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To cite this article: M S Azmi *et al* 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **788** 012008

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Investigation on the static and dynamic behavior of BCC lattice structure with quatrefoil node manufactured using fused deposition modelling additive manufacturing

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Abstract. This paper presents a vibration characteristics of BCC lattice structure with quatrefoil node which has been made using the fused deposition modelling (FDM) additive manufacturing (AM) technique. By conducting vibration testing, the effect of strut diameter design parameter on the natural frequency values of the BCC lattice structure with quatrefoil node is investigated. The results showed that the natural frequency values of the lattice structure material can be greatly affected by the strut diameter sizes due to increase in stiffness as the strut diameter increases. The mathematical equation is also derived to calculate the total area moments of inertia of the lattice structure model and the validity of this developed model is shown through comparison of the results with experimental work of the three-point bending test which shown similar increase trend. Thus, this study provides information on the influence of strut diameter design parameter on its vibration characteristic.

Keywords. Vibration analysis; Flexural bending; BCC lattice structure; Additive manufacturing; fused deposition modelling

1. Introduction

Nowadays, with an increasing need for energy conservation and environmental protection much effort has been expended by researchers to pursue high energy efficiency in motor and engine design and manufacture. Since then, many new vehicles with better technology for fuel saving have been developed [1-2]. However, to get high energy efficiency studies cannot be limited only to improvements to engines and motors but need to be directed to reducing the overall weight of the associated vehicles. This more holistic method can be highly beneficial, especially in aeronautics industry where lower airplane weight can make a huge difference to fuel consumption and emissions, especially during takeoff and landing [3-4]. Due to this reason, many vehicle manufacturers are now using lighter weight materials for airframe structures, especially composite materials such as carbon fiber reinforced polymers (CFRP) and glass fiber reinforced polymers (GFRP) [5-6]. The use of these materials is generally favored because not only the overall weight of the vehicles or devices can be reduced, but the mechanical performance of these materials can be as high, or superior to, that of conventional metallic materials [7-9]. However, these forms of composite material are expensive. Fortunately, there is another method to



reduce overall weight, and this is to use lower density porous structures such as lattices. A lattice is a cellular structure, and this can have certain advantages. Besides being lightweight, it has an intrinsically periodic configuration which provides uniform mechanical performance throughout the whole structure [10]. Lattice structures also have been proven to have a generally high strength to weight ratio and excellent energy absorbance performance, both of which are excellent for crash protection applications [10-14]. However, current uses of lattice structures are limited to minor parts, especially in strongly dynamic applications. This is due to a general lack of available studies of the dynamic performance of lattice structures. There are very obviously many different potential sources of vibration in structural dynamics applications and so research into the dynamic behavior of lattices is important if they are to be used as substitutes for bulk materials in highly loaded structures. For this reason, Lou et al., (2014) and Li et al., (2015) explored the dynamic performance of lattice structures made from a hot-pressing process and studied the effects of damage on their dynamic behavior [15-16]. Another more recent study by Syam et al., (2018) and Elmadhi et al., (2017) investigated the dynamic performance of strut-based lattice structures made by utilizing different manufacturing techniques within the additive manufacturing process (AM) [17-18]. Using AM to fabricate lattice structures allows a higher degree of complexity to be incorporated within lattice designs and therefore possibly to reduce their size. However, these studies only focused on vibration isolation applications. Therefore, this new study presents a vibration analysis of a BCC lattice structure with quatrefoil node which has been made by using fused deposition modeling (FDM) AM. Later, the lattices will be applied to an arm body-part of a lightweight automated device. This study incorporates investigations into the effects of the strut diameter size design parameter on the natural frequency of BCC lattice structure with quatrefoil node.

2. Lattice structure design

2.1. Lattice structure design

There are many options at the design stage that can be implemented by manipulating key design parameters (number, position and diameter of struts or nodes, for example) or by simply selecting different topological designs of the lattice structure. If there is no constrained set there will be too many feasible designs that can be proposed and realized by using additive manufacturing. In the example of Syam et al., (2018) there is a design of a lattice unit cell which is obtained by fixing the number of nodes and varying the strut configuration while maintaining the symmetry, and from these six models are proposed [17]. On the other hand, Elmadhi et al., (2017) designed a lattice structure based on a developed equation design tool which allowed the construction of a range of lattice unit cells with predefined volume fractions [18]. Therefore, lattice structures in this study were constrained using other design parameters, the strut diameter size on the octahedral cell. This is also known as body-centered-cubic (BCC) cell topological design, as shown in figure 1, where $\emptyset$ is the strut diameter size. The computer aided design (CAD) drawing was done by using the solid modelling SolidWorks software. The details of the design of the BCC unit cells are described in a previous article [19]. The BCC unit cells were tessellated into 32 layers in the length, 6 layers in the width and 3 layers in the height, to form a lattice structure bar with dimensions of 160 mm x 30 mm x 15 mm. Figure 2 shows an illustration of the lattice structure bar from the SolidWorks software.

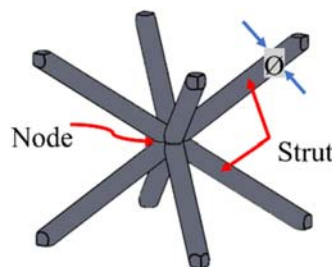


Figure 1. Illustration of BCC lattice's one-unit cell.

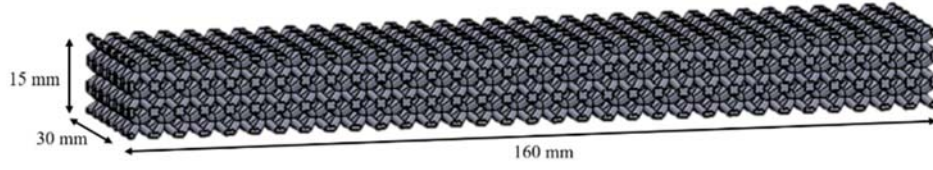


Figure 2. Illustration of a BCC lattice structure bar sample.

2.2. Derivation of total area moments of inertia

Study by Syam et al. (2018) introduces design parameter variables which are the total area moment of inertia and also mass as a straight forward method to estimate both natural frequency and mechanical strength [17]. However, the models of lattice structures proposed in their study are consist of sphere-shaped nodes which is different from the BCC model proposed in the present study which has quatrefoil shaped nodes. Hence, the derivation of total area moment of inertia in present paper is concentrating on to calculate the total area moment of inertia in the quatrefoil nodes. The final derivation will consist of summation of total area moment of inertia in the struts and nodes where derivation for the struts will be directly adapted from Syam et al. (2018) as the design of the struts of BCC lattice in this study are similar to the BCC lattice model in their study which has cylindrical shaped. Hence, the following mathematical modelling are developed to calculate the total area moment of inertia for BCC quatrefoil node design in this study using similar computational method namely parallel axis theorem and rotation of axes. For convenience, the total area moment of inertia about each axis in Cartesian coordinates are as follows:

$$I_x = \sum_{i=1}^{N_{strut}} I_{xsi} + \sum_{j=1}^{N_{node}} I_{xnj} \quad (1)$$

$$I_y = \sum_{i=1}^{N_{strut}} I_{ysi} + \sum_{j=1}^{N_{node}} I_{ynj} \quad (2)$$

$$I_z = \sum_{i=1}^{N_{strut}} I_{zsi} + \sum_{j=1}^{N_{node}} I_{znj} \quad (3)$$

where I_{xsi} , I_{ysi} and I_{zsi} are the area moments of inertia of the i -th strut whilst I_{xnj} , I_{ynj} and I_{znj} are the area moments of inertia of the j -th node in the lattice structure about the x , y and z axes respectively, and where i and j are the indices for the struts and nodes respectively. The total numbers of struts and nodes are denoted by N_{strut} and N_{node} respectively. The derivation to calculate the total area moment of inertia in the struts used cylindrical shape. Meanwhile, for the quatrefoil nodes' derivation, figure 3 shows the illustration of ellipses and square in the nodes. For convenience a node in the lattice structure is assumed to be an area of two ellipses with radius a and b minus a square with width equals to approximately $2b$, as illustrated in figure 3. As the BCC topological design has an isotropic configuration the area of the node is the same for the x , y and z directions. From the parallel axis theorem, the moments of inertia for every node are calculated as follows:

$$I_{xnj} = I'_{xnj} + [2\pi ab - (2b)^2]x_n^2 \quad (4)$$

$$I_{ynj} = I'_{ynj} + [2\pi ab - (2b)^2]y_n^2 \quad (5)$$

$$I_{znj} = I'_{znj} + [2\pi ab - (2b)^2]z_n^2 \quad (6)$$

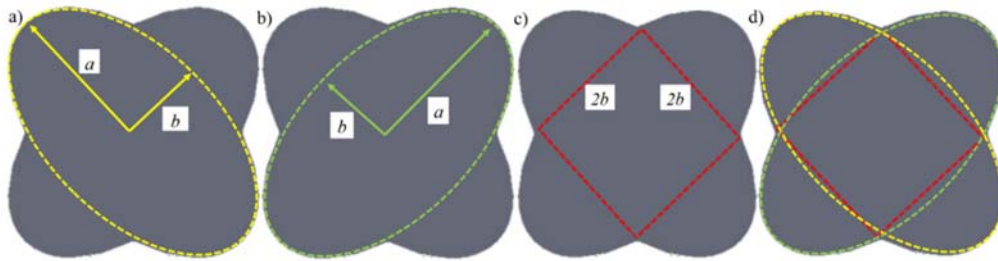


Figure 3. a) first ellipse, b) second ellipse, c) square, d) merged node illustrations.

where x_n , y_n and z_n are the distances between the node's centroid of a unit cell. The first integral in equations (4) - (6) represents the moment of inertia of the area with respect to the centroidal axis. The second integral in equations (4) - (6) is equal to the total area. These equations (4) – (6) express that the moment of inertia of an area with respect to an arbitrary axis is equal to the moment of inertia of the area with respect to the centroidal axis parallel to the real axis plus the product of the area multiplied by the distance between the two axes. To calculate I'_{xnj} , I'_{ynj} and I'_{znj} , the rotation of axes is used to rotate the square and ellipses' axes at the nodes. Hence the I'_{xnj} , and I'_{ynj} become

$$I'_{xnj} = [\cos \theta_j]^2 I_{xnj \text{ ellipse}} + [\sin \theta_j]^2 I_{ynj \text{ ellipse}} + [\cos \theta_j]^2 I_{ynj \text{ ellipse}} + [\sin \theta_j]^2 I_{xnj \text{ ellipse}} - [\cos \theta_j]^2 I_{xnj \text{ square}} + [\sin \theta_j]^2 I_{ynj \text{ square}} \quad (7)$$

$$I'_{ynj} = [\cos \theta_j]^2 I_{ynj \text{ ellipse}} + [\sin \theta_j]^2 I_{xnj \text{ ellipse}} + [\cos \theta_j]^2 I_{xnj \text{ ellipse}} + [\sin \theta_j]^2 I_{ynj \text{ ellipse}} - [\cos \theta_j]^2 I_{xnj \text{ square}} + [\sin \theta_j]^2 I_{ynj \text{ square}} \quad (8)$$

where the equation of moment of inertia of ellipses and square are obtained from [20]. By inserting the respective equations into equation (7) and (8) and with substitutions, therefore, equations (1) – (3) become

$$I_x = \sum_{i=1}^{N_{strut}} [\cos \theta_i]^2 \frac{d_s l_s^3}{12} + [\sin \theta_i]^2 \frac{d_s^3 l_s}{12} + l_s d_s x_s^2 + \sum_{j=1}^{N_{node}} [[\cos \theta_j]^2 \frac{\pi a b^3}{4} + [\sin \theta_j]^2 \frac{\pi a^3 b}{4}] + [[\cos \theta_j]^2 \frac{\pi a^3 b}{4} + [\sin \theta_j]^2 \frac{\pi a b^3}{4}] - [\cos \theta_j]^2 \frac{(2b)^4}{3} + [\sin \theta_j]^2 \frac{(2b)^4}{3} + [2\pi a b - (2b)^2] x_n^2 \quad (9)$$

$$I_y = \sum_{i=1}^{N_{strut}} [\cos \theta_i]^2 \frac{d_s^3 l_s}{12} + [\sin \theta_i]^2 \frac{d_s l_s^3}{12} + l_s d_s y_s^2 + \sum_{j=1}^{N_{node}} [[\cos \theta_j]^2 \frac{\pi a^3 b}{4} + [\sin \theta_j]^2 \frac{\pi a b^3}{4}] + [[\cos \theta_j]^2 \frac{\pi a b^3}{4} + [\sin \theta_j]^2 \frac{\pi a^3 b}{4}] - [\cos \theta_j]^2 \frac{(2b)^4}{3} + [\sin \theta_j]^2 \frac{(2b)^4}{3} + [2\pi a b - (2b)^2] y_n^2 \quad (10)$$

$$I_z = \sum_{i=1}^{N_{strut}} [\cos \theta_{xyi}]^2 \frac{d_s^3 l_s}{12} + [\sin \theta_{xyi}]^2 \frac{d_s l_s^3}{12} + l_s d_s z_s^2 + \sum_{j=1}^{N_{node}} [[\cos \theta_j]^2 \frac{\pi a^3 b}{4} + [\sin \theta_j]^2 \frac{\pi a b^3}{4}] + [[\cos \theta_j]^2 \frac{\pi a b^3}{4} + [\sin \theta_j]^2 \frac{\pi a^3 b}{4}] - [\cos \theta_j]^2 \frac{(2b)^4}{3} + [\sin \theta_j]^2 \frac{(2b)^4}{3} + [2\pi a b - (2b)^2] z_n^2 \quad (11)$$

Equations (9) – (11) will be used to calculate the total area moment of inertia of the lattice models.

2.3. Mass calculations

The masses of the lattice structure bar samples are calculated straightforwardly by multiplying the total volume of the lattice structure bar samples acquired directly from the SolidWorks software by the material density such as in equation (12). The acrylonitrile-butadiene-styrene (ABS) polymer material used to fabricate the samples in this study has density, $\rho = 1050 \text{ kg/m}^3$ noting that this material density is obtained from the safety datasheet provided by the manufacturer (3D Systems Inc.).

$$\rho = \frac{m}{v} \quad (12)$$

3. Experimental work

3.1. Sample preparation

The lattice structure bar samples were fabricated by utilizing the Cubepro machine fused deposition modeling (FDM) type additive manufacturing technology (AM) with standard print quality mode-process parameter combinations (200 μm layer resolution, strong print strength-cross print pattern) [21]. The lattice structure bar samples were fabricated from acrylonitrile-butadiene-styrene (ABS) polymer material, and more details of the printing process are given in previous articles [19, 22]. Strut diameter size was varied in three sizes which were 1.4 mm, 1.6 mm and 1.8 mm. This was done at the early design

drawing stage. The reason for selecting strut diameter sizes from 1.4 mm to 1.8 mm was because although the Cubepro machine is able to fabricate a lattice structure with a strut diameter size as low as 1.0 mm [23], the fabricated parts can be too delicate and not suitable for dynamic applications. Hence, a 1.4 mm strut diameter size was deemed to be the smallest appropriate diameter. Conversely, if the strut diameter size is fabricated to be more than 1.8 mm diameter, the fabricated part will lose its lightweight properties and will end up being too close to that of the bulk material it is intended to replace. The successfully fabricated lattice structure bar samples are shown in figure 4. Meanwhile, for the purpose of comparison, the solid ABS samples were bought in the ABS sheet form and was cut into same dimension as the lattice structures cellular material bar samples. On the other hand, the commercially available automated device's real arms were also tested.

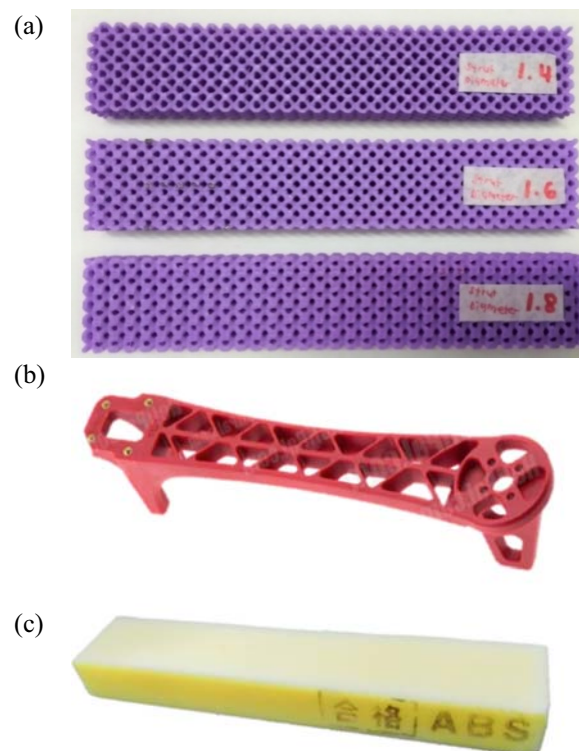


Figure 4. (a) ABS lattice structures cellular material bar sample with 1.4 mm, 1.6 mm and 1.8 mm strut diameter size (top view) (b) commercial real arm (c) solid ABS.

3.2. Experimental procedure

In this study, the modal vibration testing was conducted by utilizing an electro-dynamic shaker to excite the lattice structures cellular material bar samples in order to characterize its vibration characteristics. The actual set-up used are shown in figure 5. The experimental set-up consisted of signal generator and analyser, force sensor to monitor the input force, accelerometer vibration sensor to measure the vibration response amplitude, dynamic electro shaker and signal power amplifier. Each lattice structure cellular material bar sample was clamped at one end to the test rig as boundary condition to mimic the single-end load-bearing application. The test rig used was made of heavy mild steel to suppress the system from any unwanted ambient vibrations. The shaker excited the lattice structures cellular material bar samples at one excitation point in vertical direction via stinger attached at the free end of the bar samples while the vibration sensor was attached using wax on the samples to measure the vibration responses at 2 parallel and 4 uniformly spaced measurement points at the edge along the lattice structures cellular

material bar samples as illustrates in figure 6. The total number of measurement points were 10. During experiment, a random excitation with frequency from 0-1200 Hz range was generated by DataPhysics Quattro Ace and amplified by the signal power amplifier. The measured data were then analysed by using *SignalCalc 240 Dynamic Signal Analyzer* software provided by Dataphysics. Three sets of measurements were taken for each lattice structures cellular material bar samples to ensure the consistency of the readings taken. FRF graphs in this study is presented in terms of ratio between acceleration of steady-state response over force intensity. Vibration test was also conducted on bulk solid ABS material and commercially available automated device real arm for comparison purposes.

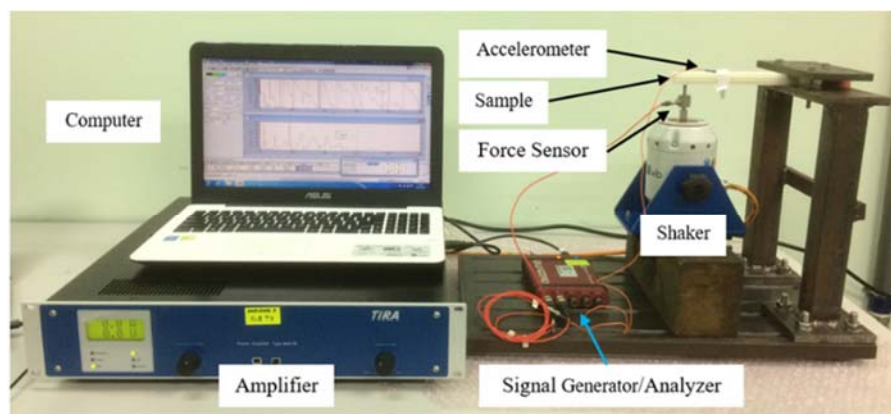


Figure 5. Experimental set-up for vibration test.

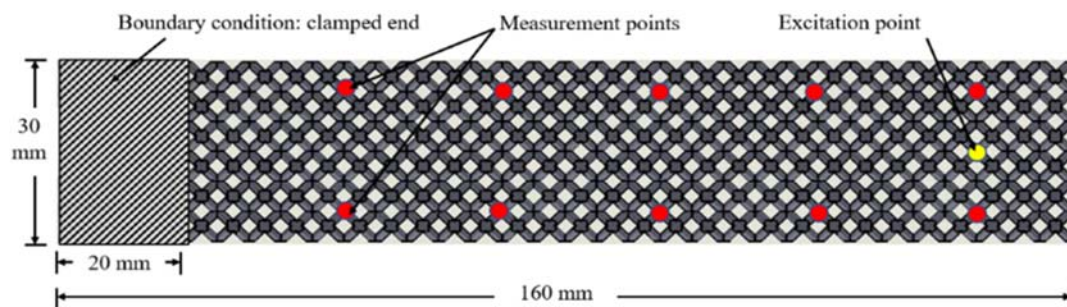


Figure 6. Boundary condition, excitation and measurement points location of the lattice structures cellular material bar sample (top view).

3.3. Bending test

Three-point bending tests were conducted to evaluate the influence of the strut diameter size design parameter on the stiffness of the lattice structure bar samples. A bending test was chosen as it was the closest form of static test that could be used to determine properly the stiffness relating to the conditions of the lattice structure bar samples when they are subjected to a transverse loading during vibration test. Similarly, bending tests were also conducted on the solid ABS and commercially available automated real device-arm for comparison purposes. The three-point bending tests were performed using an Instron 5585 machine with a test set-up for three-point bending as shown in figure 7 where the red arrows indicate rollers and loading directions. The tests were carried out by applying an axial loading force at a speed rate of 1.5 mm/min. The support span of the samples during the experiment was 120 mm. All three samples were deflected until rupture occurred in the outer surfaces of the test samples. The readings acquired from the test were recorded using the Instron BlueHill-3 software. Three tests were carried out for each strut diameter size, and for the solid ABS and real arm; noting that the total number of tests was fifteen.

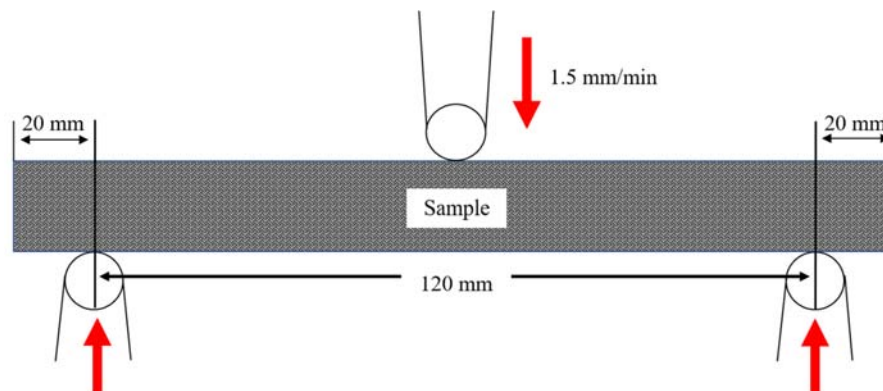


Figure 7. Schematic diagram set-up for the three-point bending test.

4. Results and discussion

4.1. Vibration analysis of the effects of the strut diameter size design parameter

In this section, the effects of strut diameter size on the natural frequencies of the lattice structure bar samples were presented. The results for all three lattice structure bar samples, the solid ABS and the commercial automated real device-arm are shown in figure 8. The natural frequencies for the first two vibration modes were extracted from the FRF graphs and tabulated in table 1. The calculated mass and total area moment of inertia in table 1 were calculated using equation (12) and equations (9) – (11) respectively.

Table 1 Vibration test results, masses and calculated area moments of inertia of the samples.

Sample	Actual Mass (g)	Calculated Mass (g)	Calculated Total Area Moment of Inertia ($\times 10^{-8} \text{ m}^4$)	Natural Frequency (Hz)	
				1st mode	2nd mode
1.4 mm $\emptyset$	22.49	24.12	6.45	279	404
1.6 mm $\emptyset$	30.30	30.01	8.04	382	527
1.8mm $\emptyset$	35.89	36.03	9.73	415	545
Solid ABS	72.94	75.60	-	858	1029
Real arm	46.26	-	-	414	654

Figure 9 shows the natural frequency values for the first two vibration modes for the lattice structure bar samples, the solid ABS and the commercial real-arm. Based on figure 9, the lattice structure cellular material bar with 1.8 mm strut diameter design parameter size exhibits 48.74% - 34.90% higher natural frequency values for the first two vibration modes as compared to sample with 1.4 mm strut diameter and 8.64% - 3.42% higher natural frequency values for the first two vibration modes as compared to sample with 1.6 mm strut diameter. Meanwhile, the solid ABS exhibits much higher natural frequency values and the commercial real arm shows almost the same first mode natural frequency but different in second mode natural frequency to that of lattice structure cellular material bar with 1.8 mm strut diameter design parameter size. This is because the lattice structure bars consist of multiple struts and nodes as compared to the real arm. During transverse loading excitation from the shaker, there are unknown interactions between the struts of one-unit cell to another unit cell especially in the higher mode of vibrations where more interactions present making it more complicated. Further investigation is needed in order to see these interactions. Furthermore, with almost identical first mode natural frequency, lattice structure with 1.8 mm strut diameter size has 22.42% lower mass as compared to the real arm. Meanwhile, the mass of lattice structure with highest strut diameter size was 50.80% lower in mass as compared to the solid ABS.

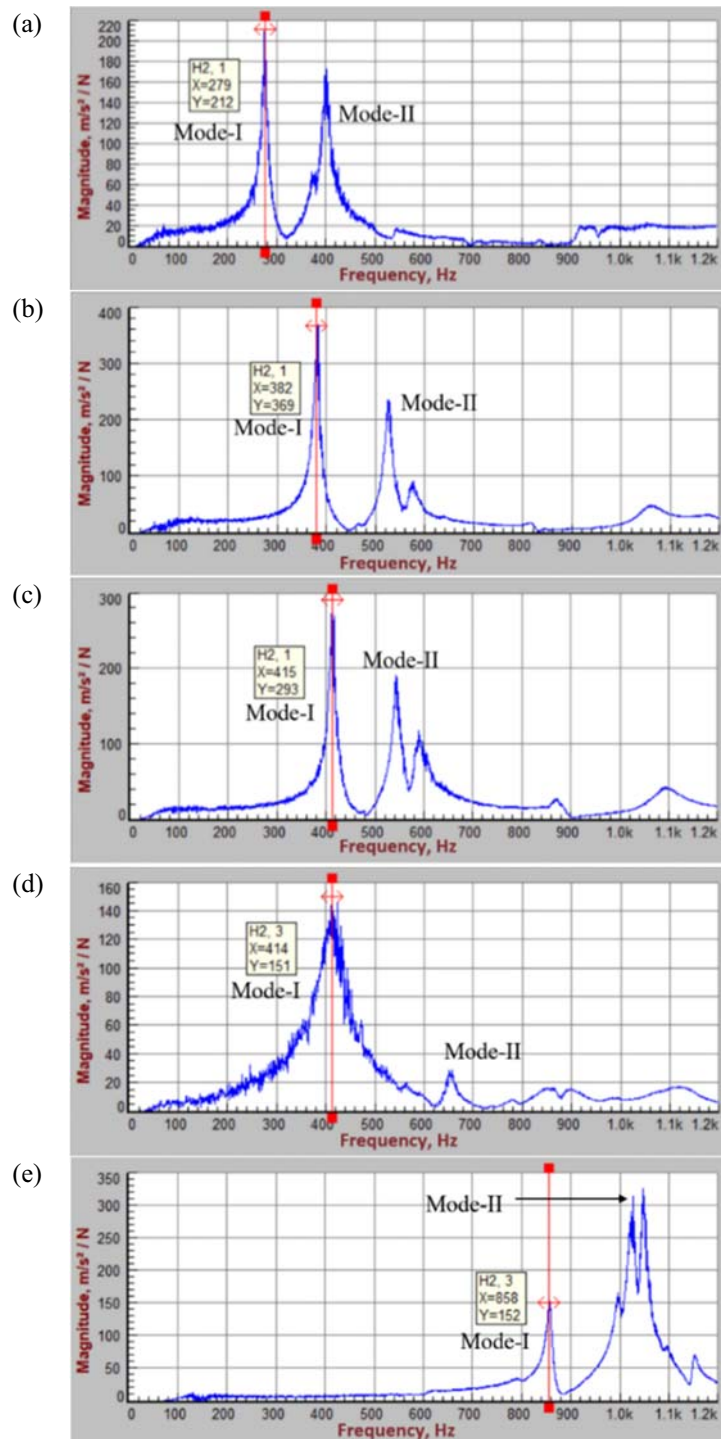


Figure 8. FRF graphs of the lattice structures cellular material bar sample with (a) 1.4 mm (b) 1.6 mm (c) 1.8 mm strut diameter size (d) commercial real arm and (e) solid ABS.

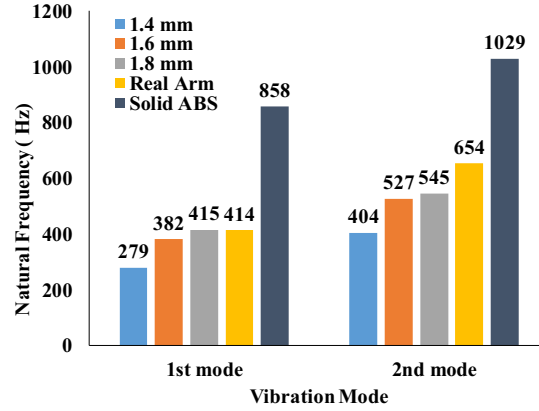


Figure 9. Natural frequency values for the first two vibration modes of the lattice structure bar samples, solid ABS and commercial real arm.

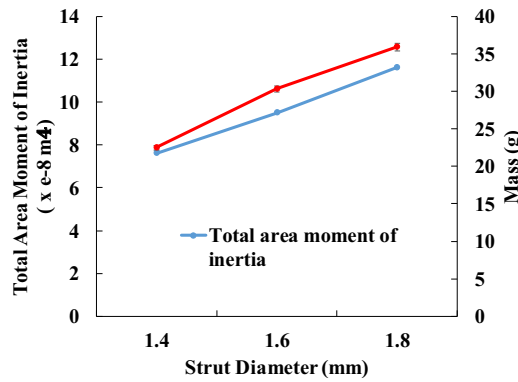


Figure 10. Effect of strut diameter size of the lattice structure bar samples on the mass and the calculated total area moment of inertia. The error bar is for the standard deviation.

Furthermore, the effect of strut diameter size of the lattice structure on the mass and the calculated total area moment of inertia are shown in figure 10. From these results, it can be clearly seen that the calculated total area moment of inertia of the lattice structures cellular material increase with the strut diameter design parameter size due to an increase in its cross-sectional area. The stiffness of the lattice structures cellular material bar samples is proportional to its total area moment of inertia. Therefore, an increase in the total area moment of inertia increases the stiffness and this will eventually result in higher natural frequency values based on equation (13).

$$\omega_n = \left(\frac{k}{m}\right)^{1/2} = \left(\frac{3EI}{ml^3}\right)^{1/2} \quad (13)$$

where ω_n is the natural frequency [Hz], k is the stiffness [N/m] m is the mass [kg], l is the length of the lattice structure bar [m] and E is the elastic modulus of ABS material. From this relationship, it is clear that the natural frequency is also proportional to the total area moment of inertia. Based on

figure 10, it also can be seen that the mass of the lattice structure cellular material bar samples increases as the strut diameter design parameter size increase. Based on equation (1), the mass of the lattice structure cellular material bar samples is supposedly inversely proportional to the natural frequency (i.e. an increase in mass will lower the natural frequency). However, the results show otherwise. Thus, this suggests that despite an increase in mass of the lattice structure when the strut diameter design parameter size increases, the stiffness component of the lattice structure cellular material bar samples is more significant compared to the mass component of the lattice structure cellular material bar samples. A similar outcome can be seen especially for the solid ABS which yields much higher natural frequency values due to its much superior stiffness as compared to the other tested bar samples.

4.2. Bending test

In this section, the results of the three-point bending tests for each strut diameter size, the solid ABS and the commercially available real-arm were averaged and then plotted on force against displacement graphs, as shown in figure 11 where the sharp drop in the plots indicate that rupture has occurred in the outer surfaces of the samples.

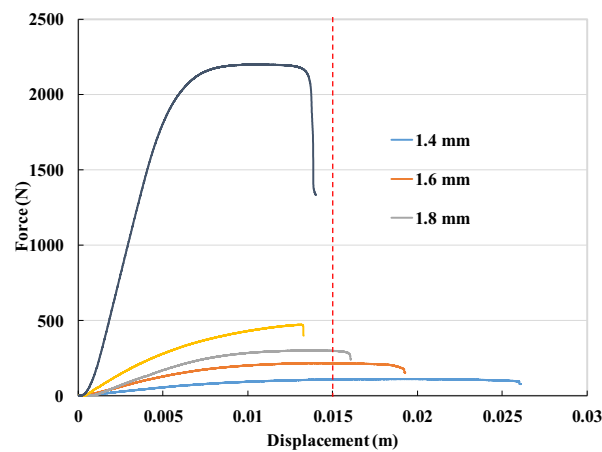


Figure 11. Force-displacement curves for all samples from the three-point bending tests.

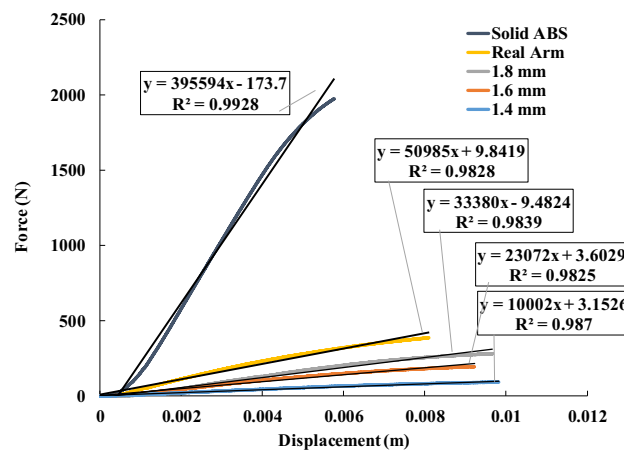


Figure 12. Linear part of the force-displacement curves for all samples from the three-point bending tests.

The responses in the bending tests were composed of a clear linear elastic region followed by a nonlinear region. The nonlinearity was attributed to plasticity and kinematic nonlinearity upon unloading [24]. Based on figure 11, it can be seen that all the lattice bar samples were much more flexible with displacements of more than 15 mm before rupture was seen to occur. On the other hand, both the solid ABS and real-arm samples yielded less than 15 mm displacement. Furthermore, only the Solid ABS was able to resist a maximum load of more than 500 N of loading force. Nevertheless, all samples resisted a sufficiently high maximum load capability making them feasible enough for use as load-bearing body parts for an automated device. The averaged maximum loads and displacements at maximum load for all samples are tabulated in table 2. Meanwhile, figure 12 shows the cut-up linear part of figure 11 that is used to obtain the stiffness of the samples by calculating the gradients of the graphs. The calculated values are tabulated in table 2. The values show that increase in the strut diameter design parameter size to 1.8 mm will significantly increase the stiffness of the lattice structures cellular material by up to 44.69% and 233.80% as compared to that of 1.6 mm and 1.4 mm strut diameter design parameter size respectively. Meanwhile, the solid ABS shows much higher bending stiffness at 395.59 kN/m followed by the real arm at 50.99 kN/m. Hence, it can be concluded that the higher strut diameter design parameter size will result in a higher stiffness value. This result is aligned with the study conducted by Ushijima et al. (2011) which predicted that specific stiffness of BCC lattice structures cellular material increases with increasing strut diameter design parameter over cell size [25].

Table 2 Three-point bending test results.

Sample	Maximum Load (N)	Displacement at Maximum Load (mm)	Stiffness (kN/m)
1.4 mm Ø	113.68	19.57	10.00
1.6 mm Ø	219.16	14.25	23.07
1.8 mm Ø	302.67	13.78	33.38
Solid ABS	2207.20	13.36	395.59
Real arm	477.33	13.69	50.99

5. Conclusions

This paper has proposed a static and dynamic study for BCC lattice structure with quatrefoil shaped node that was fabricated using FDM AM. The influence of strut diameter on stiffness and natural frequency of lattice model have been discussed. The findings were that the natural frequency values increase as the strut diameter size increased. This was due to a higher value of the total area moment of inertia as the strut diameter size increased; this eventually resulted in a higher stiffness of the lattice structure bar sample. To prove this statement three-point bending tests were conducted and the findings from these tests showed agreeable results. The highest strut diameter size yields the highest bending stiffness values. It was also shown that the stiffness components of the lattice bar samples were more significant than the mass components. In addition, vibration and three-point bending tests were also conducted on a solid ABS and commercial real-arm. Both the solid ABS and the real-arm exhibited high natural frequency values and significantly higher bending stiffness values respectively. However, it should be noted that both the solid ABS and the real-arm were larger in mass. Nevertheless, all the lattice structure bar samples showed promising results and generally high potential for use in certain dynamic load bearing body parts in automated device applications.

Acknowledgement

The authors would like to gratefully acknowledge the Ministry of Education Malaysia (MoE) for financial support through a research grant [Grant No. FRGS/1/2016/TK03/FKM-CARE/F00316] and the Universiti Teknikal Malaysia Melaka (UTeM) for providing all the equipment used for fabrication and testing of the specimens.

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