Experimental Testing of Additively Manufactured Lattice Structure for Vibration Isolation

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Abstract: This study present design, analysis and exploratory testing of strut build lattice structure for vibration attenuation. There is maximum number of achievable choices for creating strut build lattice structure within definite volume. This design space is reduced by fixing several lattice variable parameters. Modal analysis is used to find the Eigen frequencies and mode shapes of the lattice structure to study the resonance condition. The total area moment of inertia, mass, stiffness, compressive strength and deformation of the lattice structure are studied and compared with each selected six models to select the optimum lattice structure for attenuation of vibration. To find theoretical dynamic and static mechanical properties of lattice structure, finite element modeling has been used. The optimum lattice structure has been manufactured by fused deposition modeling (FDM) 3D printing process.

Index Terms – *3D Printing, lattice structure, vibration attenuation*

I. INTRODUCTION

3D printing permits nearly infinite geometric design privilege. Additively manufactured (AM) lattice structure provides enhanced properties as compared to solid structures [2]. It have low mass, high degree of design space and high impact energy absorption also it can be topologically optimized to various problems. The main focus of this study is to use a lattice structures for vibration isolation by using modeling and design tool. These structures are used to defeat waves of vibration which transmitting from one structure to another structure. Vibration damages the structure and causes manufacturing errors. So for that this paper presents design of lattice structure and how to select optimum lattice structure from many feasible lattice structures which all have same volume and made up of same material. Also it is designed in such a way that it can be able to sustain a mass load. Basically these types of structures are used in metrological and machine frames also it is used as support in small rotary machines [4].

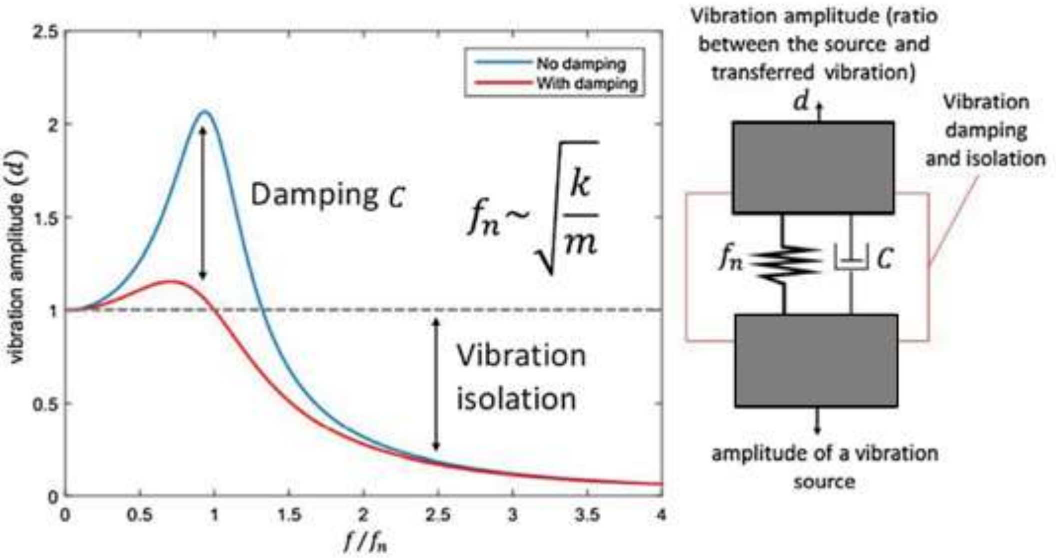


Figure 1: Frequency ratio (f/f_n) vs. Vibration amplitude (d)

Figure 1 shows the comparison between vibration isolation and vibration damping where it is assumed that vibration is propagated from outside vibration origin to machine. In figure X coordinate is the ratio r and Y coordinate is the displaced amplitude of vibration. The r is the ratio of external vibration frequency (f) to the modal frequency (f_n). To isolate the more part of vibration r must be have higher value for that fn must smaller than f. also by lowering the stiffness of the structure, larger value of r can be obtained but there is practical limitation because if stiffness is lowered then structure will ineffectual to sustain the mass load [1].

1.1 Lattice Structure Design

A lattice structure is made up of nodes and struts. Strut is a horizontal cylindrical member which has node at each end and node is a spherical member where one or more than one struts meets. If the number of struts and nodes are not constrained then there will be large design space available.

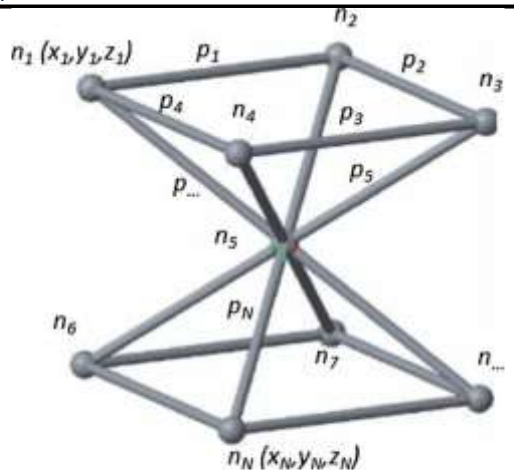


Figure 2: 9 nodes and 16 struts basic lattice configuration

The various design parameters include number of nodes, position of nodes and diameter of nodes and struts. By constraining them number of design layouts are remarkably reduced. In this paper six lattice models are presented with nine nodes as shown in figure 3. Each model has fixed volume of 40×40×40mm^3. The diameter of strut is taken as 4mm and diameter of node is taken as 5mm.

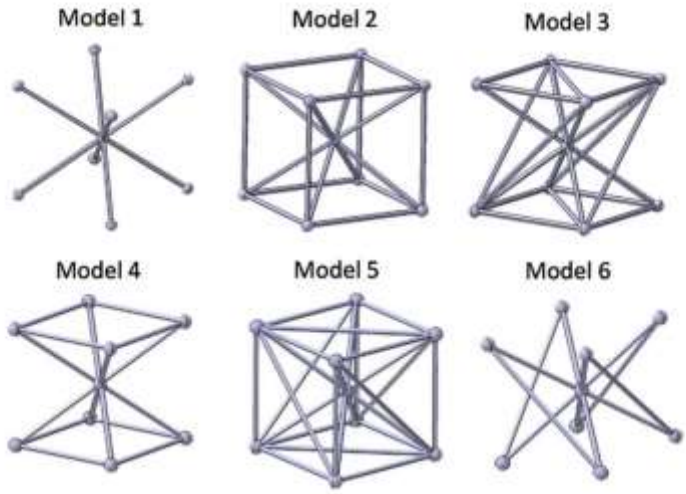


Figure 3: Single cell six lattice models

1.2 Variable Design Parameter

1) **Total area moment of inertia (J)**

It is consist of J_x , J_y and J_z about x, y and z axes respectively

$$J = J_x + J_y + J_z \quad (1)$$

$$\mathrm{J_x} = \sum_{\mathrm{i=1}}^{\mathrm{Ns}} \mathrm{J_{xsi}} + \sum_{\mathrm{j=1}}^{\mathrm{Nn}} \mathrm{J_{xnj}} \quad (2)$$

$$J_y = \sum_{i=1}^{Ns} J_{ysi} + \sum_{j=1}^{Nn} J_{ynj} \quad (3)$$

$$J_z = \sum_{i=1}^{Ns} J_{zsi} + \sum_{j=1}^{Nn} J_{znj} \quad (4)$$

Where,

J_{xs} , J_{ys} and J_{zs} are struts area MOI about x, y and z axes respectively

J_{xn} , J_{yn} and J_{zn} are node area MOI about x, y and z axes respectively

i is strut index and j is node index, N_S And N_n are the number of struts and nodes.

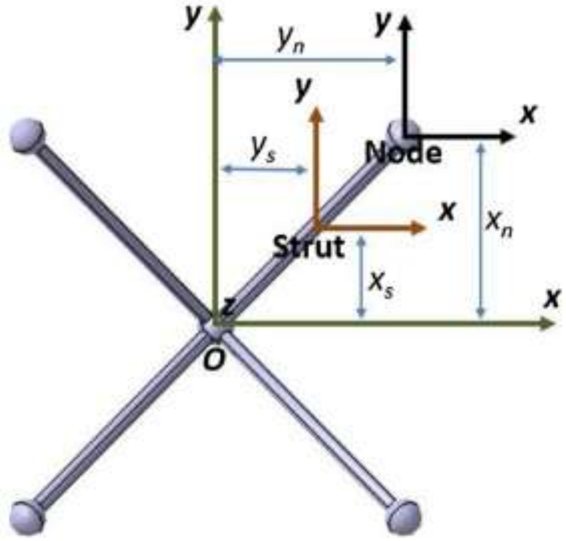


Figure 4: Illustration for inertia calculation in XY plane

The Huygens-Steiner theorem is used to finding out J for each strut and each node and it is find out with respect to global Co-ordinate system.

$$J_{xsi} = \left(\frac{l_s {d_s}^3}{12}\right) + \left(l_s d_s \times {X_s}^2\right) \tag{5}$$

$$J_{xsi} = \left(\frac{d_s {l_s}^3}{12}\right) + \left(l_s d_s \times {Y_s}^2\right) \tag{6}$$

$$J_{zsi} = J_{xsi} + J_{ysi} \tag{7}$$

Similarly,

$$J_{xni} = \left(\frac{\pi {d_n}^4}{64}\right) + \left(\frac{\pi {d_n}^2}{4} \times {X_n}^2\right) \tag{8}$$

$$\mathrm{J_{yni}} = \left(\frac{\pi \mathrm{d_n}^4}{64}\right) + \left(\frac{\pi \mathrm{d_n}^2}{4} \times \mathrm{Y_n}^2\right) \tag{9}$$

$$J_{yni} = \left(\frac{\pi {d_n}^4}{64}\right) + \left(\frac{\pi {d_n}^2}{4} \times {Y_n}^2\right) \tag{9}$$

$$J_{zni} = J_{xni} + J_{yni} \tag{10}$$

Where, l_s is strut length, $d_s$is strut diameter, d_n is node diameter, X_s ,Y_s and Z_s are the distance between origin o and C.G of struts and X_n ,Y_n and Z_n are the distance between origin o and C.G of nodes. Since node is spherical in shape it has same area moment of inertia at any orientation [10].

$$J_{xn} = J_{yn} = J_{zn} = \left(\frac{\pi {d_n}^4}{64}\right) \tag{11}$$

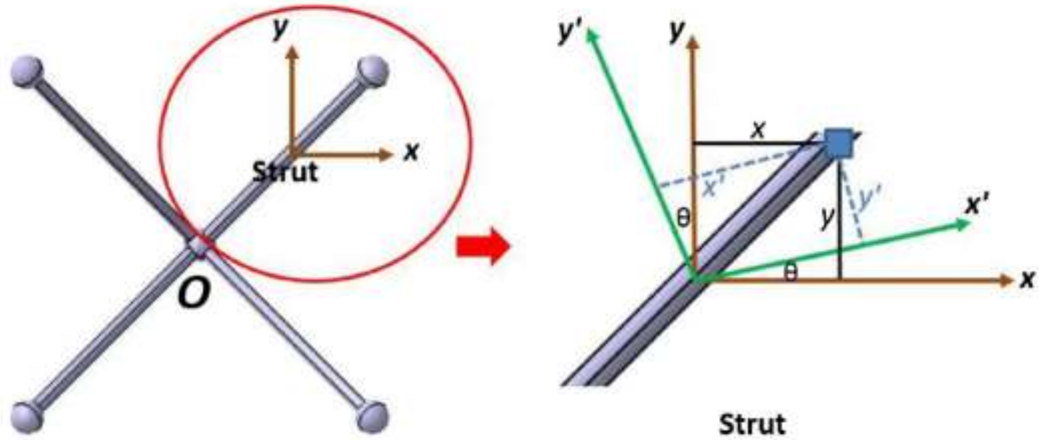


Figure 5: Application of parallel axis theorem in 2D XY plane

$$J_x = \int y^2 \, dA \, J_{x'} = \int y'^2 \, dA \tag{12}$$

$$J_y = \int x^2 \, dA \, I_{y'} = \int x'^2 \, dA \tag{13}$$

$$x' = x \cos\theta + y \sin\theta \tag{14}$$

$$y' = y\cos\theta - x\sin\theta \tag{15}$$

$$J_{x'} = J_x \cos\theta^2 + J\sin\theta^2 - J_{xy}\sin 2\theta \tag{16}$$

Similarly,

$$J_{y'} = J_x \sin\theta^2 + J_y \cos\theta^2 + J_{xy}\sin 2\theta \tag{17}$$

$$J_{x_{si}} = [\cos\theta_i]^2 \frac{d_s {l_s}^3}{12} + [sin\,\theta_i]^2 \frac{l_s {d_s}^3}{12} \tag{18}$$

$$J_{y_{si}} = [\cos\theta_i]^2 \frac{l_s {d_s}^3}{12} + [sin\,\theta_i]^2 \frac{d_s {l_s}^3}{12} \tag{19}$$

$$J_{z_{si}} = [\cos\theta_{xyi}]^2 \frac{l_s {d_s}^3}{12} + [sin\,\theta_{xyi}]^2 \frac{d_s {l_s}^3}{12} \tag{20}$$

Where,

θ_{xyi} is the strut rotation with respect to XY-plane. The calculations for $J_{x_{si}}$, $J_{y_{si}}$ and $J_{z_{si}}$ are summarized as follows

$$J_{x_{si}} = [\cos\theta_i]^2 \frac{d_s {l_s}^3}{12} + [sin\,\theta_i]^2 \frac{l_s {d_s}^3}{12} + \left(l_s d_s \times {X_s}^2\right) \tag{21}$$

$$J_{y_{si}} = [\cos\theta_i]^2 \frac{l_s {d_s}^3}{12} + [sin\,\theta_i]^2 \frac{d_s {l_s}^3}{12} + \left(l_s d_s \times {Y_s}^2\right) \tag{22}$$

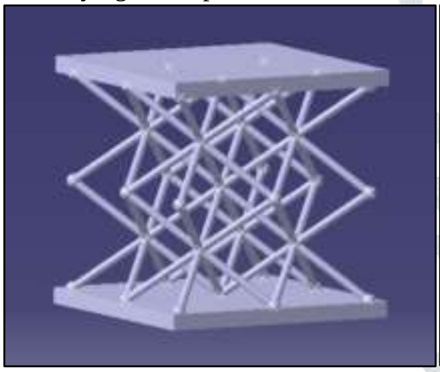
$$J_{z_{si}} = [\cos\theta_{xyi}]^2 \frac{l_s {d_s}^3}{12} + [sin\,\theta_{xyi}]^2 \frac{d_s {l_s}^3}{12} + \left(l_s d_s \times {Z_s}^2\right) \tag{23}$$

So, Equ.21, 22 and 23 gives total area moment of inertia for strut about central co-ordinate system.

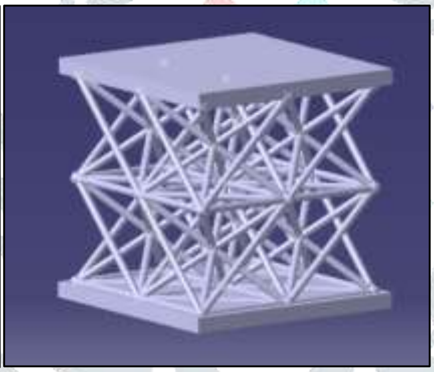
II. DESIGN OF LATTICE STRUCTURE

2.1 CAD models of lattice structure in 2×2×2 configurations

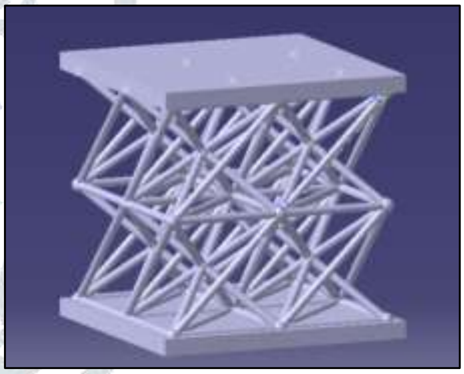
The six lattice models are created by varying struts form and pattern by keeping the symmetry and permitting replication in all three directions. Lattice structures are unusually used in the form of single unit cell. So they are generated and manufactured in the form of 2×2×2 unit cell configuration such that each lattice structure contains eight similar lattice cells which have simple cubic shape. Each lattice model has 27 nodes (n) and struts (p) are varied. The volume of each lattice model is 80×80×80 mm^3 and 5mm thickness plate is attached at top and bottom of the structure. The plates are attached to mount an accelerometer sensor for carrying out impact test.



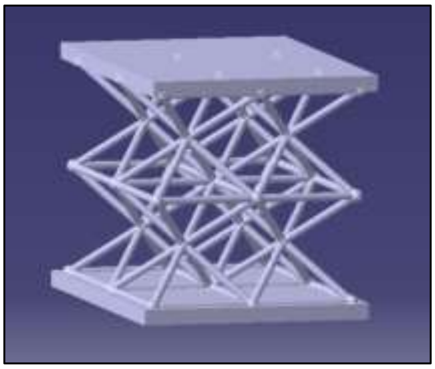
Model 1: p=64



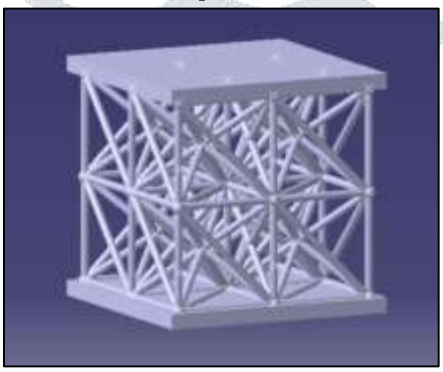
Model 2: p=115



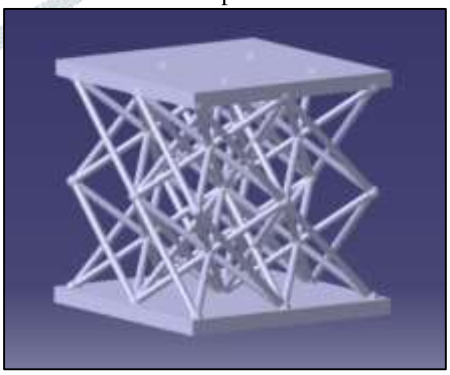
Model 3: p=141



Model 4: p=100



Model 5: p=153



Model 6: p=88

Figure 5: Representation of six 2×2×2 lattice CATIA models configurations

2.3 Axial stress analysis of each strut due to compressive stress

To sustain the load, axial stress induced in any strut member should be less than yield strength of lattice material. If induced axial stress in strut member is greater than yield strength, then structure will be not able to sustain the load and it will be fail.

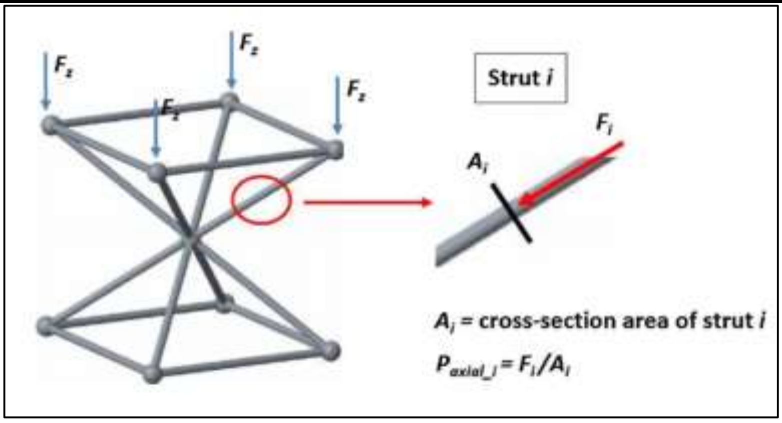


Figure 6: Illustration for axial stress analysis for each strut

An axial compressive stress for each i_{th} strut is designated by σ_{axiali}

$$\sigma_{axiali} = \frac{F_i}{A_i} \tag{24}$$

Where F_i is axial compressive force and A_i is area of cross-sectional of strut perpendicular to F_i.

$$F_i = \frac{A_s E}{l_s} d_f \text{ And } A_s = \pi \left(\frac{d_s}{2}\right)^2 \tag{25}$$

Where d_f axial deformation and E is is modulus of elasticity.

A stiffness matrix technique is used to find σ_{axiali} foe each node. An axial force at each strut is calculated by constructing global stiffness matrix and global displacement matrix.

$$F_{axial} = \mathbf{KD} \tag{26}$$

$$\mathrm{K} = \mathbf{T^T K' T} \tag{27}$$

$$\mathrm{D} = \mathbf{TD'} \tag{28}$$

The local stiffness matrix **K′** is calculated as: $K' = \frac{A_s E}{l_s}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}$ (29)

The displacement matrix **D'** is defined as:

$$\mathbf{D}' = \begin{bmatrix} d_{f1_x} \\ d_{f1_y} \\ d_{f1_z} \\ d_{f1\,origin_x} \\ d_{f1\,origin_y} \\ d_{f1\,origin\,z} \\ \cdots \\ \cdots \\ \cdots \\ d_{fn_x} \\ d_{fny} \\ d_{fnz} \\ d_{fn_{origin\,z}} \\ d_{fn_{origin\,z}} \\ d_{fn_{origin\,z}} \end{bmatrix} \tag{30}$$

To transfer the local displacement and stiffness matrices to the global coordinates, a transformation matrix **T** is used, where T is defined as:

$$\mathbf{T} = \begin{bmatrix} \lambda_{1x} & \lambda_{1y} & \lambda_{1z} & 0 & 0 & 0 & \cdots\ \cdots & \lambda_{nx} & \lambda_{nx} & \lambda_{nx} & 0 & 0 & 0 \\ 0 & 0 & 0 & \lambda_{1x} & \lambda_{1y} & \lambda_{1z} & \cdots\ \cdots & 0 & 0 & 0 & \lambda_{nx} & \lambda_{nx} & \lambda_{nx} \end{bmatrix} \tag{31}$$

Where,

$$\lambda_x = \cos\theta_x = \frac{x_{F_i} - x_{F_{origin}}}{l_s} \tag{32}$$

$$\lambda_x = \cos\theta_x = \frac{x_{F_i} - x_{F_{origin}}}{l_s} \tag{33}$$

$$\lambda_z = \cos\theta_z = \frac{z_{F_i} - z_{F_{origin}}}{l_s} \tag{34}$$

$$l_s = [\sqrt{(x_{F_i} - x_{F_{origin}})^2 + (y_{F_i} - y_{F_{origin}})^2 + (z_{F_i} - z_{F_{origin}})^2}] \tag{35}$$

Finally the F_{axial} matrix is calculated as:

$$F_{axial} = \mathbf{T^T K' T\ TD'} \tag{36}$$

After computation of matrices F_{axial} and D, all the axial forces and displacements of each strut can be obtained.

2.3 Modal Analysis of selected lattice structures

Modal analysis is used to find the Eigen frequencies and mode shapes of the lattice structure to study the resonance condition. By using modal analysis it can be expect kind of shape and their nature to vibrate naturally will be slow or fast. While doing modal analysis it necessary to check the frequencies of the structures are coincides with natural frequency of the structure, if they are coincides then structure may even fail. By knowing the modal frequency of a component it is able to design them to avoid specific ranges to prevent resonance. Results of modal analysis as follows:

Table 1: Summary of simulated modal frequencies of lattice structure

Sr. No.	Modal Frequency	M1	M2	M3	M4	M5	M6
1	First	107.49	804.24	484	355.84	934.73	400.89
2	Second	107.77	805.6	532.43	356.15	940.94	401.19
3	Third	422.25	828.91	1046	821.41	1188.4	850.95
4	Fourth	758.61	1623.7	1241.2	845.8	1797.8	1265.1
5	Fifth	775.33	2081.2	1667.8	1380.5	1827.4	1664.4
6	Sixth	779.73	2084.5	1685	1381.8	1851.9	1667.9

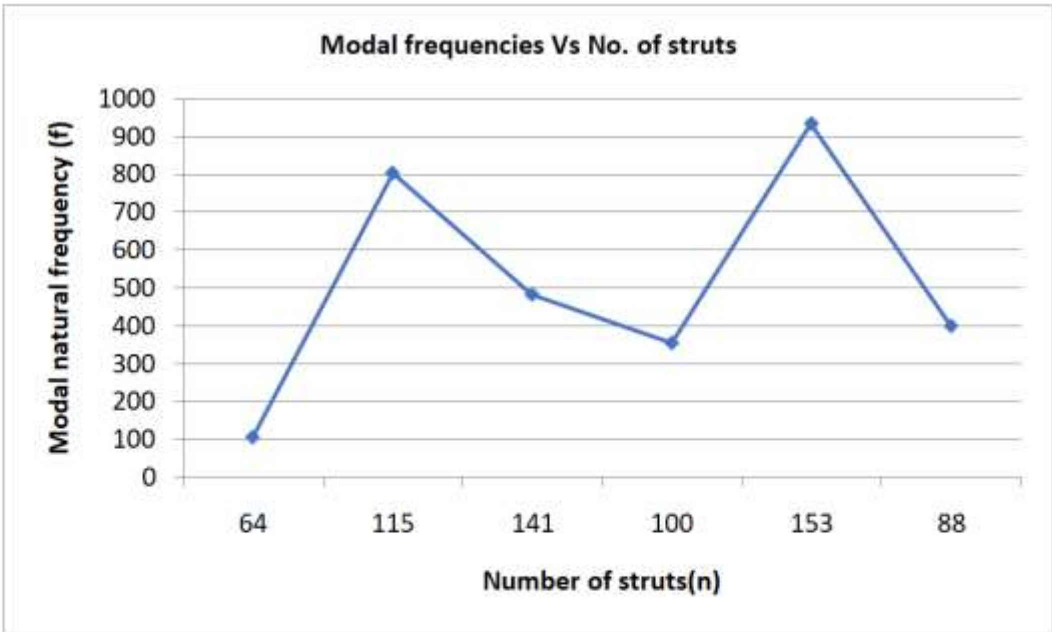


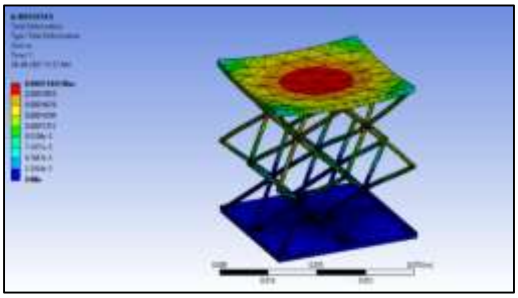
Figure 7: Modal frequencies Vs total number of struts

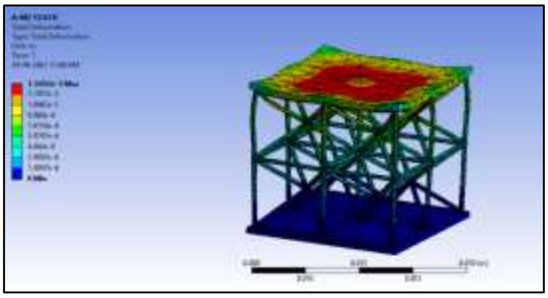
From the summery of all model frequencies of lattice structure, it is observed that model 1 has lowest natural frequency 107.49 H_Z and model 5 has highest natural frequency 934.73H_Z. Also model 1 has lowest number of struts 64 and model 5 has maximum number struts 153. So it can be right to say that modal frequency can be varied by varying number of struts. The relation between modal frequency and total number of struts is shown in fig. 3.17. These frequencies show the realistic behaviour of model under loading conditions, when the frequency of external excitation is matches with these modal frequencies. Then resonance will occur and amplitude of system vibration increases suddenly. So at these frequencies, model changes its behaviour. So it should be avoided by limiting the external load excitation on the system.

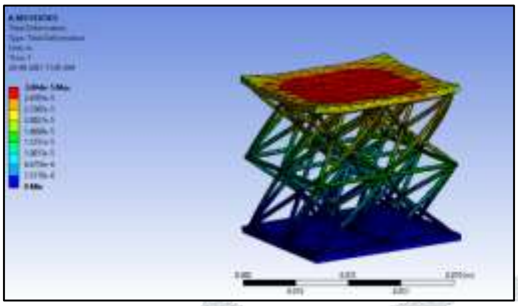
2.4 FEA analysis to estimate deformation and equivalent stress

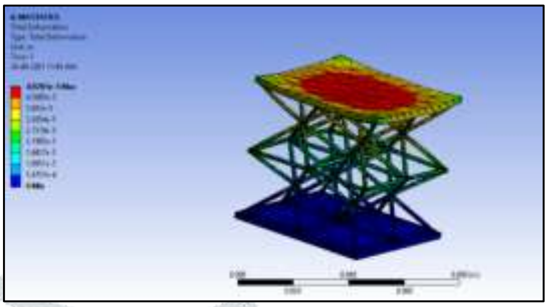
Polylactic Acid (PLA) material is selected for lattice fabrication. The boundary conditions is taken as the bottom nine nodes of all models are fixed and load is applied at each top nine nodes of all six models in vertically downward direction. By considering small DC motor which has mass<0.5 kg, design load is selected is 5N

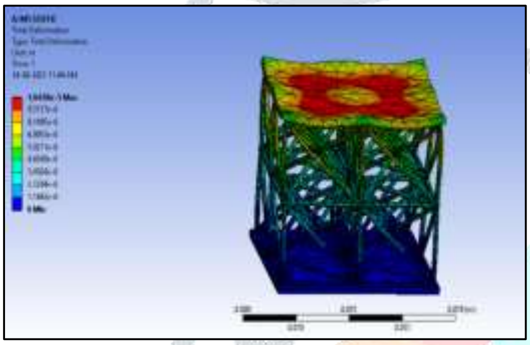
a) Deformation of lattice models


Model 1


Model 2


Model 3


Model 4


Model 5

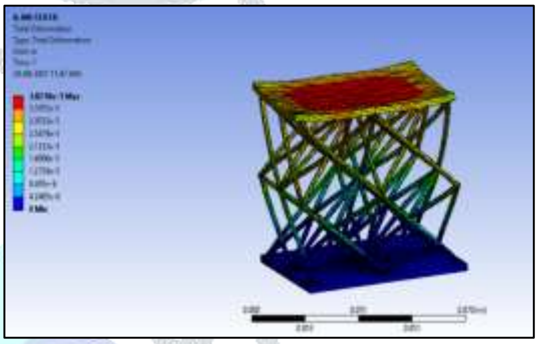
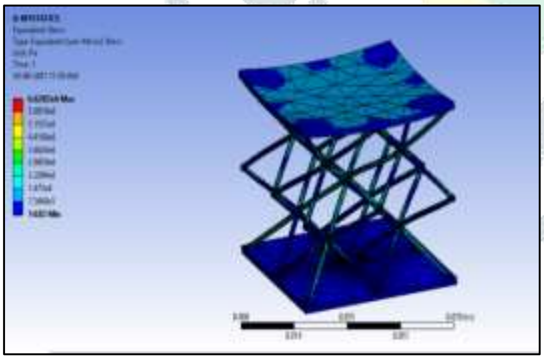
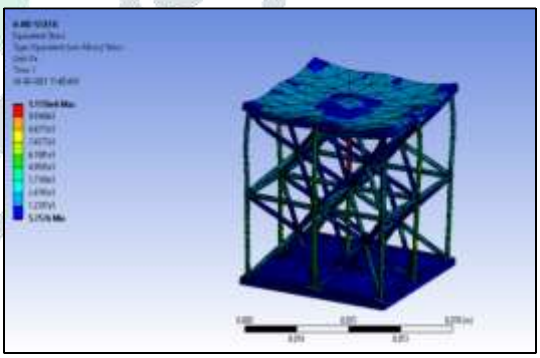
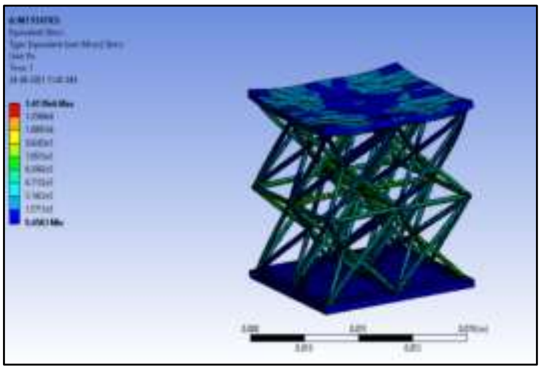

Model 6

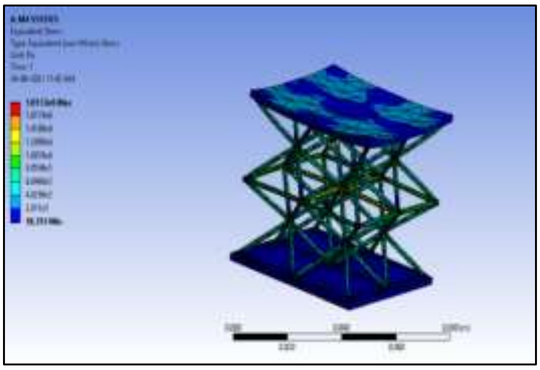
Figure 8: FEA simulation to eastimate the deformation of lattice structures

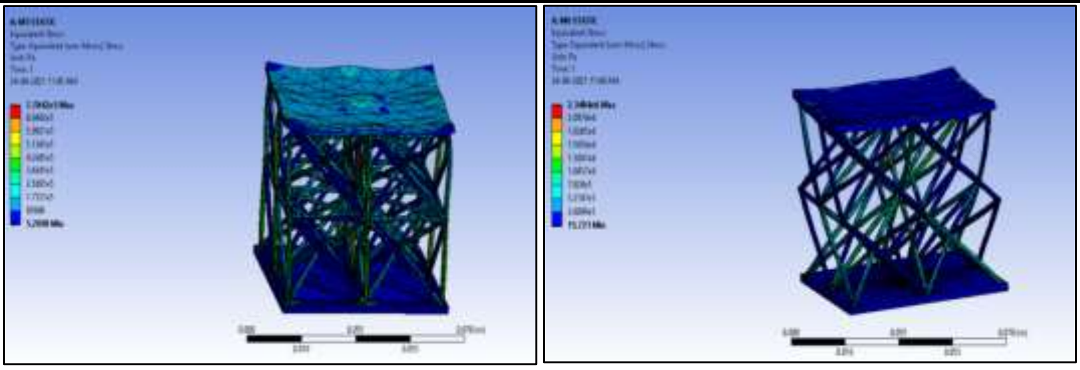
b) Von-mises stress of lattice structure


Model 1


Model 2


Model 3


Model 4



Model 5 Model 6

Figure 9: FEA simulation to eastimate the von-misses stress of lattice structures

Table 2: Summary of simulated deformation and stress values for all six lattice models

Model No.	Deformation (mm)	Equivalent Stress (mPa)
1	**0.2144 - δ_{max}**	**6.6282- σ_{max}**
2	0.01345	1.1156
3	0.0301	1.4139
4	0.0413	1.8133
5	**0.0105 - δ_{min}**	**0.7704 - σ_{min}**
6	0.0382	2.3484

It is observed that model 1 has highest deformation of 0.2144 mm and model 5 has lowest deformation of 0.0105mm. Also in model 1 has maximum equivalent stress of 6.6282mPa and model 5 has lowest of 0.7704mPa. So model 5 is more rigid as compared to all remaining models

III. FABRICATION OF LATTICE STRUCTURE

Among six models model 1 is fabricated by using fused deposition modeling (FDM) additively manufactured printer where model is created by adding layer by layer. This process is selected because of high degree of design freedom. PLA is material is selected because of its strength and stiffness [1].

Table 3: Material properties of PLA (Polylactic Acid)

Material Properties	
Young's Modulus	3.45 GPa
Poisons Ratio	0.39
Density	1250 $^{kg}/_{m^3}$

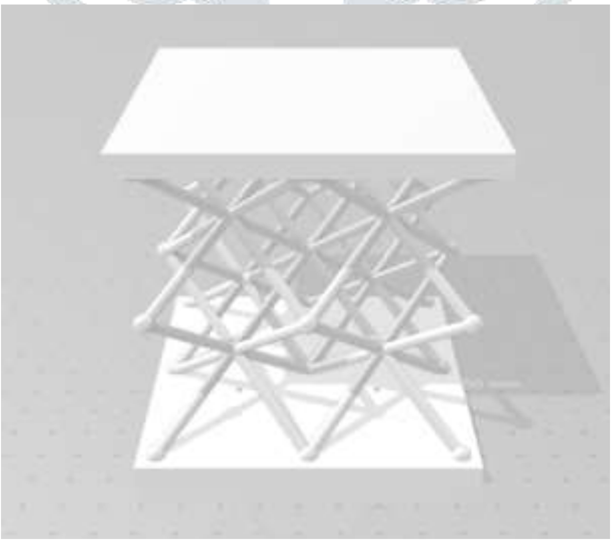


Figure 10: STL file model for 3D printing

This STL file of model is put into cura software where it is generate G-codes for layer by layer manufacturing. WOL 3D's YUVA V2 printer is used to make this model and it takes around 10 hours to build with100% fill density.

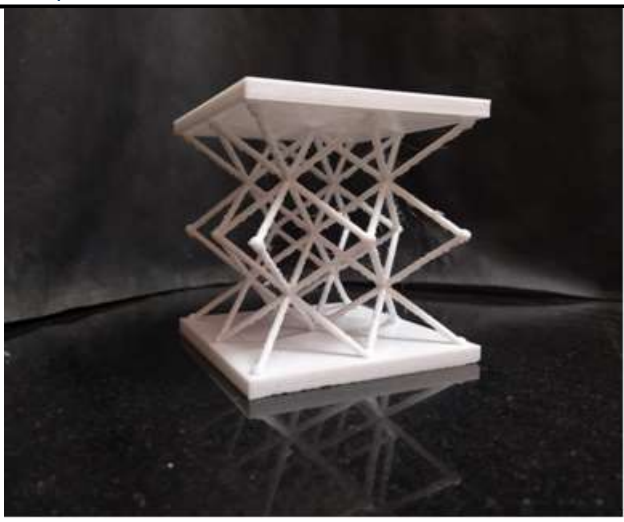


Figure 11: 3D printed model

IV. CONCLUSION

From this study it is observed that out of six models model 5 has lowest deformation and equivalent von-misses stress at the design load so it will absorb more vibration energy. Also among six models, a model 5 has highest natural frequency because of maximum number of struts. So it can be say that natural frequency can be adjusted by varying the number of struts of the lattice structure. Also this strut-based lattice structures are

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