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# Passive vibration attenuation: a comparison study

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# Passive vibration attenuation: a comparison study

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**Abstract.** A comparison study between two different approaches for passive vibration attenuation of solar array had been presented. A powerful finite element software "Ansys" was used to perform this study. Finite element models of a cantilever rectangular aluminum plate are created in order to model the satellite solar array. Stiffeners and circular patches (passive masses) are used separately in order to attenuate the vibration of the plate. The output results of the mode shapes and the FRF of the plate are investigated in order to figure out the best and adequate vibration attenuation technique for satellite solar array, the results showed that using stiffeners have a better effect on vibration attenuation more than using circular patches. A further investigation for vibration attenuation using stiffener is conducted to study the effect of changing the aspect ratio of stiffener.

## 1. Introduction

The control of Noise and vibration has a significant importance in nearly all fields and branches of engineering, especially in the aerospace applications the vibration control is regarded as a basic design issue. In space applications many sources of vibration "during launch phase and in-orbit operations such as the maneuver of the satellite, and the adjustment / movement of payload" affects all parts of satellite [1]. One of the main goals and objectives that must be achieved during the process of satellite structure design is to reduce the mass of the satellite structure as maximum as possible in order to increase the payload fraction and reduce the satellite launching cost, and hence some satellite components such as the solar array has a large size and a relatively low weight / stiffness so they are subjected to several sources of vibration [2]. For these issues the understanding of vibration behavior in satellite solar array and the best ways to govern, regulate or deal with all sources of vibration are paid a great attention in order to reduce or eliminate it by altering the design and /or designing a suitable control mechanism.

In general there are different types or methods for vibration attenuation of space structure; as passive, active, and hybrid vibration control techniques. The energy of vibration in passive damping technique is dissipated and / or controlled by a damping element (mass for example) but without feedback capability, while in active damping technique an equal force to the external excitation force is applied in the opposite direction by an actuating element as a piezoelectric actuator, finally the hybrid damping technique combines both passive and active elements for vibration attenuation [3].

The passive vibration attenuation technique is the most suitable technique for space application because it is simple, requires no power and also can be used to suppress a wide range of mechanical vibrations. Taking into consideration that the two main designing issues in passive damping techniques is to maintain the weight of the damping element as minimum as possible and maximizing the vibration damping as maximum as possible [4].

M. Ansari [5], studied the effect of the shape of the passive masses (patches) that are used to attenuate the vibration of light weight structures. It was found that the optimal shape of added masses that gives higher damping is circular. Circular patches gave 10% more damping ratio when compared with square ones.

E. Askar [1], studied the effect of using circular patches as passive masses to attenuate a plate vibration. By optimizing the locations of the circular patches using Ansys workbench to find the optimal locations that give maximum vibration attenuation, it was found that the amplitude of the plate frequency response function (FRF) was minimized by 65% and 81% and the resonant frequencies were maximized by 19% and 7% for the first and second bending modes, respectively.

S. Jafarpour [6], used both conventional and super finite element methods to study the vibration of stiffened plates, both the Mindlin plate and Timoshenko beam theories were used in order to formulate the plate and stiffeners equation of motion respectively. The proposed method helps to model any configuration of plate and stiffeners. It was proven that the fundamental natural frequency value increases as the stiffeners number is increased but at a certain extent the rate of increase in natural frequency decreases with adding more stiffeners because of the effect of added mass, and it was also found that the maximum fundamental frequency is obtained when the orientation angle of stiffener is 80°.

Amar N. [7], studied the free vibration characteristics of stiffened plates. The study concluded that the square plate (a/b = 1) has the maximum fundamental frequency followed by the rectangular long/narrow plates (a/b = 1.5 and 2) also the clamped boundary condition shows its superiority to the simply supported one. It was proven that the increase in the number of stiffeners to a certain value leads to the increase in the fundamental frequency, also the study figured out that when the value of stiffener depth to plate thickness is increased the fundamental frequency increases.

C. KUMAR [8], studied the forced vibration characteristics of a stiffened plate with a various arrangement of stiffeners and different boundary condition, it was proven that the natural frequency value is maximum as the concentrated load position coincides with the location of the stiffener, also the Natural frequencies in the case of biaxial loading condition is higher compared to axial loading condition and The frequency parameters increase by increasing the restraint at the edges according to the boundary condition.

## 2. Finite element model creation and verification

Using ANSYS workbench package version R15.0, The solar array panel is modeled by an isotropic aluminum rectangular plate with the same characteristics and boundary conditions of the FE model introduced in [1] as shown in both table 1 and figure 1.

**Table 1.** FE model Specification.

| | Property | Value |
|---|---|---|
| Mechanical properties of the plate | Material | Aluminum 5056 |
| | Density ( $kg/m^3$) | 2660 |
| | Young's modulus E (GPa) | 70 |
| | Poisson's ratio | 0.33 |
| Plate dimensions | Length (mm) | 500 |
| | Width (mm) | 200 |
| | Thickness (mm) | 1.2 |
| Number of model elements and nodes | Number of elements | 4036 |
| | Number of nodes | 28962 |

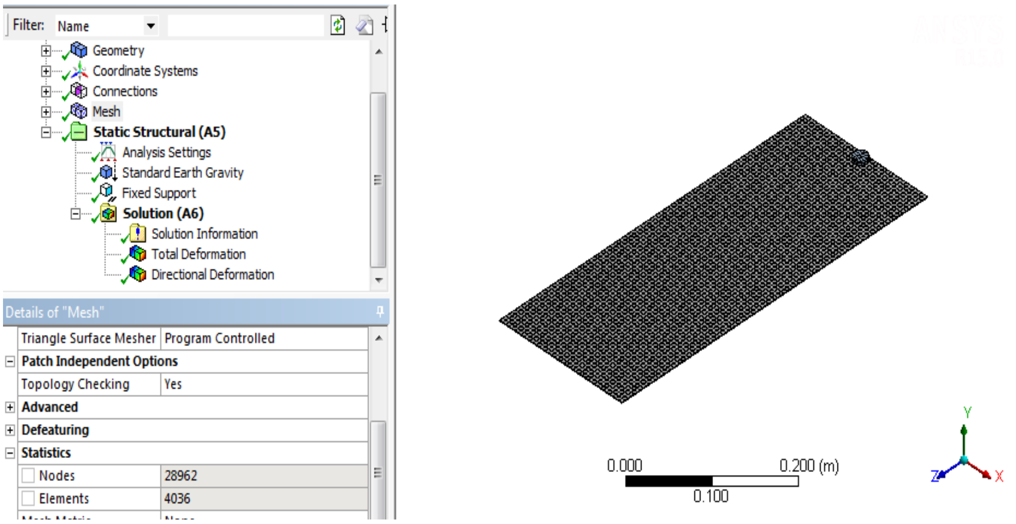


**Figure 1.** Ansys FE model for the aluminum plate.

A transverse excitation force of 1 N with sine sweep signal varies from 0 to 200 Hz is applied at the plate structure In order to study the plate FRF. The created FE model is verified by comparing the output results from Static structural analysis, Modal analysis and Harmonic response with the published results introduced in [1]. The comparison showed that the resulting values are almost the same as shown in table 2 and table 3.

**Table 2.** Maximum deformation comparison.

| | [1] | Obtained result |
|---|---|---|
| Max. deformation (mm) | 25.781 | 25.781 |

**Table 3.** First six mode shapes comparison.

| Mode shape number | Mode shape type | Natural frequency (Hz) | |
|---|---|---|---|
| | | [1] | Obtained result |
| 1st | Bending | 3.8 | 3.7681 |
| 2nd | Torsional | 21.1 | 21.091 |
| 3rd | Bending | 23.9 | 23.927 |
| 4th | Torsional | 67.5 | 67.529 |
| 5th | Bending | 67.7 | 67.729 |
| 6th | Torsional | 125.8 | 125.79 |

## 3. Results and discussion

By using the verified FE model, the vibration attenuation is conducted by adding the aluminum circular patches with optimal diameter and locations as introduced in [1], then instead of patches an aluminum rectangular cross section stiffeners are used to attenuate the vibration. Four different FE models for plate with different stiffeners number and cross sections are created in order to select the best stiffeners arrangement that produces best vibration attenuation. In order to have a reasonable comparison all created

FE models are made from the same material and have almost the same weight. The arrangement of plate models with circular patches and stiffeners is illustrated in the following tables and figures.

**Table 4.** Plate with aluminum patches/stiffeners characteristics.

| Property | Plate with patches | Plate with stiffeners | | | |
|---|---|---|---|---|---|
| | | 1 stiffener | 2 stiffeners | 3 stiffeners | 3 stiffeners |
| Mass (kg) | 0.36312 | 0.36326 | 0.36355 | 0.36724 | 0.36355 |
| Material | Aluminum (with the same properties as given in table 1) | | | | |

**Table 5.** Optimal patches diameters and locations.

| Patch number | Diameter (mm) | Center location (mm) | |
|---|---|---|---|
| | | X | Y |
| 1 | 44 | 43.887 | 150 |
| 2 | 44 | 64.01 | 100 |
| 3 | 44 | 81.397 | 50 |
| 4 | 44 | 184.47 | 150 |
| 5 | 44 | 144.14 | 100 |
| 6 | 44 | 200.8 | 50 |

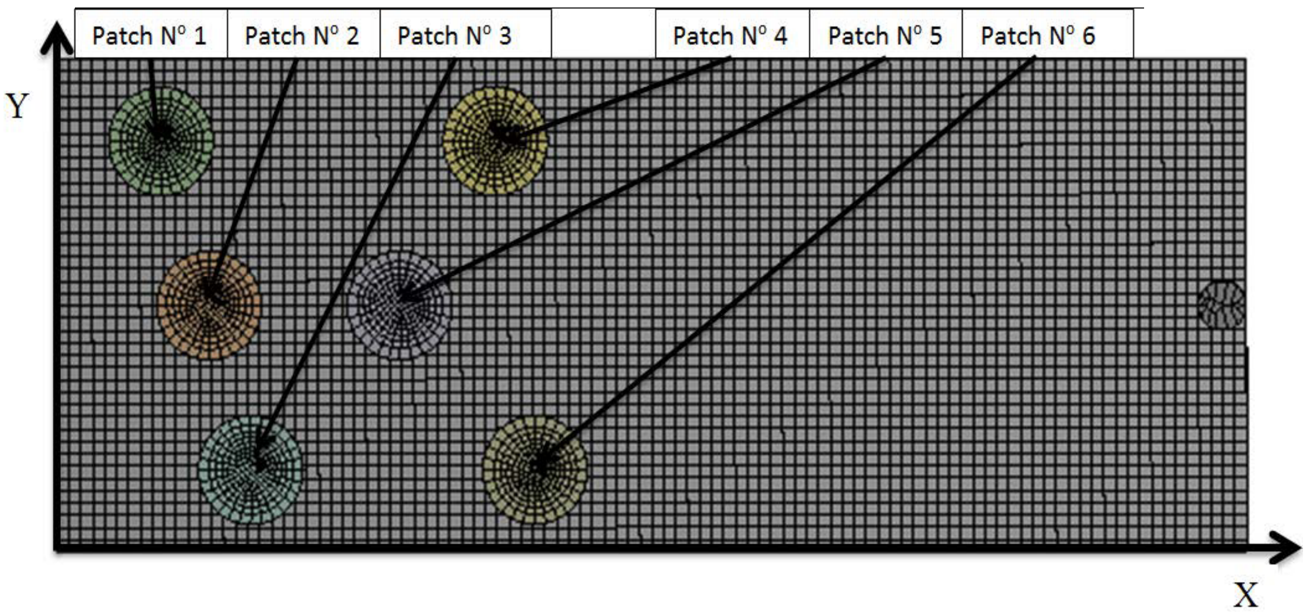


**Figure 2.** Plate with optimal patch location.

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**Table 6.** Plate with stiffener arrangement.

| N° of stiffeners | Stiffener center location Y [mm] | Stiffener cross section [mm] | Length [mm] |
|---|---|---|---|
| 1 | 100 | 4 x 5.5 | 500 |
| 2 | 50<br>150 | 1.5 x 2 | 500 |
| 3 | 50<br>100<br>150 | 1.5 x 1.5 | 500 |
| 3 | 50<br>100<br>150 | 1 x 2 | 500 |

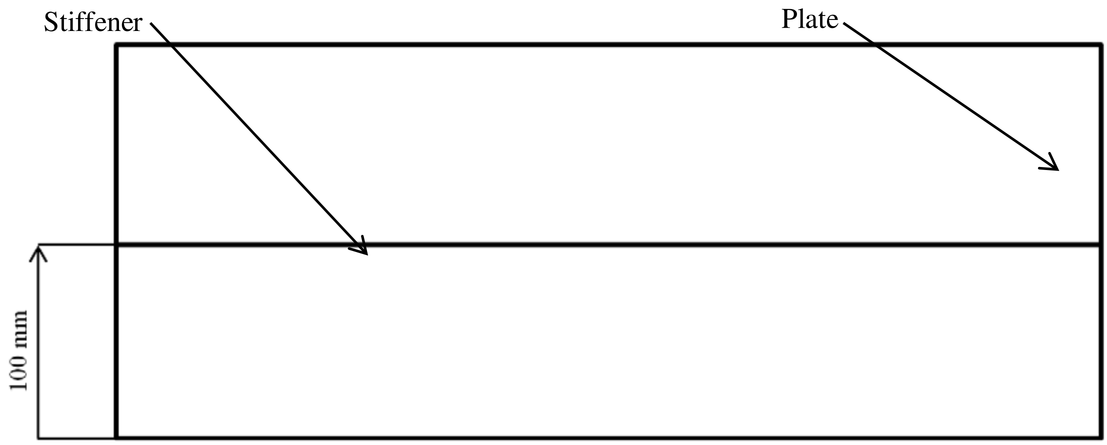


**Figure 3.** Plate with one stiffener arrangement.

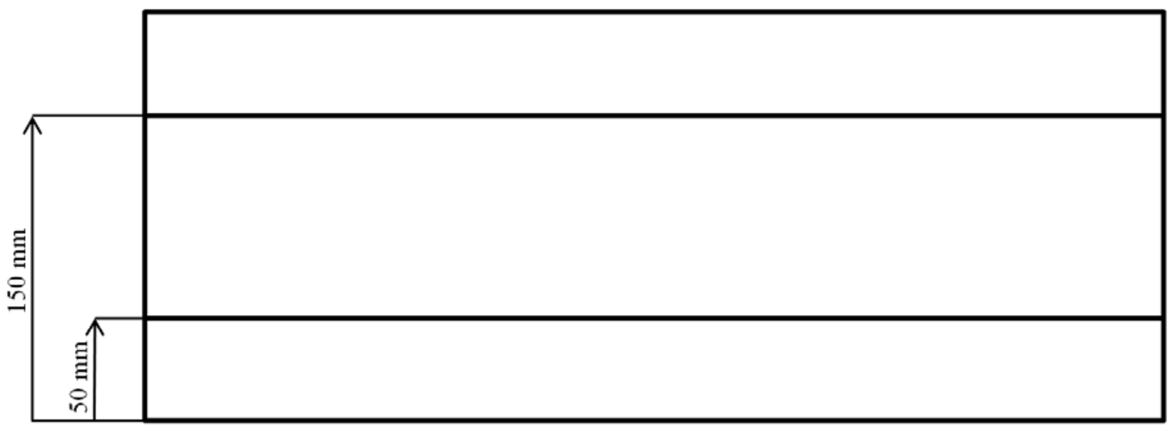


**Figure 4.** Plate with two stiffeners arrangement.


**Figure 5.** Plate with three stiffeners arrangement.

*3.1 analyzing the output results of FE models of plate with stiffeners*

By comparing the first six natural frequencies and also the amplitude FRF of the plate at different resonant frequencies, it was found that the best vibration attenuation (minimizing the amplitude FRF and maximizing the resonant frequency of the plate vibration) occurred when using one stiffener as follow.

**Table 7.** Bare plate / plate with stiffeners first six natural frequencies.

| Modes of Vibration | Bare plate | Natural frequency (Hz) | | | |
|---|---|---|---|---|---|
| | | 1 stiffener | 2 stiffeners | 3 stiffeners | 3 stiffeners |
| 1 | 3.7681 | 9.568 | 5.6833 | 5.3004 | 5.699 |
| 2 | 21.091 | 27.324 | 22.602 | 22.649 | 22.042 |
| 3 | 23.927 | 59.827 | 35.874 | 33.415 | 35.918 |
| 4 | 67.529 | 83.496 | 74.612 | 73.543 | 72.359 |
| 5 | 67.729 | 135.97 | 99.501 | 92.925 | 99.442 |
| 6 | 125.79 | 146.45 | 146.1 | 140.13 | 140.18 |

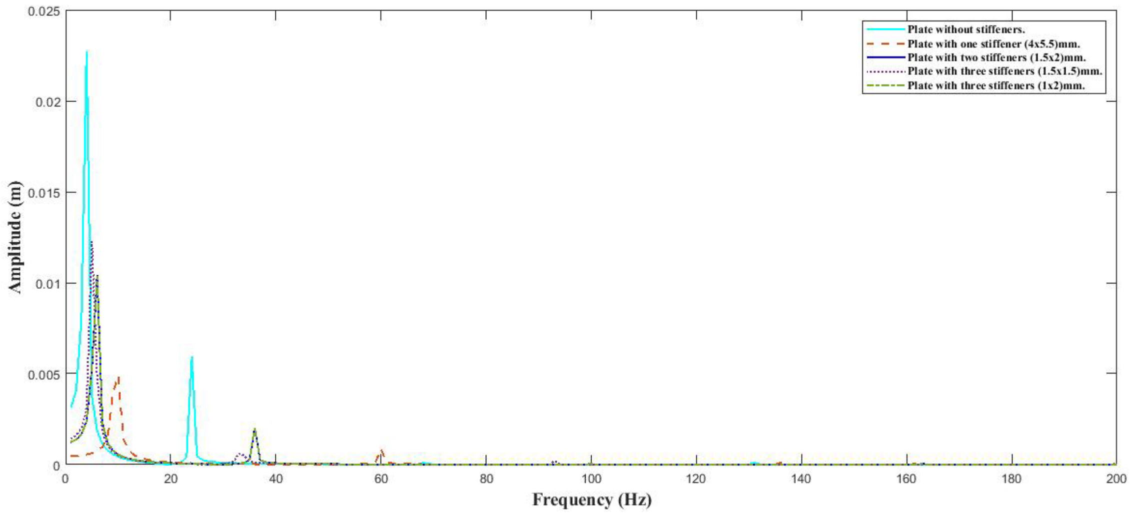


**Figure 6.** Bare plate / plate with different stiffeners arrangement FRF comparison.

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*3.2 Comparing the effect of using stiffeners and patches on Vibration Attenuation*

Vibration attenuation of mechanical structures at low frequencies is a challenging issue, and it is clear that by analyzing the output results of vibration attenuation using circular patches and stiffener with nearly same volume and mass. the first and third resonant frequencies of stiffened plate is more than the double of that for plate with patches and also the maximum FRF amplitude occurs at the first mode in both cases and its value for stiffened plate is approximately less than the half of the FRF amplitude of the plate with patches.

**Table 8.** Plate with patches/stiffeners first six natural frequencies.

| Modes of Vibration | Natural frequency (Hz) | | |
|---|---|---|---|
| | Bare Plate | Plate with patches | Plate with stiffener |
| 1 | 3.7681 | 4.3854 | 9.568 |
| 2 | 21.091 | 24.512 | 27.324 |
| 3 | 23.927 | 24.757 | 59.827 |
| 4 | 67.529 | 69.457 | 83.496 |
| 5 | 67.729 | 70.667 | 135.97 |
| 6 | 125.79 | 131.97 | 146.45 |

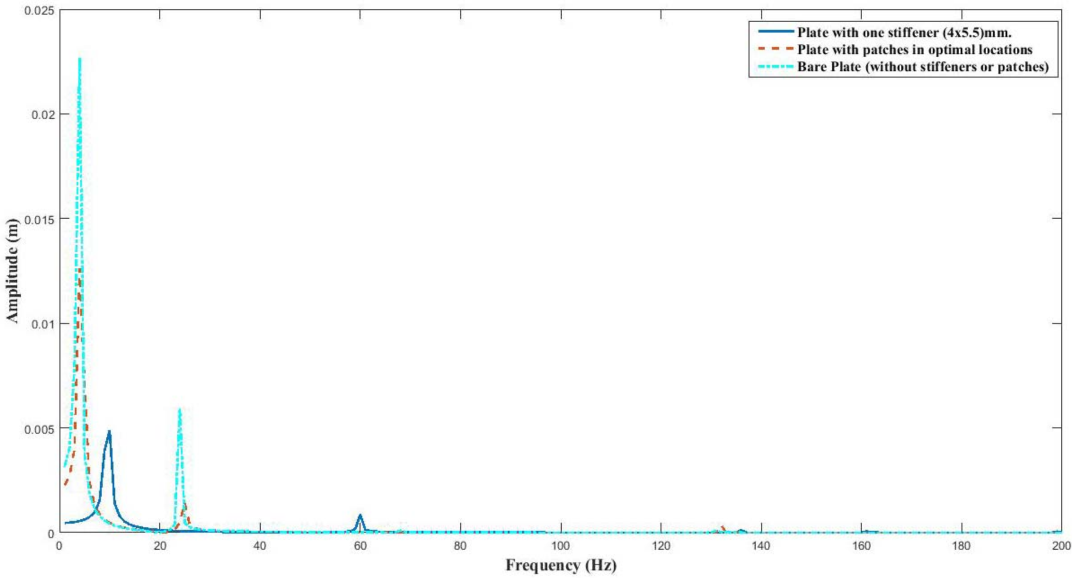


**Figure 7.** Plate with patches \ stiffener FRF comparison.

*3.3 Effect of Stiffener aspect ratio on vibration attenuation*

The effect of changing the ratio (a/b) of the stiffener's cross section area is studied by carrying up seven different trials using an aluminum plate with three stiffeners as seen in figure (8), for a reasonable comparison the variation in mass and volume of stiffeners is very small with changing the values of "a" and "b" as illustrated in table (9).

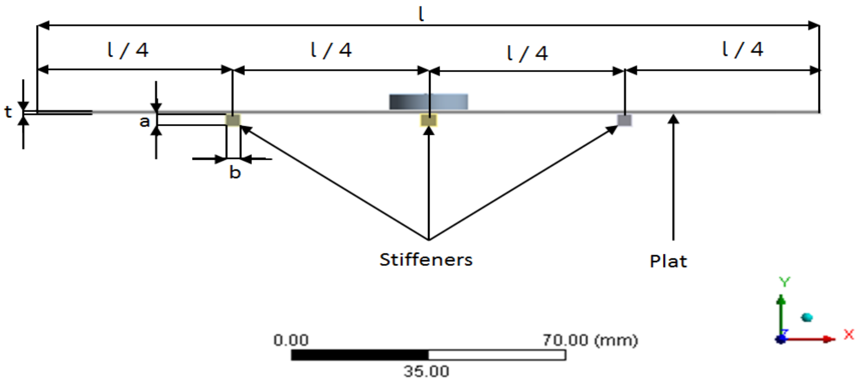


**Figure 8.** Aluminum plate with three stiffeners arrangement.

*3.3.1 Static structural analysis.*

**Table 9.** Maximum total deformation of plate with different aspect ratio stiffeners.

| Model N° | Aspect ratio a/b | Plate max. deformation (mm) |
|---|---|---|
| 1 | Bare plate | 25.781 |
| 2 | 1/7 | 16.589 |
| 3 | 2/6 | 9.0584 |
| 4 | 3/5 | 6.0329 |
| 5 | 4/4 | 4.7277 |
| 6 | 5/3 | 1.589 |
| 7 | 6/2 | 3.8785 |
| 8 | 7 / 1 | 4.1298 |

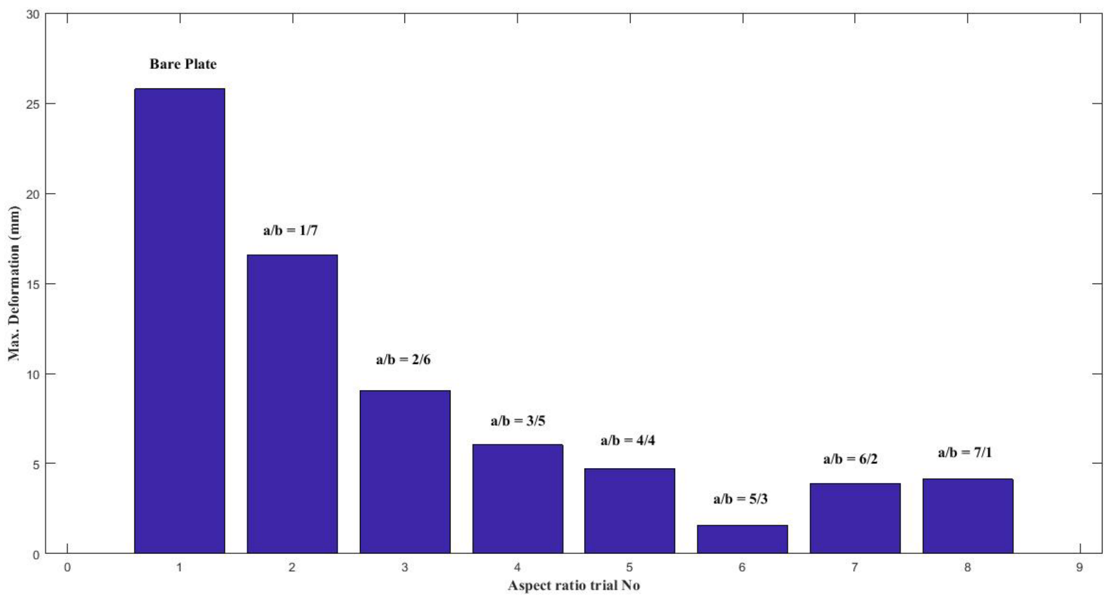


**Figure 9.** Plate maximum total deformation w.r.t b/a ratio.

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By analyzing the values of the plate total maximum deformation at different aspect ratios (a/b), it was found that the value of plate total maximum deformation starts to decrease by increasing the aspect ratio, the rate of total deformation decrease is changing inversely with the increase in the aspect ratio. The best value was achieved when the aspect ratio was (a/b = 5/3) and then the maximum deformation starts to increase by increasing the aspect ratio.

*3.3.2 Natural frequencies Comparison*

**Table 10.** Plate natural frequencies at different mode shapes.

| Model N° | Aspect ratio a/b | Natural frequency (Hz) | | | | | |
|---|---|---|---|---|---|---|---|
| | | Mode 1 | Mode 2 | Mode 3 | Mode 4 | Mode 5 | Mode 6 |
| 1 | Bare Plate | 3.7681 | 21.091 | 23.927 | 67.529 | 67.729 | 125.79 |
| 2 | 1/7 | 4.7052 | 24.585 | 29.794 | 78.296 | 83.574 | 144.73 |
| 3 | 2/6 | 6.372 | 28.958 | 40.754 | 92.539 | 113.52 | 171.35 |
| 4 | 3/5 | 7.8063 | 31.788 | 51.782 | 102.72 | 142.65 | 169.23 |
| 5 | 4/4 | 8.8126 | 31.99 | 61.681 | 106.39 | 158.59 | 181.07 |
| 6 | 5/3 | 14.742 | 56.354 | 87.079 | 126.69 | 162.3 | 213.19 |
| 7 | 6/2 | 9.7129 | 25.503 | 73.425 | 94.834 | 160.47 | 197.95 |
| 8 | 7/1 | 9.4075 | 22.785 | 68.881 | 85.712 | 158.46 | 189.29 |

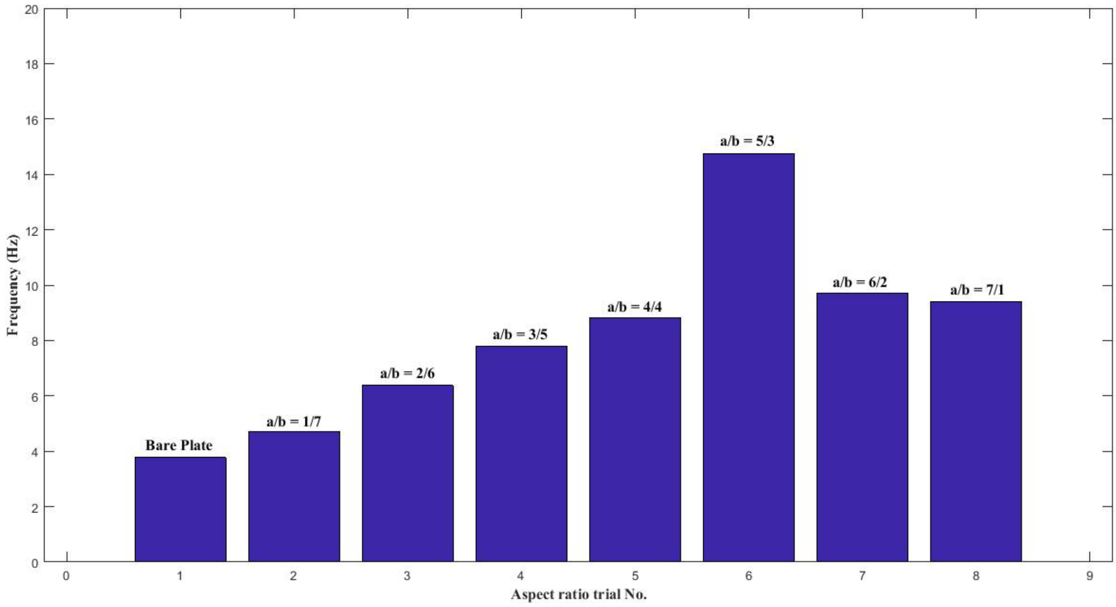


**Figure 10.** 1st mode shape natural frequency at different aspect ratios.

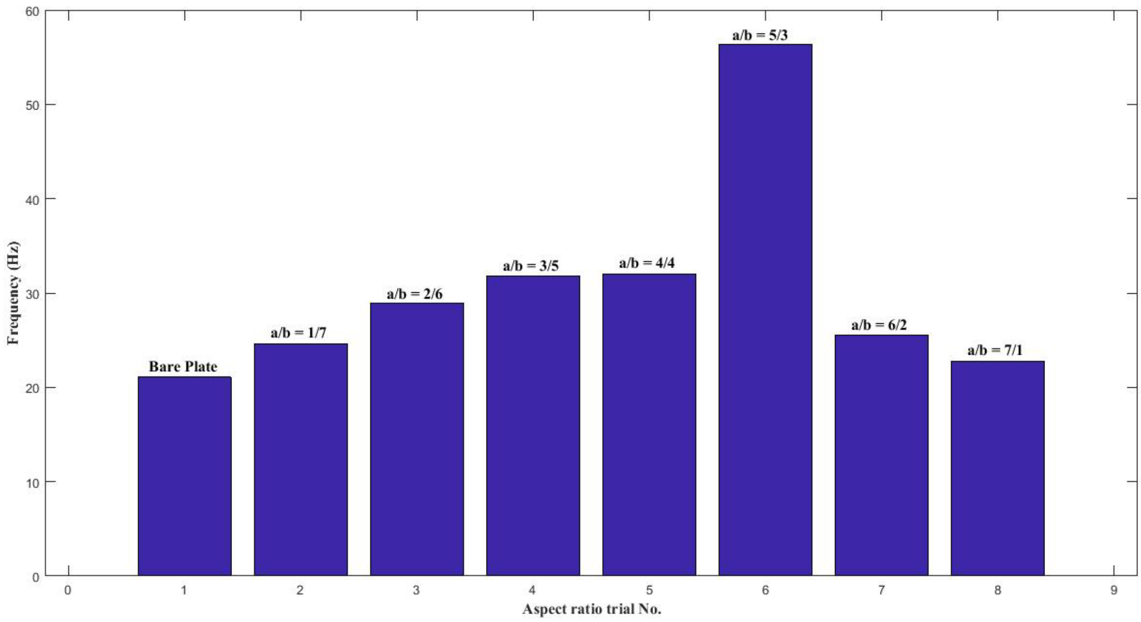


**Figure 11.** 2nd mode shape natural frequency at different aspect ratios.

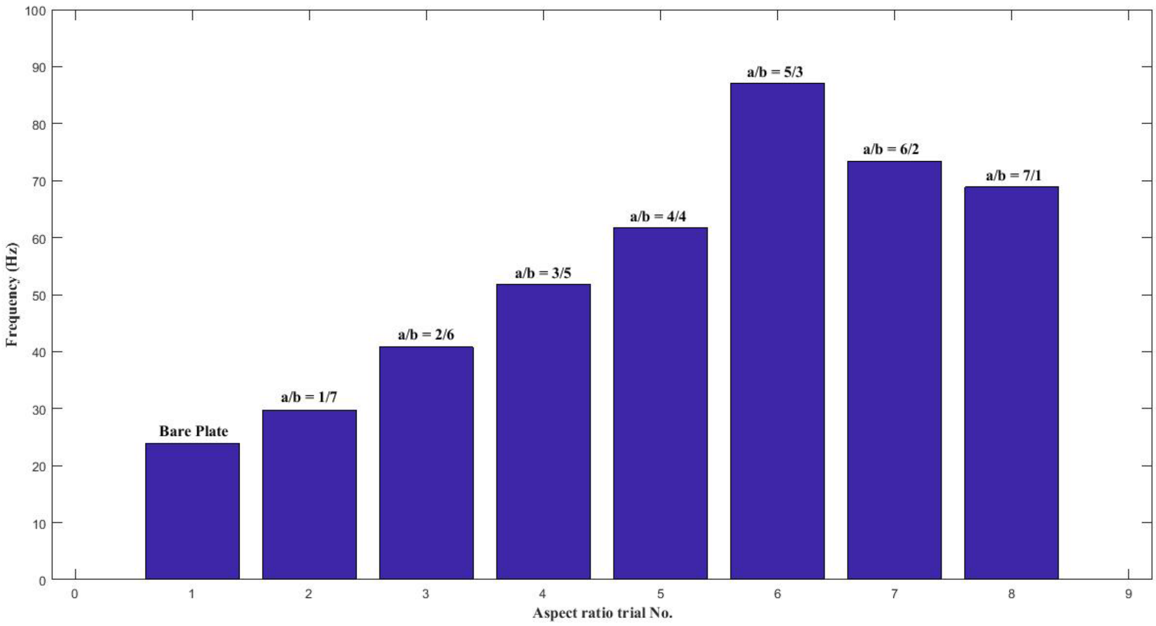


**Figure 12.** 3rd mode shape natural frequency at different aspect ratios.

By analyzing the values of the plate natural frequencies at the first three modes, it was found that the values of natural frequencies starts to increase by increasing the aspect ratio (a/b), the highest mode natural frequencies are achieved at aspect ratio (a/b = 5/3). The difference between the values of the resonant frequencies with respect to aspect ratio decreases at higher mode shapes. Using aspect ratios (a/b <1) is inefficient for vibration attenuation.

*3.3.3 FRF Comparison.*

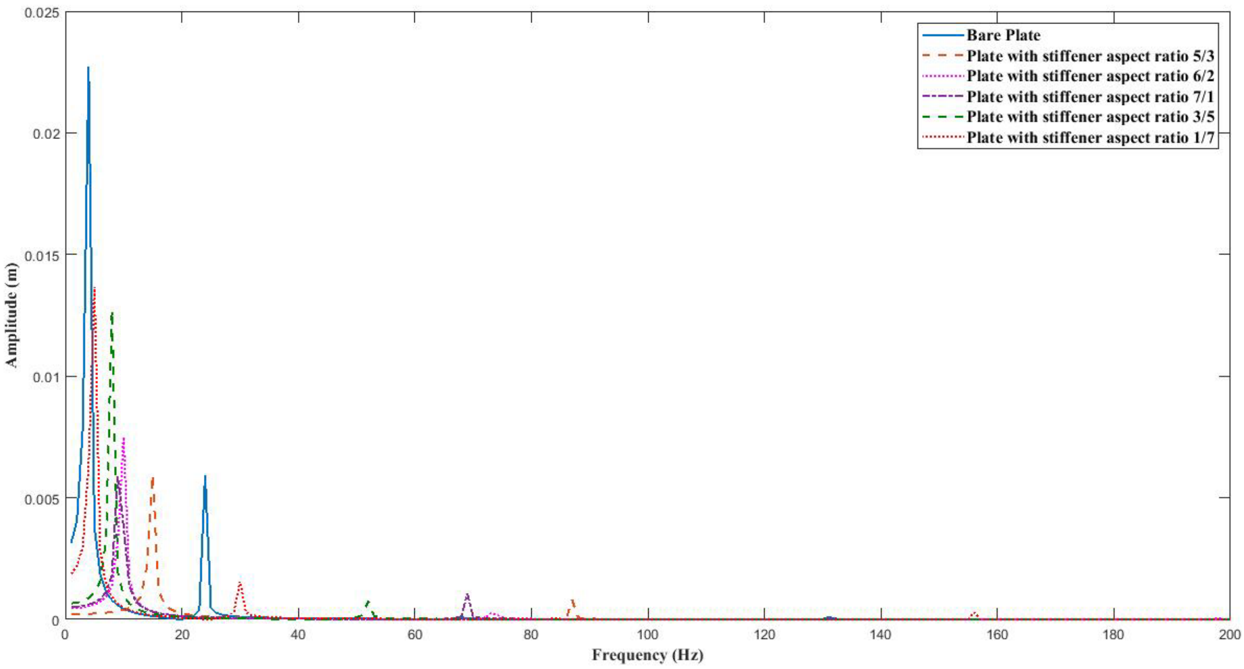


**Figure 13.** FRF of plate with different b/a ratios.

By analyzing the values of the plate FRF, it was found that the values of maximum FRF amplitude is very high when the aspect ratio (a/b < 1), the smallest FRF amplitude occurs when the aspect ratio (a/b = 5/3).

From the previous results, it is obvious that the vibration characteristics of the plate are improved by increasing the ratio (a/b) of the stiffener until a certain extent although keeping the stiffener's.

## 4. Conclusion

It can be concluded from simulations that using stiffeners is the best solution for passive vibration attenuation when compared with circular patches, the first and third resonant frequencies of stiffened plate is more than the double of that for plate with patches, also the FRF amplitude of the stiffened plate is approximately less than the half of the FRF amplitude of the plate with patches. The stiffener aspect ratio (a/b) has a great effect on vibration attenuation. The highest natural frequencies and smallest FRF amplitudes were achieved at aspect ratio (a/b = 5/3) and it is obvious that for aspect ratios (a/b <1) the vibration attenuation is not efficient.

## References

[1] Emad A 2017 *Passive Vibration Attenuation Analysis and Optimization of Satellite Solar Cell Panels.* (Cairo Egypt: Military Technical College PH. D. Dissertation)

[2] Venkata T and Seetha P 2013 Embedded computer based active vibration control system for vibration reduction of flexible structures. *Journal of Computer Science* **9**

[3] Hongli J, Jinhao Q and Pinqi X 2010 *Semi-active Vibration Control Based on Switched Piezoelectric Transducers*, ed Mickaël L (Nanjing University of Aeronautics and Astronautics, China)

[4] Aglietti1 G Aglietti G, Langley R, Rogers E and Gabriel S 2004 Model Building and Verification for Active Control of Microvibrations with Probabilistic Assessment of the Effects of Uncertainties. *J. Mechanical Engineering Science* **218** part c

[5] Masoud A 2013 Optimal Vibration Control in Structures using Level set Technique. *A thesis of Doctor of Philosophy, Waterloo, Ontario, Canada*
[6] Shahed H, Mohammed K, and Saeed A 2012 Vibration analysis of stiffened plates using finite element method. *Latin American Journal of Solids and Structures* **9**
[7] Amar N, Laren S and Prasant T 2018 Free vibration characteristics of stiffened plates. *International Journal of Advanced Structural Engineering* **10**, pp 153–167
[8] Chetak K 2018 Free Vibration Analysis of Rectangular Plate with Partial Stiffener Subjected to Static Uniform Loading. *A thesis of Master of technology, NIT, Jamshedpur*