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# Multifunctional Components Using Engineered Lattice Structures as Materials

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**Abstract:** Designing a component with multiple functions, i.e. load bearing and noise attenuation, can increase the effectiveness of each component, and reduce the complexity of the overall system. Current multifunctional components include metal foam. It is believed that the pores of the metal foam can be engineered and optimized for desired characteristics. These components are comprised of lattice structures, whose physical properties as functions of the geometric characteristics have been previously determined. The current structure design focuses on thermal management and pressure-wave attenuation potential. COMSOL Multiphysics was used to determine the scale of thermal conductivities needed for a desired temperature drop. The initial design, metal foam, and a traditional finned heat sink were analyzed with COMSOL Multiphysics. Experimental tests were then conducted; and, the results were compared to those of COMSOL. Future work includes optimizing the structure for thermal management, and including noise attenuation in the analysis.

**Keywords:** Multifunctional components, lattice structures, effective thermal conductivities

## 1. Introduction

A component that serves multiple functions can greatly simplify systems. To design a multifunctional component engineered lattice structures are used. This paper details the thermal management of a component. The individual bulk thermal conductivities of lattice structures were determined as functions of strut diameter. Physically, the orientation of an engineered cube is a thermal resistor comprised of three parts; the fluid (air), the structure (unit cell), and the fluid-structure interaction. COMSOL is employed to determine the bulk thermal conductivity of three different unit-cell orientations. The equations were then modified to be functions of a scaled diameter, so that cube lengths could be changed. A preliminary heat sink was designed with the aid of COMSOL, and was cast, along with a traditional finned heat sink of the same mass. Experimental measurements were taken of the engineered heat sink, finned heat sink and commercially-available aluminum foam for comparison of their performances.

## 2. Methods

In order to design a component, the characteristics of an individual structure need to be determined. The first part (Sect. 2.1) describes the analysis in determining the effective thermal conductivity of each lattice structure. The second part (Sect. 2.2) details the design of a preliminary structure to experimentally test, and determination of the accuracy of the analysis.

### 2.1 Thermal Characteristics of Individual Lattice Structures

The thermal conductivities were determined for three lattice structures (Figure 1): Cube, Supercube, and Ultracube. First, the analysis was solved ignoring the effects of convection. Air was prescribed as the fluid inside the structure; and, the solid was set to aluminum (thermal conductivity, K=160 W/mK). One side of the cube, as well as the air at the surface was set to 400 K. The opposite side of the cube was set to 300 K. All other sides were thermally insulated.

**Figure 1: Different cubic geometries. From left to right; Cube, Supercube, and Ultracube.**

Assuming the heat flow is uni-directional, Fourier's law of heat conduction results in:

$$q"_x = -k_{effective}\, dT/dx \quad (1)$$

Simplifying equation 1 by substituting dT with the prescribed temperature drop of 100 K and dx by the length of the cube, Equation 1 can be written as:

$$\int q"_x\, dA = -k_{effective} A\, \Delta T/\Delta x \quad (2)$$

Utilizing the surface integration of COMSOL, the only unknown is the effective thermal conductivity. This method gave an artificial heat flux through the cube. A different approach was then considered in order to eliminate this unrealistic result, and give a more natural heat flow. Two copper blocks were placed at each end of a cube. The temperature difference was then prescribed to the outer side of the copper cubes; and, the rest of the sides were thermally insulated. This method removed the artificial temperature gradient. A new post-processing technique had to be used in order to define the effective thermal conductivity of the lattice cubes. A thermal resistance network was used, where:

$$R_t = R_1 + R_2 + R_3 \quad (3)$$

R1 and R3 are the known resistance of the copper blocks. Thus, the thermal resistance can be substituted into equation 2 resulting in:

$$\int q"_x\, dA = -\,\Delta T/R_T \quad (4)$$

Convection effects were then investigated by adding a volume force to the fluid. The components are designed for passive thermal management; therefore, natural convection must be included in the analysis. Natural convection will occur when buoyancy forces overcome viscous forces. This can be determined from a critical value of the Rayleigh number. If the Rayleigh number is above 1708, natural convection will occur. The Rayleigh number can be determined by:

$$Ra_x = Gr_x Pr = \frac{g\beta(T_1 - T_2)L^3}{\alpha\nu} > 1708 \quad (5)$$

If the effects of convection can be neglected, equations for the bulk thermal resistance of the lattice structures is only related to the bulk thermal conductivity. If convection cannot be neglected, equations for the bulk thermal resistance of the lattice structures are then comprised of the bulk thermal conductivity and convective heat transfer within the lattice structure.

### 2.2 Initial Component Design

In order to determine the scale of thermal conductivities, a preliminary model in COMSOL was developed. A 2D axi-symmetric model was built. The model had a heat source encompassed by a radial heat sink. The heat sink was divided into sub-materials, varying the thermal conductivity through the structure. The outer surface of the sink was set to external natural convection. The thermal conductivities were varied until a scale that gave an acceptable outer temperature was found.

A geometric relationship was determined for a radial heat sink that kept the lattice structures of the radial component as cubic as possible. The relationships were a function of how many slices (fins) and how many columns.

The unit-cell equations that were derived are a function of length of the cube and diameter of the struts. With current manufacturing processes, the minimal strut diameter is limited to one millimeter (1 mm). A script was then written to determine the thermal conductivities by varying the number of slices and columns in the heat sink. No orientation of slices could be developed with the initial equations derived. The thermal analysis of the lattice structures was re-analyzed and the material was varied. It was determined that the initial equations derived could be scaled to K_new_material/K_original.

Using Mold Star 22™ (K=43.2 W/mK), a component could be designed with 15 slices and 5 columns. The engineered structure and corresponding finned heat sink (of the same material and mass) are shown in Figure 2.

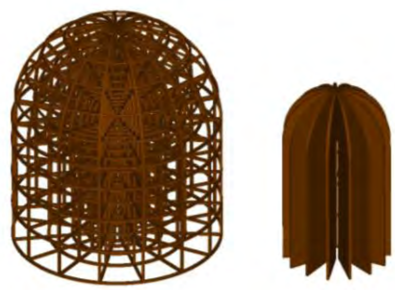

**Figure 2: (Left) preliminary and (right) finned heat sink**

## 3. COMSOL Modeling

The COMSOL analysis was solved for the aluminum foam, finned heat sink and engineered sink. Each model included a heat source encompassed by the heat sink. The power of the source varied. If the outer surface of the sink was set to external natural convection: the characteristic length was found; the fluid was set to air; and, the ambient temperature was set to 298 K. The outer temperature of the structures was then averaged in post-processing to compare to experimental results.

A 2D axi-symmetric COMSOL model was solved for the aluminum foam, with a bulk thermal conductivity of 5.8 W/mK. The outer surface of the foam was set to external natural convection.

Prior to using COMSOL, a numerical model was solved of a single heat-sink fin, providing a range of convective heat transfer coefficients of 7-8 W/K*m^2, depending on the power input.

Two 3D models of the finned heat sink were solved in COMSOL. One was of the entire finned sink, while the second was of a singular fin. The first analysis of the entire heat sink used external natural convection; and, two additional analyses prescribed a constant convective heat transfer of 7 and 8 W/K*m^2, respectively. The 3D singular-fin model was analyzed with these same boundary conditions.

A single slice of the preliminary structure was solved for using COMSOL. There are two reasons for only solving a single slice, and not the entire structure. The first reason was that the analysis of the single fin, compared to the full analysis of the finned heat sink, differed by a maximum of only 10.27% (depending on the power input). The second reason was due to computing power: we did not have the capabilities of solving a full 3D model in an acceptable amount of time. The COMSOL model of the preliminary structure was analyzed with setting the surfaces to a constant convective thermal coefficient. The models were solved with the thermal coefficient varying from 7-10 W/K*m^2.

## 4. Experimental Setup

The aluminum foam, finned heat sink and engineered structure were tested. Each heat sink was designed to cover a 150 W heating element. The power of the element was controlled by varying the voltage; and, both the voltage and current were measured with a digital multi-meter (DMM). The outer surface of each heat sink was measured with a type-K thermocouple. The experimental results are shown in Figure 3.

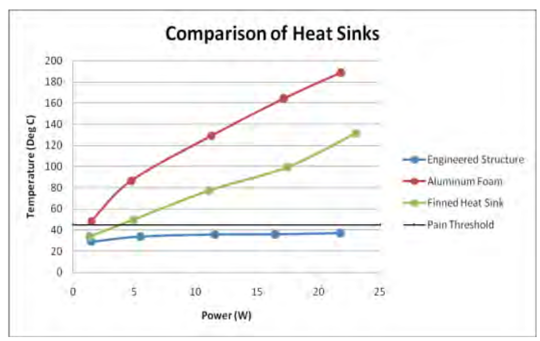


**Figure 3: Experimental results of heat sinks**

## 5. Comparison of COMSOL and Experimental Results

To compare the COMSOL and experimental results, a linear equation was derived from the experimental results for the foam and finned heat sink. The linear relationship of the aluminum foam had a maximum error of 19.48%, which occurred at the lowest power input. The error was only 2.69% at the highest power input. The maximum error from the derived equations compared to the FEA for the aluminum foam had a maximum of 37.26% and minimum of 11.70%.

The FEA and experimental results are shown in Figure 4.

A linear relationship of the experimental results was also derived for the finned heat sink. The maximum error of the derived equation was 4.67% and the minimum error was 0.37%. Comparing the FEA results to the derived equation resulted in a maximum error of 8.58% and minimum of 1.11%. The FEA results of the external natural convection, a constant convective thermal coefficient, and experimental results, are shown in Figure 5.

The experimental results did not yield a linear relationship for the engineered structure; therefore, a quadratic equation was derived over the range of the tested power inputs. The maximum error in the derived equation was 3.26% and the minimum error was 0.07%. The results of the experiment and the FEA of varying the convective thermal coefficient from 7-10 W/K*m^2 are shown in Figure 6.

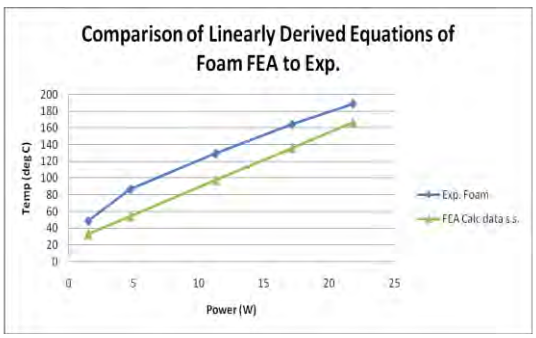


**Figure 4: Foam Results**

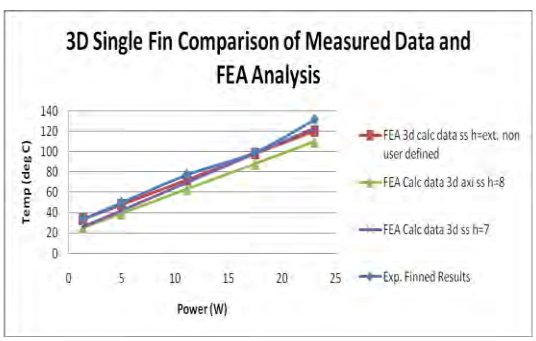


**Figure 5: Fin Results**

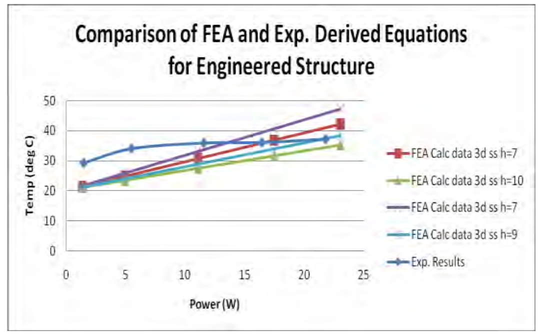


**Figure 6: Engineered Structure Results**

## 6. Discussion

The experimental results show that engineering a porous structure can result in a much more effective component. The pain threshold for a typical person is considered 45 degrees Celsius. When tested, the engineered structure was always below the pain threshold. The foam and finned heat sink exceeded the pain threshold at about 5 W.

The engineered structure combines the idea of a porous material with a finned structure. The engineered structure has the same mass as the finned heat sink; thus, if size is not a design constraint a component can utilize the lattice structures to greatly increase the efficiency.

The metal foam is not a valid direct comparison to the engineered structure. The metal foam was not sized to match the mass of the engineered structure. The foam tested was on the same size scale as the finned heat sink, however. Thus, if size is a design constraint, the finned heat sink could be a better option than the metal foam. To determine if a finned heat sink would be a better design, metal foam comprised of Mold Star 22™ would need to be analyzed in order to make a direct comparison.

The comparisons of the FEA and experimental data show a good correlation between our model and experimental setup. For simplicity, the engineered structure did not have a COMSOL model with a boundary of external natural convection. The numerical analysis as well as the results of the single fin show that the convective thermal coefficient does not vary greatly. The convective thermal coefficient will vary more in the structure results due to the pores, and will change with increasing power.

Knowing the range is between 6-12 W/K*m^2, the COMSOL results (although linear) provide a good basis for validating a design.

The offset in each graph is due to the room temperature of the experiment being slightly higher than the value used in the COMSOL model.

## 7. Conclusion

An engineered porous structure can be designed that greatly increases the effectiveness of thermal management. The engineered structure can also be coupled with the previously-defined structural characteristics of the lattice structure to design a singular component with both structural stability and desired thermal traits. Future analysis of noise attenuation of individual lattice structures will also be coupled to current derived equations thus increasing the effectiveness of a singular component.

## 8. References

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