DESIGN OF PERIODIC CELLULAR STRUCTURES FOR HEAT EXCHANGER APPLICATIONS

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Abstract

Powder based metal additive manufacturing processes generally produce parts with a textured surface. Although surface roughness is undesirable in most cases, it can be advantageous for applications such as heat exchangers or catalyst support structures. While stochastic metal foams have been used for these applications, it is not possible to tailor the foam geometry to the application. SFF techniques permit designers to use different cell geometries and orientations to achieve specific performance objectives. More specifically, the unit cell type and size, ligament size, and cell orientation have a considerable influence on surface area, volume, density and fluid flow behavior of the lattice structure. This paper illustrates the effect of the orientation of hexagonal periodic cellular structures on heat transfer and pressure drop. Finite element analysis of air flowing through the hot hexagonal periodic cellular structures shows the effect of orientation on heat transfer and pressure drop. The analysis indicates that the ideal orientation is dependent on the specific application.

Introduction

Periodic cellular materials are a class of materials whose properties can be tailored to enhance thermal, electrical and/or mechanical properties. Open celled materials with high porosity, large surface area and light weight are very well suited for heat exchange applications. Cellular materials have historically been classified as either stochastic (e.g. random cell geometry) or periodic (e.g. engineered repeating cell geometry). There are several methods of fabricating periodic cellular structures described in the literature, but Solid Freeform Fabrication (SFF) techniques give maximum flexibility in terms of tailoring the shape and size of the lattice to the needs of the application. SFF techniques are capable of producing near net shaped parts with textured surfaces as shown in Figure 1. Although surface roughness is undesirable in many applications, it can be advantageous for applications such as heat exchangers or catalyst support structures. Surface roughness can affect both the heat transfer and pressure drop characteristics of the fluid flow through periodic cellular structures. Heat transfer and pressure drop is also affected by cell geometry, cell size, foam density and the cell orientation, where cell size and foam density are a function of ligament radius and ligament length for a given cell geometry in periodic cellular structures.

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A great deal of research has been done with regard to estimating and optimizing the foam density and cell geometry for heat transfer applications, however there has been relatively little work on optimization of the cellular structure's orientation with respect to fluid or gas flow. With SFF techniques, virtually any cell geometry at any orientation and of any ligament radius can be fabricated. The Electron Beam Melting (EBM) process is one SFF technique which has the capability to produce fully dense metal parts in materials of interest for heat exchanger applications such as aluminum or copper (Figure 2).

(a)   **(b)**
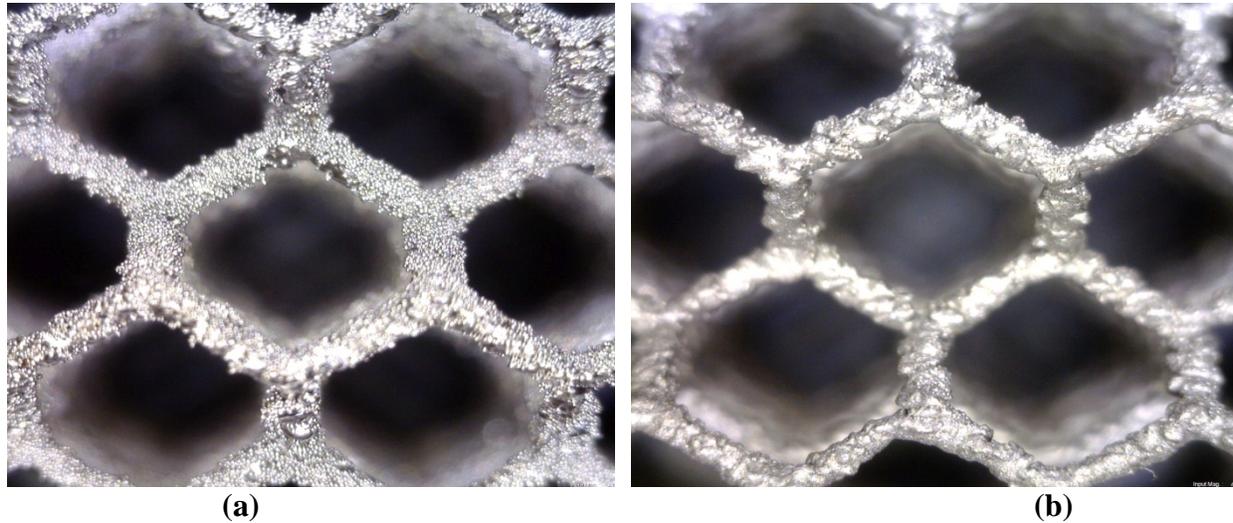
Figure 1 Micrographs of Ti-6Al-4V hexagonal cellular structure made by EBM process before (a) and after (b) sand blasting

In the present work, a hexagonal periodic cellular structure made via the EBM process is modeled, and the effect of orientation on heat transfer and pressure drop has been analyzed. A combination of five different lattice orientations (0, 30, 45, 60, and 90 degrees) in each of the X, Y, and Z axes has been considered, making a total 125 different lattice orientations. The hexagonal cellular structure (Figure 3a) has the same shape with respect to the X and Y axes, hence an orientation of 30 and 60 degrees in the X axis yields the same effective cell orientation with respect to gas flow direction (Figure 3b). When redundant cell orientations are excluded, the number of unique orientations is reduced to 15. Finite element analyses of those 15 orientations have been performed using COSMOS Floworks 2008 to estimate the heat transfer and pressure drop. The attributes of the hexagonal cellular structure compared using three non-dimensional groups include (a) the Nusselt number (N_u) characterizing the heat transfer, (b) the friction factor (f) for the pressure drop and (c) the Reynolds number (R_e) for the fluid flow rate.
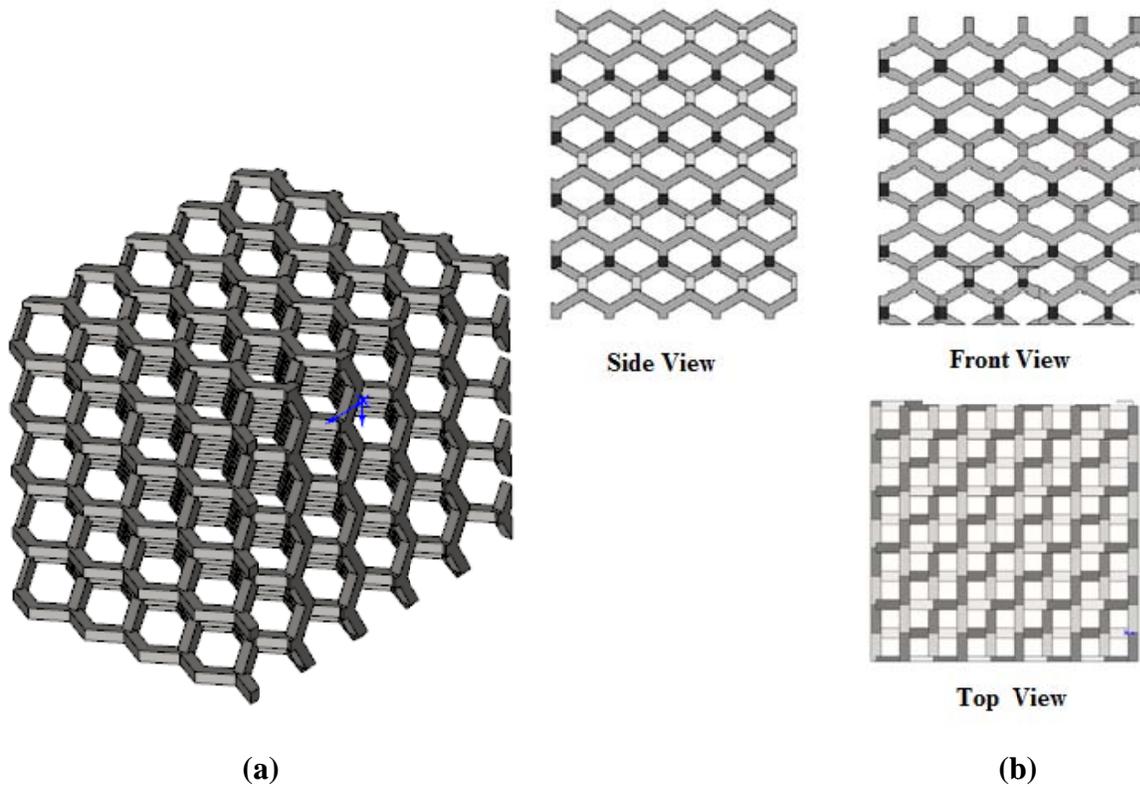
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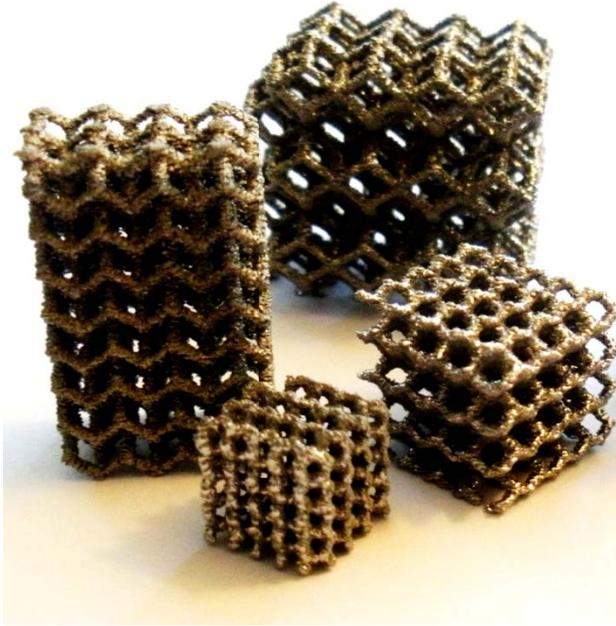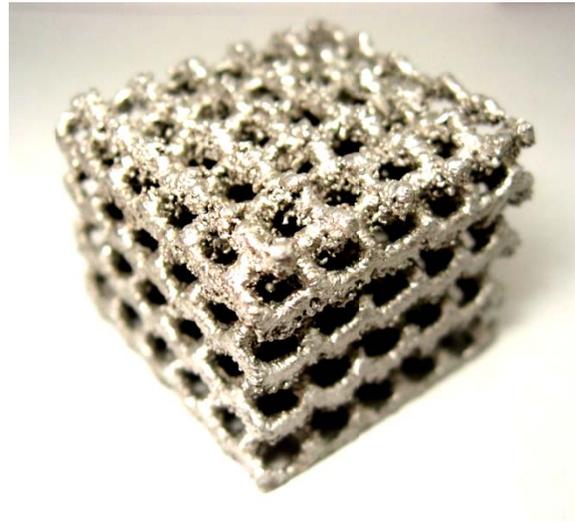
(a) **(b)**

Figure 2 (a) Ti-6Al-4V and (b) Al-6061 cellular periodic structures made by EBM process

Side View
Front View
Top View

(a) **(b)**

Figure 3 (a) Perspective view (b) First angle orthographic projection of the hexagonal periodic cellular structure

Background

Transport phenomenon in stochastic metal foams has received considerable attention during the last decade. Lee et al. [1] studied the application of metal foams as high performance air-cooled heat sinks in electronics packaging. Sathe et al. [2] studied combustion in metal foams for application in porous radiant burners. Hunt and Tien [3], Calmadi and Mahajan [4], Boomsma and Poulikakos [5], Lu et al. [6] and Fuller et al. [7] have addressed the thermal characteristics of metal foams. Wadley and Queheillalt [8] have compared the thermal properties of copper textile-based lattice structures with the stochastic open-cell metal foams.

Tian et al. [9] studied the effects of topology upon fluid flow and heat transfer within cellular copper structures of different density. Experiments were performed with the prototype sandwiches of two different cell geometries and two orientations, 0 and 90 degrees with respect to the front plane. The friction factor was found to be independent of coolant velocity and dependent on open area ratio. For a particular porosity, the heat dissipation rate was found to be directly proportional to the surface area density. For a fixed surface area of pure copper, the optimal porosity was found to be 0.75. The overall heat transfer was primarily dependent upon porosity and surface area density. It was weakly dependent upon orientation. Lu et al. [10] reviewed the thermal characteristics of metallic sandwich structures by analytical modeling and numerical simulation. The Reynolds number, Nusselt number and friction factor of different types of sandwich structures were calculated and compared to conclude that topology preference is highly application dependent.

Heat Transfer Model and Calculation

A CAD model of a hexagonal periodic cellular structure (Figure 3a) of size 40mm x 40mm x 11.5mm was made on the SolidWorks 2008 CAD package. The lattice was merged with a square tube such that it was possible to simulate gas entering the tube on one end (e.g. the inlet), flowing through the lattice, and then exiting via the outlet on the other end of the square tube. Although process parameter and post processing will significantly influence surface roughness, the cellular structure was assigned a surface roughness of 300 microns for purposes of this study. That figure is representative of a part fabricated with intentionally high surface roughness for purposes of heat exchange. The lattice heat exchanger was given a constant temperature of 120° C with an inlet gas temperature of 20° C flowing at a velocity of 9.55 m/s. Finite element analysis (FEA) of the heat transfer and pressure drop was conducted using COSMOS FloWorks 2008 for each lattice at orientation. The outputs of the FEA were average temperature and pressure of air at the outlet and the heat flux of the cellular structure perpendicular to the air flow. The surface area and the volume of material in the cellular structure were determined from the SolidWorks 2008 CAD model of the cellular structure.

The temperature difference between the air at the inlet and outlet refers to the amount of heat transferred to the air by the cellular structure at a given orientation. Similarly, pressure drop reflects the amount of hindrance in the path of the fluid. Relative density of the lattice structure was obtained by calculating the ratio of the volume of the material in the structure and the total

rectangular bounding box volume occupied by it. To obtain the Nusselt number, Reynolds number and friction factor, the hydraulic diameter (D_h) must be calculated. This is given by

$$D_h = \frac{4l_c \sqrt{1-\rho}}{c} \tag{1}$$

where l_c is the cell edge length , ρ is relative density, and c is the cell geometry constant [11]. The Nusselt number (N_u) is defined as the ratio of convective to conductive heat transfer across the boundary and is given by

$$N_u = \frac{hD_h}{k_f} \tag{2}$$

where h is the local heat transfer coefficient and k_f is the thermal conductivity of air. Reynold's number (R_e) defines the flow and is given by

$$R_e = \frac{\rho_f U D_h}{\mu_f} \tag{3}$$

where ρ_f is density, U is the flow velocity, and μ_f is the viscosity of the fluid (air). The friction factor (f_s) is a measure of the resistance provided to the flow by the cellular structure. Friction factor is given by

$$f_s = \frac{\frac{\Delta p}{L} D h}{\frac{p_f U^2}{2}} \tag{4}$$

where Δp is the difference of pressure at the inlet and the outlet of the setup and L is the length of the cellular structure. Table 1 shows the various values obtained from simulation and the calculated values.

Table 1 Data for different orientation of Hexagonal periodic cellular structure (α, β, γ is the angle in x, y and z plane respectively)

| Trial Num	Orientation (α, β, γ)	Avg. Vel. (ms^{-1})	Outlet Temp. (°C)	Temp Increase (°C)	Static Press. (Pa)	Press. Drop (Pa)	Heat Flux (Wm^{-2})	Nusselt Number	Reynolds Number	Friction Factor
1	(0,0,0)	5.275	35.254	15.254	101350	25	3.277	3.508	1558.575	0.175
2	(30,0,0)	5.341	32.987	12.787	101355	30	2.763	2.962	1324.802	0.172
3	(45,0,0)	5.349	37.685	17.485	101355	30	4.987	3.404	1155.105	0.149
4	(0,45,0)	5.382	40.121	19.921	101344	19	5.846	3.875	1285.952	0.103
5	(0,30,0)	5.363	38.359	18.159	101352	27	5.650	5.198	1621.539	0.187
6	(30,30,0)	5.382	37.488	17.288	101356	31	6.711	4.952	1242.449	0.163
7	(30,45,0)	5.152	39.288	19.088	101360	35	6.357	3.806	1065.308	0.179
8	(45,30,0)	5.341	36.169	15.969	101355	30	5.987	4.168	1074.468	0.139
9	(45,45,0)	5.417	37.849	17.649	101361	36	5.689	3.083	937.486	0.140
10	(45,45,45)	5.321	35.786	15.586	101354	29	5.773	4.072	1058.621	0.134
11	(45,45,30)	5.319	35.782	15.582	101354	29	5.687	3.394	894.947	0.113
12	(45,30,30)	5.346	36.387	16.187	101356	31	5.471	3.249	930.063	0.124
13	(30,30,30)	5.339	35.869	15.669	101357	32	5.677	3.869	1031.584	0.143
14	(30,45,30)	5.3414	36.327	16.127	101355	30	5.184	3.308	994.645	0.129
15	(45,30,45)	5.4312	37.861	17.661	101363	38	5.431	2.968	948.519	0.148

Analysis and Discussion

A total of 15 unique lattice orientations with respect to the X, Y, and Z axes were considered. Orientation has been denoted in terms of the rotation angles α, β, γ about the X, Y, and Z axes respectively. Figure 4 shows how the pressure drop is influenced by the lattice orientation with respect to the gas flow direction. For example, orientation #15 (45,30,45) produces the highest pressure drop while orientation #4 (0,45,0) produces the lowest pressure drop among the considered orientations. This is intuitively explained upon examination of Figures 5(a) and 5(b). Figures 5(a) and 5(b) show a view of the lattice from the inlet looking towards the outlet for orientation #4 and orientation #15 respectively. In orientation #4, the gas has a relatively unobstructed path from inlet to outlet, hence the pressure drop can reasonably be expected to be relatively low. In orientation #15, the gas has a tortuous path from inlet to outlet, hence the pressure drop can intuitively be expected to be much higher.

Figure 6 shows the predicted temperature drop from inlet to outlet as a function of lattice orientation. Orientation #4 (0,45,0) has the highest temperature difference while orientation #15 (30,0,0) has the lowest. The effect of orientation on heat transfer and pressure drop can further be explained in terms of the Nusselt number and friction factor. A Nusselt number of unity indicates sluggish flow in which heat transfer due to convection is equal to conduction. The Nusselt number seems to follow the same trend as the Reynold's number for a given orientation. Higher value of friction factor shows greater hindrance to the path of the fluid. Orientation (0,30,0) has the highest Friction Factor, Reynolds Number and Nusselt Number (Figures 7, 8, and 9 respectively).
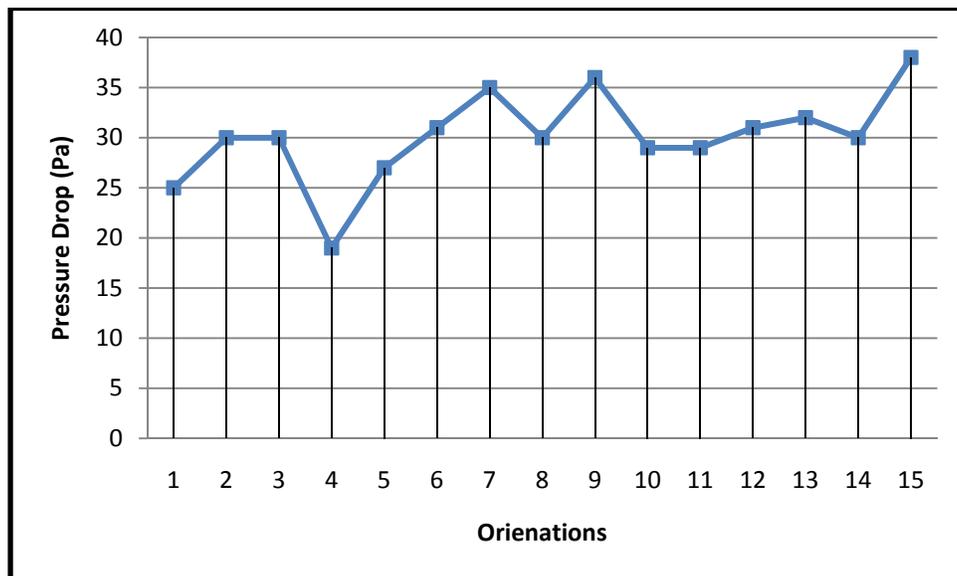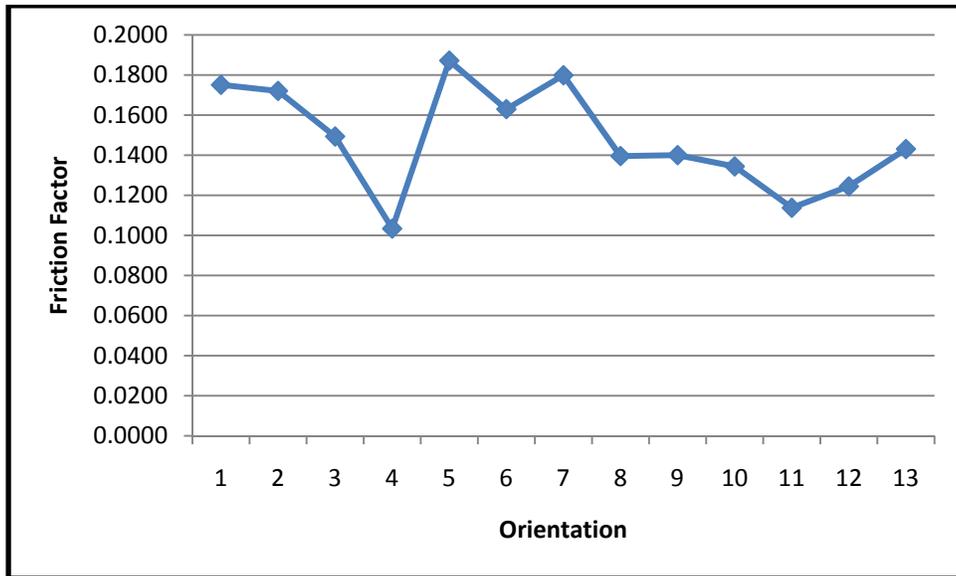

Figure 4 Pressure drop as a function of lattice orientation
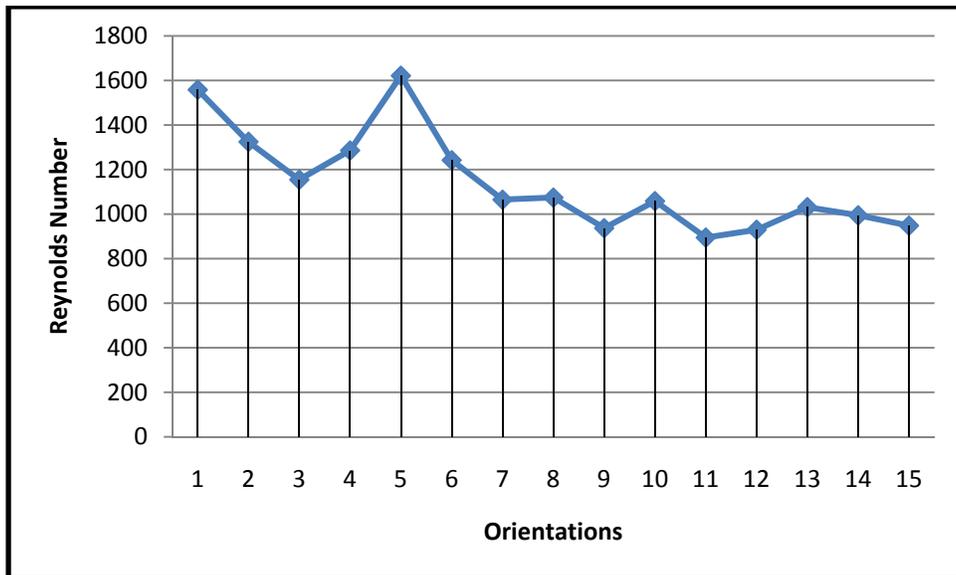
| (a) | (b) |
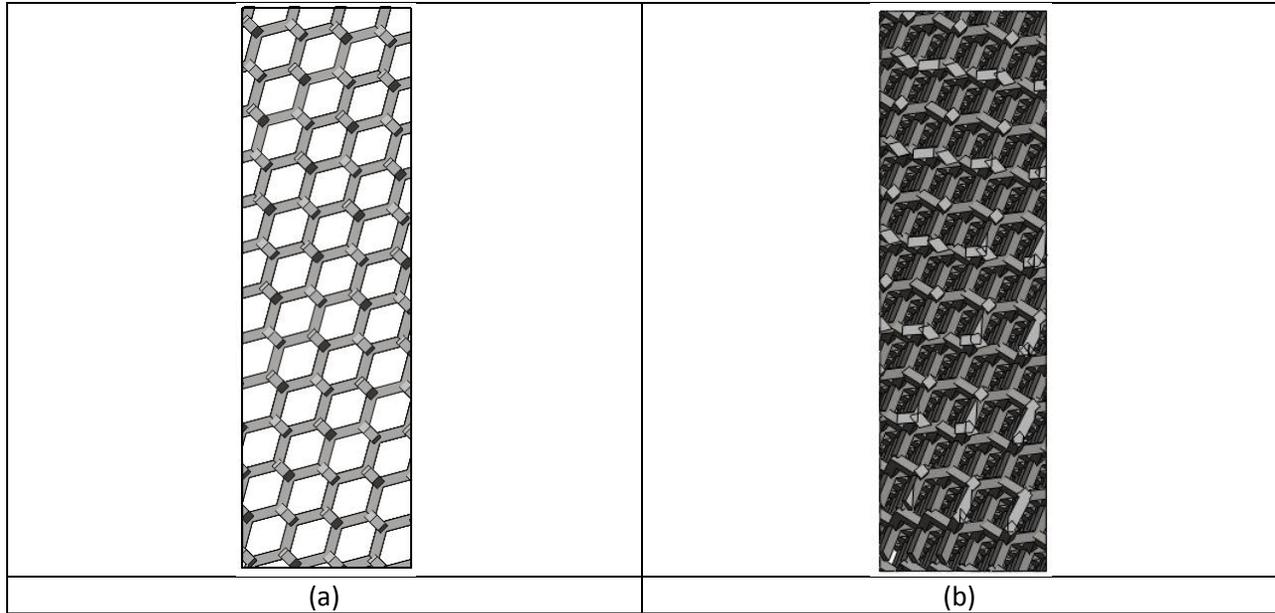
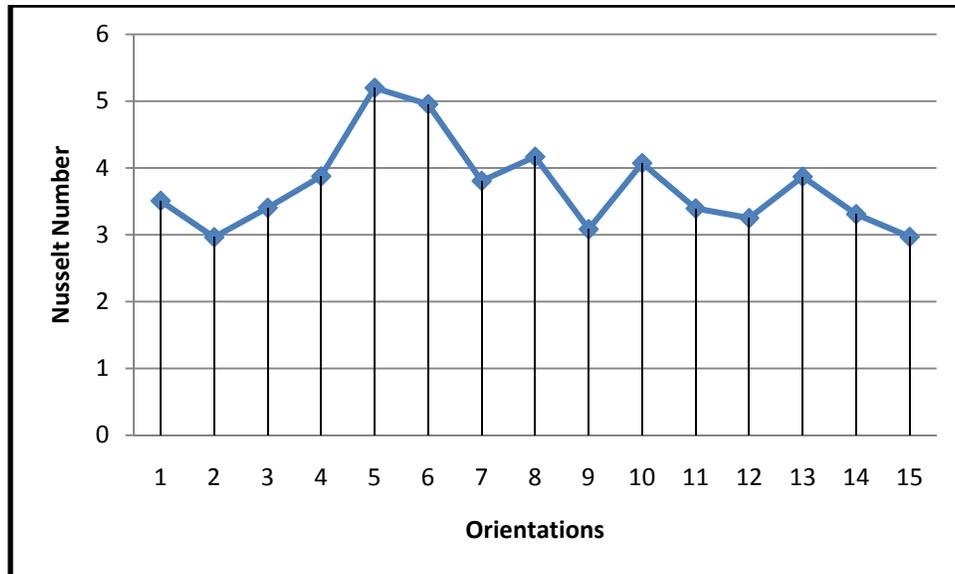**Figure 5 (a) (0,45,0) orientation; (b) (45,30,45) orientation**
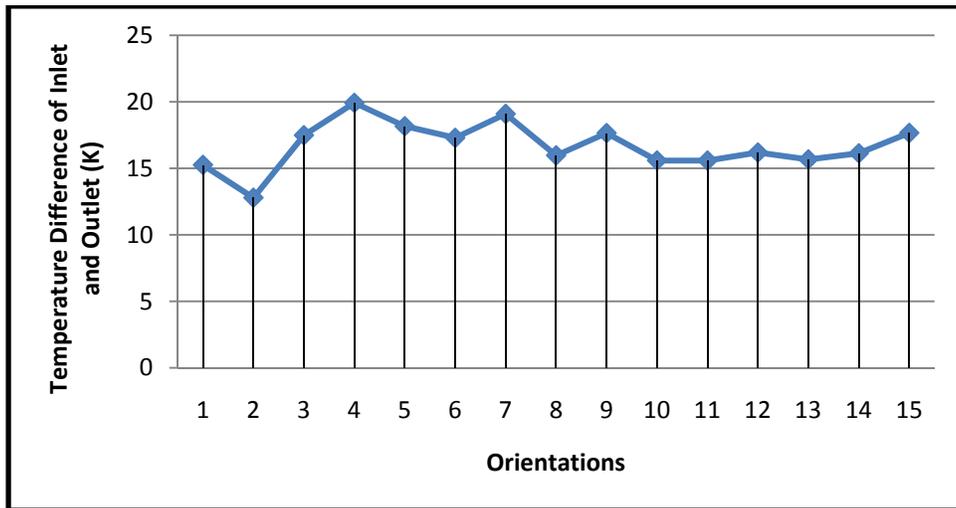
Figure 6 Temperature change versus lattice orientation

Figure 7 Friction factor versus lattice orientation

Figure 8 Reynold's number versus lattice orientation

Figure 9 Nusselt number versus lattice orientation

The implication is that the preference of the orientation depends on the needs and requirements of the application. Generally speaking, cellular materials the produce large pressure drops will require greater pumping power. the constraints are maximum allowable temperature in the metal and/or the fluid, or the pressure drop or pumping power. The thermal properties obtained in Table-1 can be used to design heat exchanger for required value heat transfer and pump capacity, implying that orientation preference depends on the application. Comparison of the performance of hexagonal periodic cellular structures with the thermal properties of stochastic and periodic structures [9,10] has been done in Figure 10. The hexagonal periodic cellular structure at the considered Reynolds number and density is close to the performance of the corrugated duct and louvered fins. The low friction factor implies lesser resistance to the fluid flow, and the lower nusselt number shows lower convection compared to copper textile mesh.

Conclusions and Future Work

This paper describes preliminary modeling effort targeted to the design of custom lattice structures that are tuned to the needs of a particular application. In this work, the influence of lattice orientation on flow and heat transfer properties has been modeled and analyzed. As expected, the orientation of the lattice with respect to gas flow direction does influence heat transfer efficiency and pressure drop.

The next step is to model the influence of different types of unit cell geometries and different lattice densities. One pressing issue for which no good solution has yet been devised is the issue of increased surface area in lattices fabricated via powder-based SFF processes. While FEA packages allow the user to specify surface roughness, the FEA does not actually use the surface roughness value to increase surface area. Rather, it is specified as a means of reflecting increased surface drag. However, surface area is known to have a very significant effect on heat transfer. It has been estimated that the surface area of an intentionally rough EBM lattice

structure is approximately double the surface area computed by the CAD model with smooth struts. While the simulation here is useful for describing trends, better analytical methods are needed to more accurately reflect the increased surface area of the actual lattice structures.

In order to determine just how significant the influence of surface area is on EBM-produced heat exchangers, efforts are currently underway to fabricate simple aluminum and copper lattice heat exchangers whose geometry matches those shown in Figure 3. Actual performance will be compared with predicted results, and those results will be used to successively refine and improve the analytical models presented in this paper as a starting point.

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