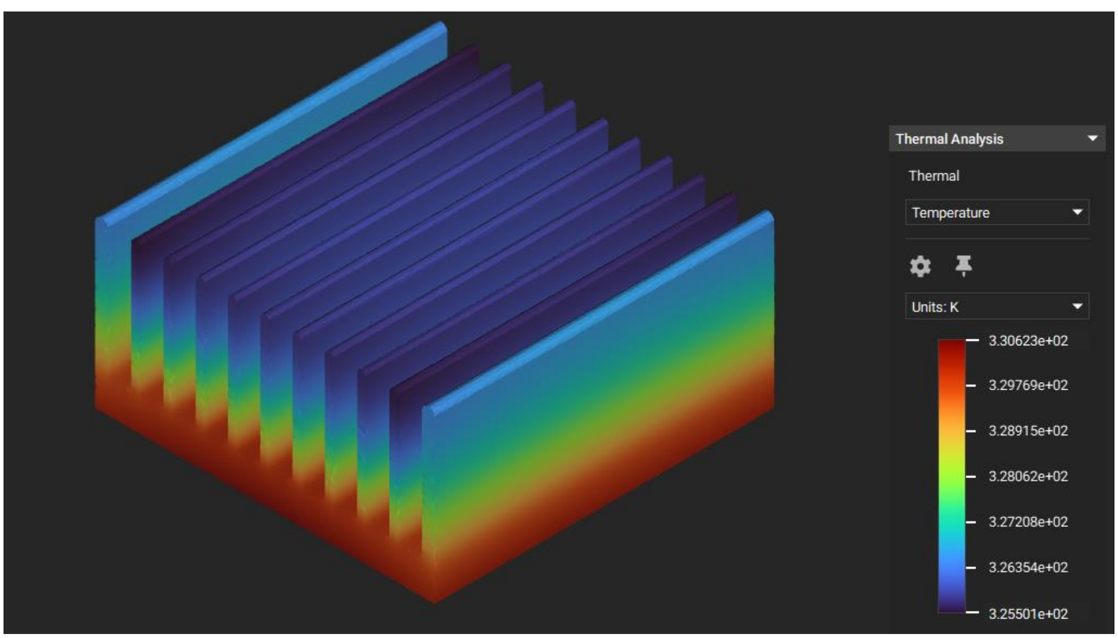


Figure 8:Results of thermal analysis of control heat exchanger in nTopology

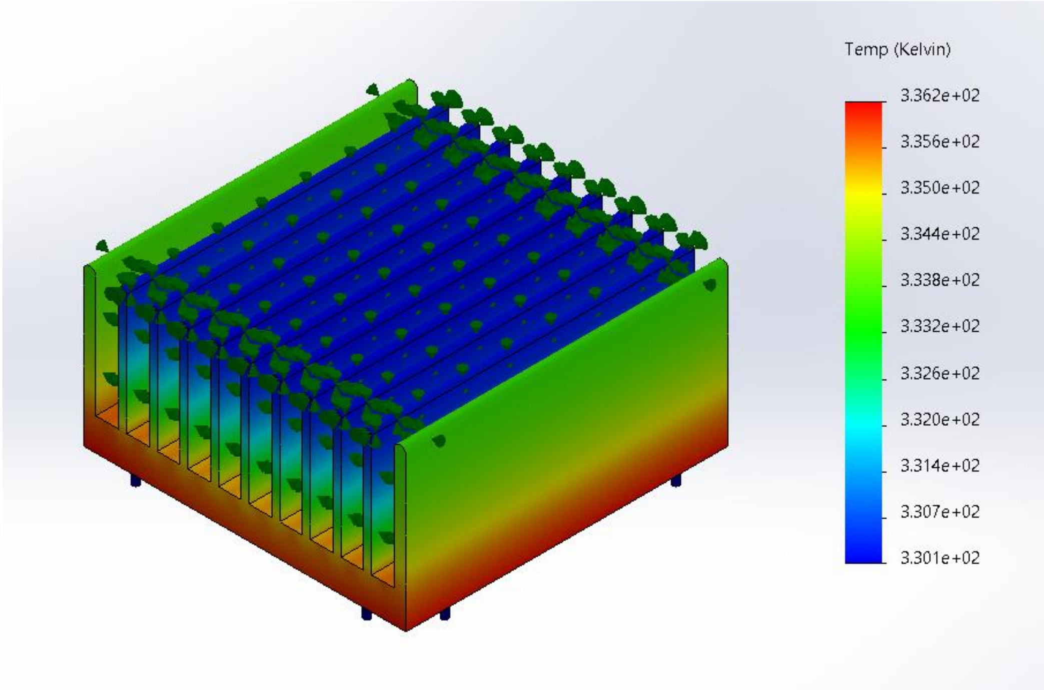


Figure 9: Results of thermal analysis of control heat exchanger in Solidworks

	Tmax	Tmin	deltaT
Control HX SW	336.2	330.1	6.1
Control HX	330.625	325.501	5.124
error %	1.658239143	1.393214178	16

Table 2:Error calculated between thermal analysis results between Solidworks and nTopology for the control heat exchanger

The subsequent heat exchangers were developed by adding a temperature thickness modifier block in place of the approximate thickness variable for the walled TPMS block, as seen in figure 10. This modifier takes the thermal analysis of the heat exchanger before and adjusts the thickness of the lattice structure to the minimum and maximum thickness variables according to the temperature at each point. The resulting heat exchangers are shown in figure 11.

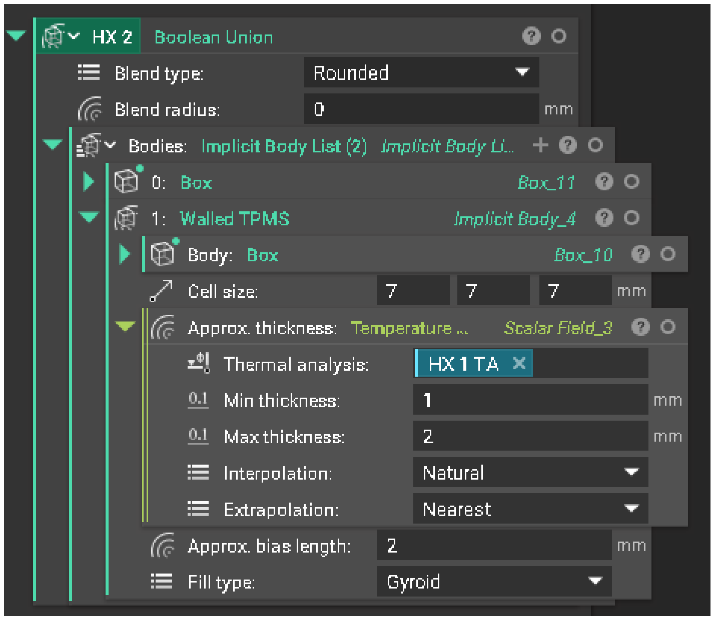


Figure 10: Ramp function used to modify the thickness of the lattice heat exchangers based on temperature

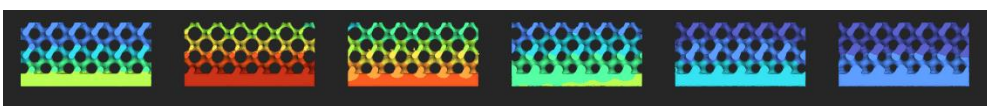
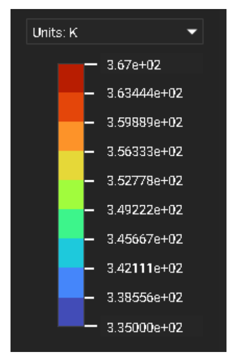


Figure 11: Heat exchangers developed and simulated with the method described



3. Results and Discussion

The heat exchangers in figure 11 are not iterations one after another. During the initial design, it was noticed that by keeping the thickness ramp the same from iteration to iteration, the design would quickly converge. This is noticeable from heat exchanger 2 to heat exchanger 3. As a result, the maximum thickness was changed after the thermal analysis of heat exchanger 2, resulting in heat exchanger 4. It was observed that by increasing the thickness of the lattice structure, the change in temperature and steady-state temperature per heat exchanger decreased, which meant better heat rejection to the surrounding area. The iterations from one heat exchanger to the next can be seen in figure 12.

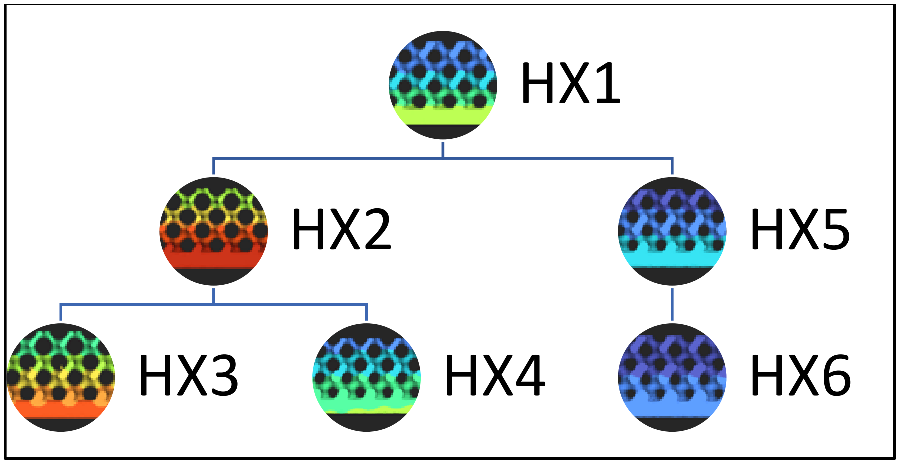


Figure 12:Sequence of iterations for the lattice heat exchangers

Since there was an inability to conduct flow analysis on the lattice heat exchangers, the best metric to compare their performance is by comparing thermal resistance. This metric can be calculated from the results of the thermal analysis of all heat exchangers.

Control HX	Tmax (K)	Tmin (K)	deltaT	Thermal Resistance (K/W)	Surface Area of cooling structure (cm^2)	volume (cm^3)	Porosity (%)	mass (g)
	330.625	325.501	5.124	0.10248	154.2276	13.8122	55.73012821	37.2929
Lattice HX								
1	351.055	339.35	11.705	0.2341	114.7329	11.8251	62.09903846	31.9277
2	367.019	344.117	22.902	0.45804	90.4433	9.4687	69.65160256	25.5656
3	360.498	344.257	16.241	0.32482	97.1518	10.0446	67.80576923	27.1204
4	349.518	339.59	9.928	0.19856	110.7157	11.9175	61.80288462	32.1772
5	345.278	336.601	8.677	0.17354	119.8484	13.1036	58.00128205	35.3796
6	341.413	335.384	6.029	0.12058	125.5366	14.3734	53.93141026	38.8082

Table 3: Thermal resistance and surface area of the cooling for all simulated heat exchangers

The control heat exchanger has the greatest surface area and therefore has the lowest thermal resistance. Heat exchanger 2 has the least amount of surface area and in return has the greatest thermal resistance. These results follow Newton’s law of cooling. While the control heat exchanger has a lower thermal resistance compared to all of the lattice heat exchangers, according to figure 13, if a lattice structure were to have the same surface area as the control heat exchanger, it could out perform the control heat exchanger.

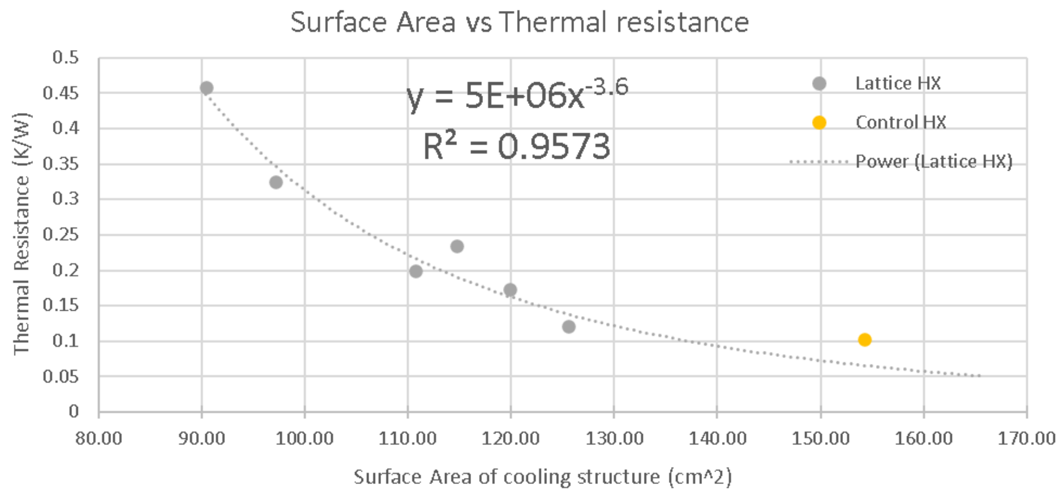


Figure 13: Surface Area vs. Thermal resistance plot with regression line

The next steps would be to continue iterating on the designs to reach a similar surface area between the lattice heat exchanger and the control heat exchanger. Another step would be to 3D print some of the heat exchangers to test for pressure drop and heat rejection. This would also be a good opportunity to observe how the flow field is affected by this complex geometry.

If this project was redone, then a computer with better specs than the one used initially used for the simulations since the complex geometries of the lattice structures are intricate and require hardware with better computational power and memory to produce results in a timely manner.

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