Modal Analysis of a Simply Supported Aluminium Plate Structure with and Without a Damping Layer

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Abstract. T The modal analysis is one of the most important of the dynamic analyses. Any structure that is analysed at self-weight or without applying external loads is called modal case. In this research paper, there are two modal studies conducted using the numerical method and the finite element method. In the first modal study, without a damping layer, modal analysis is done for isotropic rectangular aluminium plate with two opposite edges simply supported boundaries using finite element analysis. After that, these analyzed values are compared with already published literature results of numerical values of natural frequencies and vibration modes for qualification of finite element procedure. In second study, the isotropic rectangular aluminium plate structure with four edges simply supported boundaries is redesigned as sandwiched into three proportionate layers such that the top layer thickness and bottom layer thickness are optimised with respect to the optimised damping layer- middle layer thickness is determined numerically for various mode shapes of rectangular plate structure without changing any outer dimensions by applying the convergence law of maximum loss factor by using MATLAB. Thereafter, the modal analysis is done separately on plate structure with an application of damping layer and without a damping layer by using finite element analysis. Here, two types of visco-elastic materials are taken for analysis as damping layer such as silicon rubber and neoprene rubber. Finally, the results of natural frequency and vibration modes of rectangular plate with a damping layer are compared with those of a plate without damping layer, and it is observed that the natural frequencies of plate structure are moderately decreased with damping layer.

Keywords: Natural frequency · Vibration modes · Structure · Rectangular plate · Damping layer · Loss factor · Silicon rubber · Neoprene rubber

1 Introduction

The work involved in this research paper aims to measure and understand the natural frequencies and mode shapes or vibration modes of rectangular aluminium plate structure with four edges that simply supported boundary conditions with the application of a damping layer and without the application of damping layer. Visco-elastic material

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is used as a damping layer in this study to increase the vibration capacity of plate-type structures. The damping materials commonly used are polymers, like natural rubber, silicon rubber, and neoprene rubber etc.; generally, they are suitably formulated to yield high damping infrequency and temperature range of interest. Any particular vibratory system that can vibrate in a specific or characteristic manner is known as a mode of vibration. In each mode of vibration, the system vibrates at its particular natural frequency and with a specific degree of freedom. Any practical structure (continuous body) possesses infinite natural frequencies. Resonance is a condition of a system at which it produces a maximum response in the system at some frequencies than at others. These are occurred when some external load frequencies are coincide with the natural frequencies of the system, which are known as the resonant frequencies of a system. Ungar and Kerwin explained the basic theoretical formulations to estimate the loss factor of a particular damped structure, and this is base for the design of any composite structure [1], is evaluated the damping property of the viscoelastic material by using silicon rubber as a constrained damping layer between metal plates. Here, the loss factor of the viscoelastic core material is estimated by ASTME756 norms, but the core thickness is taken arbitrarily and not integrated with base metal thickness. Finally got good results and compared them with experimental results [2], derived the equations of modal analysis of a thin isotropic aluminium plate for a numerical solution and after that, a solution was performed in the ANSYS program and it was observed that the natural frequencies were gradually increasing [3], discussed the commercial S4R shell element in modal analysis of a plate with simple boundary conditions and compared compared them with the eigen value solution capability, and noticed good agreement between FEA and eigen solution [4], by applying Kirchhof's plate theory, natural frequencies were calculated for a thin plate with simply boundary conditions and compared with the results of FEM; they are very close to the eigenvalue solutions. Finally, it was observed that thenatural frequency of vibration parameter was not affected by increasing the plate thickness [5], proved that the modal analysis results of natural frequency are not consistent with the analytical values when increasing the plate thickness [6], proved that the vibration characteristics-vibration modes and natural frequencies-of the structure are merely influenced by size and boundary properties [7], have done the case study on a cantilever plate using a passive control approachin which they first measured the natural frequency of Aluminium substrate (5051) witha viscoelastic material (Dyad 606) used as damping layer of top and bottom surfaces of the base metal. Then calculated the lateral vibration response of viscoelastic plates while applying the axial loads, compared these results with experimental values, and got good agreement [8], discussed the characteristics of vibration, such as natural frequency and vibration modes, of a steel rectangular plate structure with two opposite side edges under simply supported boundary conditions and compared the results obtained by using FEM through ANSYS with the results obtained analytically [9], dealt with analytical modeling of simply supported beams of different material configurations, such as undamped aluminium panel, damped aluminium—steel composite panel, and aluminium-viscoelastic-steel composite panel, to obtain transient responses due to sinusoidal dynamic loading conditions. Here, all three configurations have the same cross-sectional area and dimensions. The analytical modeling of the CLD treatment for thin composite metallic panels was exercised, and good agreement was noticed among these three cases [10], explored the vibration experimentations ona cantilever beam for partially constrained damping layer (PCDL) treatment and fully constrained damping layer (FCDL) treatment. Over a period of time, the damping capacity of the constrained layer damping system reduces due to less shearing of the visco-elastic material, which is basically the extension of the damping layer near the curvature of the beam during different bending modes. Here, the damping performance of visco-elastic material in constrained layer damping treatment can be increased by artificially cutting the upper constrained layer and visco-elastic material. The final experimental values are compared with an undamped aluminium cantilever beam and different typically visco-elastic or damping material sandwiched cantilever beams. Based on the experiments, it is strongly recommended that the vibration response amplitude of FCDL beam and PCDL beam is less as compared to undamped beam and that the vibration response amplitude of PCDL beam is less as compared to FCDL beam [11], explored the application of different kinds of viscoelastic materials in Passive constrained layer damping treatments and ensured that damping effect is increased with the application of viscoelastic materials used as a core in beams. By varying the thickness of the viscoelastic layer, different combinations of sandwiched, simply supported beams are made and FEA has been done for vibration analysis. Thereafter, experimentation was carried out. The numerical results of developed models were validated with the experimental results. A very accurate agreement has been made between the results.

In this modest work, optimised damping layer thickness is measured numerically for different mode shapes for a given modal plate structure without changing any outer dimensions by applying the convergence law of the maximum loss factor using MATLAB. Thereafter, the modal analysis is done separately for a simply supported rectangular aluminum plate with the application of a damping layer and without the application of damping layer by using finite element analysis. There are two types of visco-elastic materials taken for analysis as damping layers: silicon rubber and neoprene rubber. The analysis simulation is performed in FEA software, ANSYS R21.

2 Problem Description

A modal plate is considered an isotropic rectangular aluminium structure consisting of length a of 1 m, width b of 0.5 m, and thickness h of 1.5 mm (0.0015 m). The material properties of the structure in Fig. 1 are as follows: Young‘s modulus, $E = 70x10^9$ N/m^2, Density, $\rho = 2770$ kg/m^3, and Poisson coefficient, $\nu = 0.33$.

2.1 Geometry and Boundary Conditions of a Plate

The rectangular plate being simply supported on four edges, the edge conditions of the plate are no displacement at x = 0, x = a and y = 0, y = b and no moment at x = 0, x = a and y = 0, y = b.

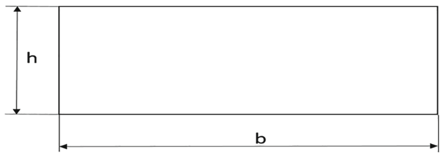


Fig. 1. Geometry of the isotropic aluminium plate

3 Analytical Approach

3.1 Mathematical Formulations for the Free Vibrations

The modal plate is a rectangular, simply supported aluminium plate that is isotropic with uniform thickness h, elasticity of modulus E, and mass density ρ. The effect of rotational inertia moments within the plate and shear strains due to transverse shearing stresses will be neglected. As per plate theory, natural frequencies and mode shapes depend on mainly material properties and the geometry and boundary conditions of the plate and are inherent properties of the elastic plate regard less of any change. In a free or natural vibration, the only inertial forces are acting as transverse forces on the plate. The following is the general equation for natural mode of vibration ω_{mn} of the modes m and n for plate structure:

$$\omega_{mn} = \pi^2\left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)\sqrt{\frac{D}{\rho h}} \tag{1}$$

where, $D = \frac{Eh^3}{12(1-\upsilon^2)}$ is the plate flexural rigidity or bending stiffness, υ is the poisons ratio for the plate material, and also m and n are the wave numbers corresponding to the trace wavelengths a and b of the motion x- and y-axes respectively.

The analysis of natural pulsations of plate under study gives us:

If m = 1, n = 1, and $D = \frac{70X\,10^9X\,0.0015^3}{12(1-0.3^2)} = 21.63\ \mathrm{N-m}$; $\omega_{11} = \pi^2\left(\frac{1}{1} + \frac{1}{0.5^2}\right)\sqrt{\frac{D}{2700X\,0.0015}}$ rad/s; and $f_{11} = \frac{\omega_{11}}{2\pi} = 18.15$ Hz.

If m = 1, n = 2, $\omega_{12} = \pi^2\left(\frac{1}{1} + \frac{4}{0.5^2}\right)\sqrt{\frac{D}{2700X\,0.0015}}$ rad/s; and $f_{12} = \frac{\omega_{12}}{2\pi} = 61.69$ Hz.

If m = 2, n = 1, $\omega_{21} = \pi^2\left(\frac{4}{1} + \frac{1}{0.5^2}\right)\sqrt{\frac{D}{2700X\,0.0015}}$ rad/s; and $f_{21} = \frac{\omega_{21}}{2\pi} = 29.03$ Hz.

If m = 2, n = 2, $\omega_{22} = \pi^2\left(\frac{4}{1} + \frac{4}{0.5^2}\right)\sqrt{\frac{D}{2700X\,0.0015}}$ rad/s; and $f_{22} = \frac{\omega_{22}}{2\pi} = 72.58$ Hz.

3.2 Mathematical Formulations for Optimization of a Damping Layer Thickness of Plate

In this work, VEM behavior is described in the frequency domain. A theoretical procedure is developed for determine the optimisation damping layer thickness at different wave patterns using MATLAB software. Thereafter, identify the maximum proportionate damping layer thickness at fundamental mode shape and consider it as the optimised

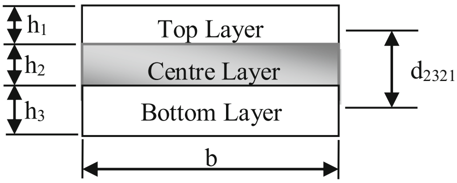


Fig. 2. Dimensional notations of plates

damping layer thickness for a given plate geometry without any change to the plate structure. At very high mode shapes, the VEM response is purely elastic, and the thickness of damping layer is also very low.

The effect of center layer- damping layer as shown in Fig. 2-on the damping capacity of the whole three layered structure is experienced by the material‘s stiffness as well as by its damping. The Complex elastic modulus or complex shear modulus defines the damping material‘s stiffness and its damping capacity. These complex forms are usually represented as $E* = E^I(1 + i\,\eta_E)$ and $G* = G^I(1 + i\,\eta_G)$ whereas η_E and η_G are loss factors and are usually assumed to be equal for a given material. Nevertheless, the loss factor of damping material is measured in frequency domain by using the below equation:

$$\text{Loss factor, } \eta_s = 10^{(0.00033(\log_{10}f)^3 - 0.118(\log_{10}f)^2 + 0.6067\log_{10}f - 0.780} \tag{2}$$

As part of the estimation of the maximum loss factor of a whole aluminium plate structure with a damping layer, initially the given plate is divided into two face plates with equal thickness and named the top and bottom layers, and then one central layer, assuming some thickness interrelated with the geometric parameter (Y) [12]. By using the maximum loss factor convergence principle, the damping layer thickness generated in the 4x3 mode shape will be considered the optimized thickness of the damping layer. The maximum loss factor $(\eta_{DC,MAX})$ of a whole sandwiched type simply supported plate structure with damping layer is measured by using below formula:

$$\text{Maximum loss factor, } \eta_{DC,MAX} \frac{\eta_s Y}{(2+Y) + 2(\sqrt{(1+Y)\left(1+\eta_s^2\right)}} \tag{3}$$

where η_s is loss factor and $Y = \frac{d_{321}^2}{D_t(1-\nu^2)}\left[\frac{E_1 h_1 E_3 h_3}{E_1 h_1 + E_3 h_3}\right]$ is a geometry parameter of the plate, although it is actually a function of sizes of layers and their elastic modulus of the layers and was initially assigned as value 3 for this application [12]. Here, D_t is the sum of the flexural rigidity of the whole sandwiched plate.

The MATLAB programming code was run in 4x3 mode and the optimized damping layer thickness for a given modal aluminium plate without any change in dimensions was 0.4711 mm ($h_2 = 0.5$ mm rounded off), which was further used for the modal analysis of plates with different damping materials as damping layers. The modal analysis details are discussed in point 4.

Table. 1. Comparision of literature values and present work values

Sl. No.	Vibration mode(m, n)	Natural frequency (Hz)				
		Reference (published)			Present	
		Eigen Value	FEA Value	ε (%)	FEA value	ε (%)
1	(1, 1)	95.97	107.9	0.1234	107	0.1149
2	(1, 2)	239.93	296.74	0.2367	295	0.2295
3	(2, 2)	383.89	325.11	0.1531	323	0.1586
4	(3, 1)	623.82	658.27	0.0552	657	0.0531
55	(3, 3)	863.75	711.4	0.1763	713	0.1745

4 Modal Analysis of a Rectangular Simply Supported Plate

In this work, there are three types of modal analyses done to achieve the requirements.

4.1 Modal No. 1

A modal analysis is performed on already published literature [4] of modal analysis of steel material and the results compared as shown in the Table 1. In this analysis, steel plate is taken, which has the following material properties: Elasticity of modulus (E) is $2x10^{11}$ N/m^2, Poisson's ratio (ν) is 0.3 and density (ρ) is 7850 kg/m^3.

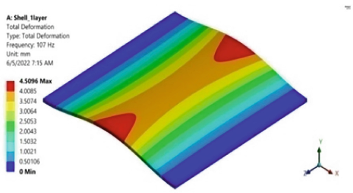
The present modelled plate structure has a volume of 2e+07 mm^3 as well as a mass of 157kg,and simulations are carried out for the plate by considering the quadratic element of size 30mm and then discretization is done with9825 elements for 29,866 nodes. The present results of natural frequencies are verified with the data reported in published literature [4] and with eigenvalue solutions, and it is agreed that the results are very close to those reported in the published literature.

$$\varepsilon\,(\%) = \frac{fS - fC}{fC} \text{ is error percentage.}$$

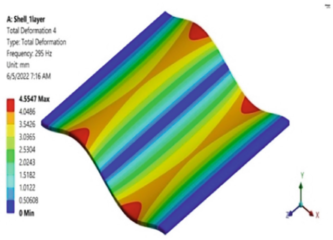
Results and Discussion: In this analysis, it is observed that the natural frequencies and vibration modes of two opposite edges of simply supported plates are analysed by applying FEM. The present results of natural frequencies are verified with the data reported in published literature and with eigenvalue solutions. It is agreed that those natural frequency results are very near to the data reported in the literature. The above frequencies are translated into geometric shapes by the following vibration patterns in Fig. 3.

4.2 Modal No. 2: A Simply Supported Rectangular Isotropic Aluminium Plate Without a Damping Layer

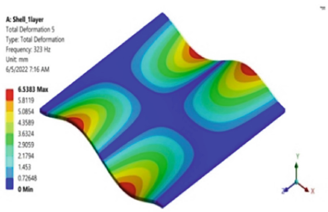
The finite element method was applied for the modal analysis of the isotropic plate as shown in Fig. 4. In this modal analysis, the material properties are considered to Elastic



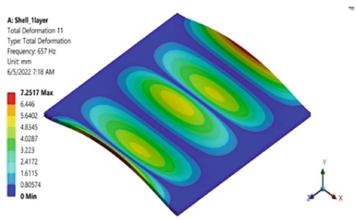
Vibration mode (a), (m=1, n=1)



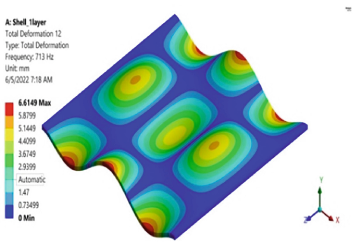
Vibration mode (b), (m=1, n=2)



Vibration mode (c), (m=2, n=2)



Vibration mode (d), (m=3, n=1)



Vibration mode (e), (m=3, n=3)

Fig. 3. Vibration patterns of reference plate structure

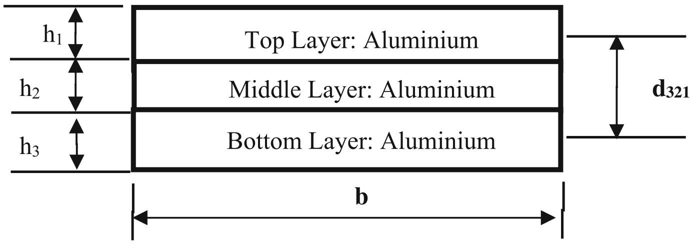


Fig. 4. Three layered isotropic aluminium plate

modulus (E) is 70x10^9 N/m^2, Density (ρ) is 2770 kg/m^3, and Poisson's ratio (ν) is 0.33. Where a (=1m) is the length of the plate in the x axis direction, width b (=0.5 m) in the y

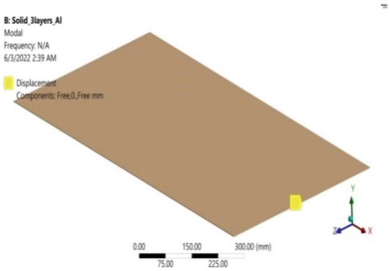


Fig. 5. Simply supported boundary conditions of Plate

axis direction, and h (=0.0015 m) is the total thickness of the plate in the z axis, where, $h = h_1 + h_2 + h_3$.

In this, the boundary conditions and meshing details of plate are shown in Figs. 5, 6 and 7, the modeled plate has a volume of 7.50e+05 mm^3 as well as a mass of 2.08 kg, and then simulations are carried out for the plate by considering the quadratic element of size 30 mm, and discretization is done with 1683 elements for 12,540 nodes.

Results and Discussion: In this analysis, it is observed that the first eleven natural frequencies and vibration modes of all edges of simply supported plates without a damping layer are analysed by applying FEM. The results of natural frequencies are verified with the analytical values and shown in Table 2. It is noticed that FEA natural frequency

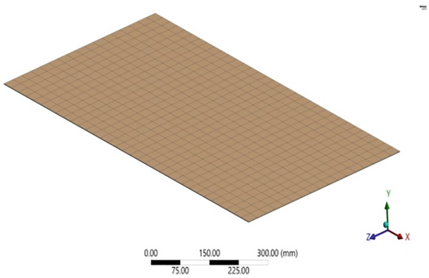


Fig. 6. Meshing details of plate

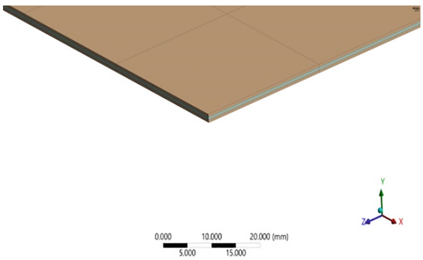
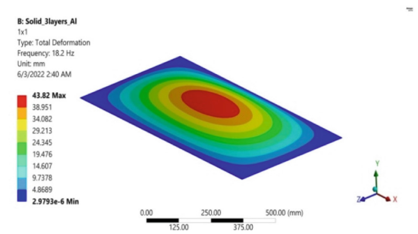


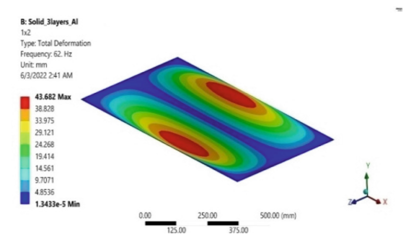
Fig. 7. Corner view of three layered plate

Table. 2. Values of natural frequencies of plate without damping layer

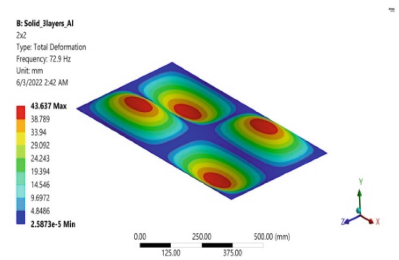
Sl. No.	Vibration mode(m, n)	Natural frequency (Hz)		
		Analytical value	FEA value	ε (%)
1	(1, 1)	18.15	18.20	0.0082
2	(1, 2)	61.69	62.00	0.0050
3	(2, 2)	72.58	72.90	0.0044
4	(2, 3)	145.16	146.00	0.0057
5	(3, 3)	163.31	164.00	0.0042
6	(3, 4)	264.92	266.00	0.0040
7	(4, 4)	290.32	292.00	0.0057
8	(4, 5)	420.97	423.00	0.0048
9	(5, 5)	453.63	456.00	0.0052
10	(5, 6)	613.31	618.00	0.0076
11	(6, 6)	653.23	660.00	0.0103



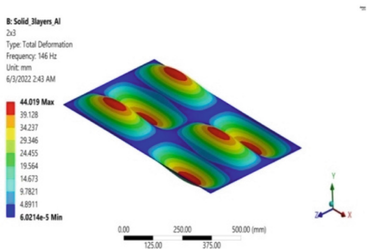
Vibration mode (a), (1, 1)



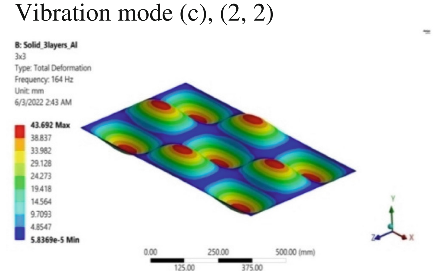
Vibration mode (b), (1, 2)



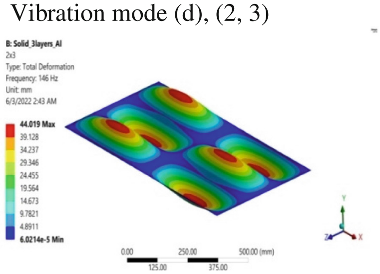
Vibration mode (c), (2, 2)



Vibration mode (d), (2, 3)

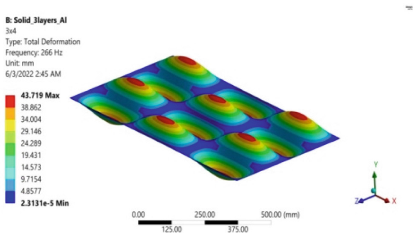


Vibration mode (d), (2, 3)

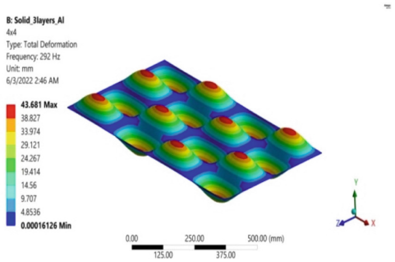


Vibration mode (e), (3, 3

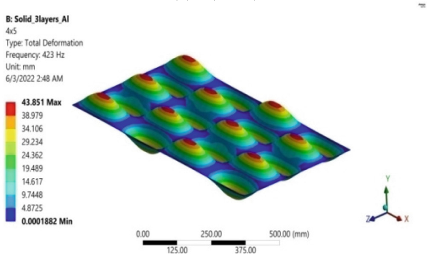
Fig. 8. Vibration patterns of a plate without damping layer



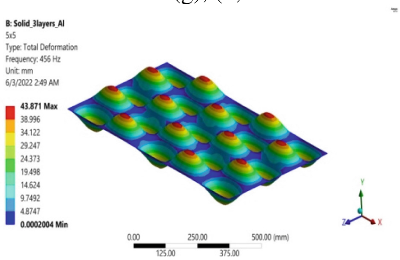
Vibration mode (f), (3, 4)



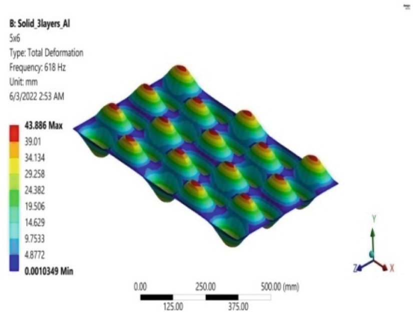
Vibration mode (g), (4, 4



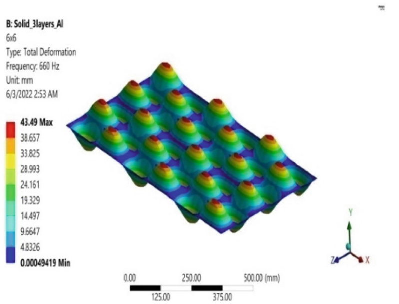
Vibration mode (h), (4, 5)



Vibration mode (i), (5, 5)



Vibration mode (j), (5, 6)



Vibration mode (k), (6, 6)

Fig. 8. (*continued*)

results are very close to the analytical values. The above frequencies are translated into geometric shapes by the following vibration patterns in Fig. 8.

4.3 Modal No. 3: A Simply Supported Rectangular Aluminium Plate with a Damping Layer-Silicon Rubber

The finite element method was applied for the modal analysis of a sandwiched aluminium plate with damping layer—Silicon rubber is as shown in Fig. 9. In this modal analysis, the material properties are considered to elastic modulus (E) is $70x10^9$ N/m^2, Density (ρ) is 2770 kg/m^3, and Poisson's ratio (ν) is 0.33. And damping layer, Silicon rubber, whose elastic modulus (E) is $1x10^6$ N/m^2, Density (ρ) is 1100 kg/m^3, and Poisson's ratio (ν) =0.47. Where a (=1 m) is a length of the plate in x axis direction, width b (=0.5 m) in y axis direction, and h (=0.0015 m) is the total thickness of plate in z axis, where

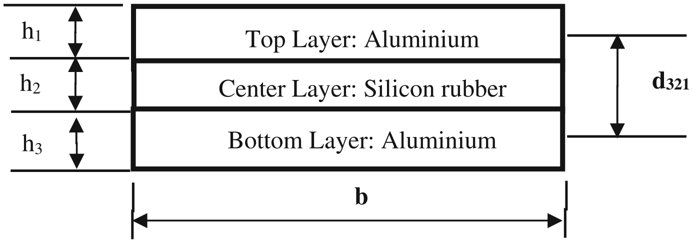


Fig. 9. Rectangular plate with silicon rubber as damping layer

$h = h_1 + h_2 + h_3$. Now $h_1 = 0.0005$ m, $h_3 = 0.0005$, and $h_2 = 0.0005$m is a damping layer thickness as mentioned earlier in Sect. 3.2.

In this, the modelled plate has a volume of 7.50e + 05 mm^3 as well as a mass of 1.66kg, and then simulations are carried out for the plate by considering the quadratic element of size 30mm, and discretization is done with 1683 elements for 12,540 nodes.

Results and Discussion: In this analysis, it is observed that the first eleven natural frequencies and vibration modes of all edges of simply supported plates with damping layer-Silicon rubber are analysed by applying FEM. The results of natural frequencies are verified with the analytical values and shown in Table 3. It is noticed that the FEA natural frequency results are very close to the analytical values. The above frequencies are translated into geometric shapes by the following vibration patterns in Fig. 10.

Table 3. Values of natural frequencies of plate with damping layer-Silicon rubber

Sl. No.	Vibration mode(m, n)	Natural frequency (Hz)		
		Analytical value	FEA value	ε (%)
1	(1, 1)	18.15	30.30	0.6694
2	(1, 2)	61.69	64.20	0.0406
3	(2, 2)	72.58	71.70	0.0121
4	(2, 3)	145.16	119.00	0.1802
5	(3, 3)	163.31	131.00	0.1978
6	(3, 4)	264.92	195.00	0.2639
7	(4, 4)	290.32	211.00	0.2732
8	(4, 5)	420.97	292.00	0.3063
9	(5, 5)	453.63	312.00	0.3122
10	(5, 6)	613.31	412.00	0.3282
11	(6, 6)	653.23	438.00	0.3294

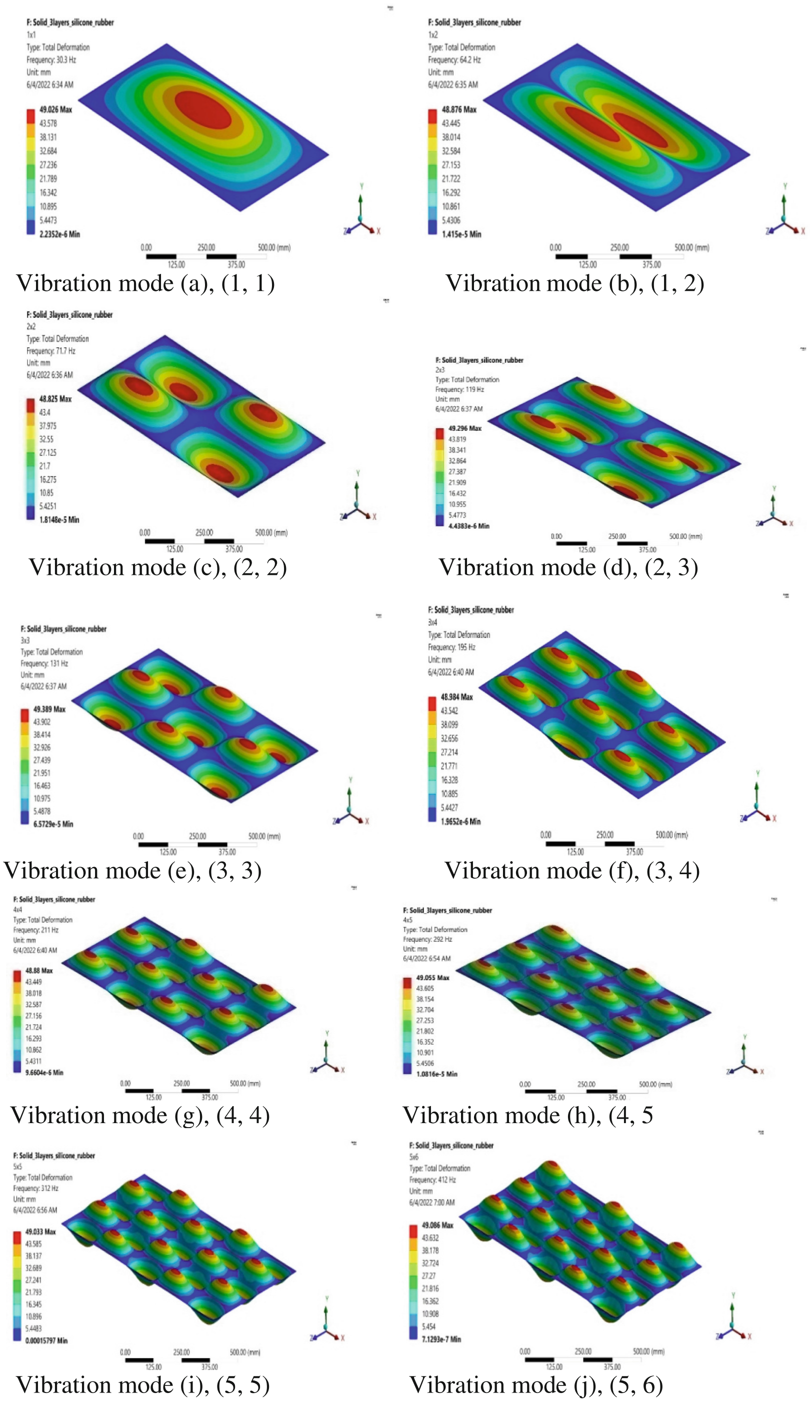
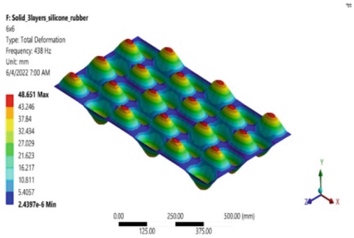


Fig. 10. Vibration patterns of a plate with damping layer—silicon rubber



Vibration mode (k), (6, 6)

Fig. 10. (*continued*)

4.4 Modal No. 2: A Simply Supported Rectangular Aluminium Plate with a Damping Layer-Neoprene Rubber

The finite element method was applied for the modal analysis of a sandwiched aluminium plate with a damping layer—neoprene rubber is as shown in Fig. 11. In this modal analysis, the material properties are considered to elastic modulus (E) is 70x10^9 N/m^2, Density (ρ) is 2770 kg/m^3, and Poisson's ratio (ν) is 0.33. And damping layer, neoprene rubber, elastic modulus (E) is 8.15x10^5 N/m^2, Density (ρ) is 960 kg/m^3, and Poisson's ratio (ν) is 0.49. Where a (=1 m) is a length of the plate in x axis direction, width b (=0.5 m) in the y axis direction, and h (=0.0015 m) is the total thickness of plate in z axis, where, $h = h_1 + h_2 + h_3$. Now $h_1 = 0.0005$ m, $h_3 = 0.0005$, and $h_2 = 0.0005$ m is a damping layer thickness as mentioned earlier in Sect. 3.2.

In this, the modelled plate has a volume of 7.50e+05 mm^3 as well as a mass of 1.62 kg, and then simulations are carried out for the plate by considering the quadratic element of size 30 mm, and discretization is done with 1683 elements for 12,540 nodes.

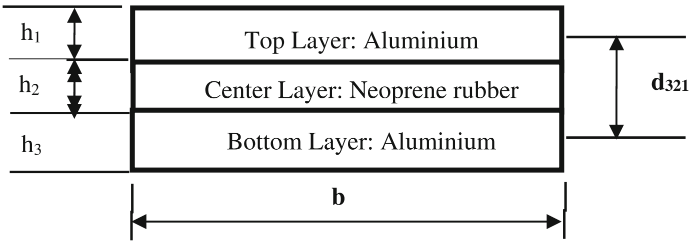
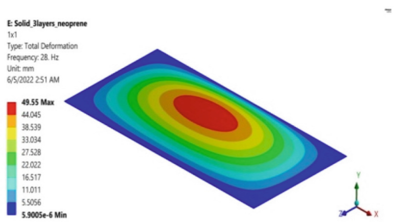


Fig. 11. Rectangular plate with neoprene rubber as damping layer

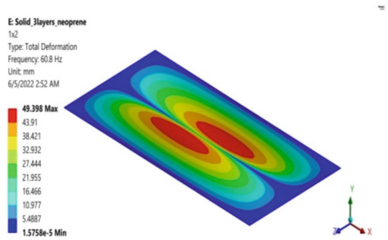
Table 4. Values of natural frequencies of plate with damping layer-neoprene rubber

Sl. No.	Vibration mode(m, n)	Natural frequency (Hz)		
		Analytical value	FEA value	ε (%)
1	(1, 1)	18.15	28.00	0.5427
2	(1, 2)	61.69	60.80	0.0144
3	(2, 2)	72.58	68.20	0.0603
4	(2, 3)	145.16	115.00	0.2077
5	(3, 3)	163.31	127.00	0.2223
6	(3, 4)	264.92	191.00	0.2790
7	(4, 4)	290.32	207.00	0.2869
8	(4, 5)	420.97	289.00	0.3134
9	(5, 5)	453.63	309.00	0.3188
10	(5, 6)	613.31	410.00	0.3315
11	(6, 6)	653.23	436.00	0.3325

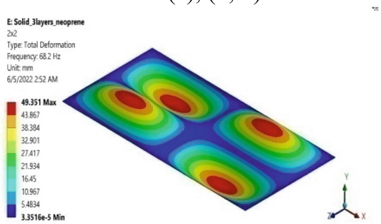
Results and Discussion: In this analysis, it is observed that the first eleven natural frequencies and vibration modes of all edges of simply supported plates with damping layer-neoprene rubber are analysed by applying FEM. The results of natural frequencies are verified with the analytical values and shown in Table 4. It is noticed that the FEA natural frequency results are very close to the analytical values. The above frequencies are translated into geometric shapes by the following vibration patterns in Fig. 12.



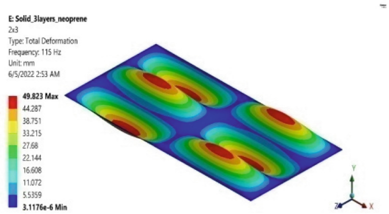
Vibration mode (a), (1, 1)



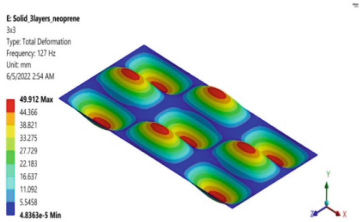
Vibration mode (b), (1, 2)



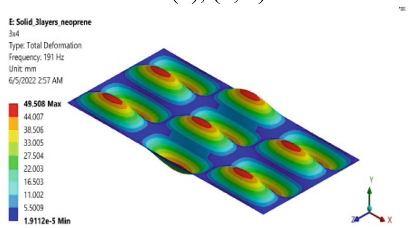
Vibration mode (c), (2, 2)



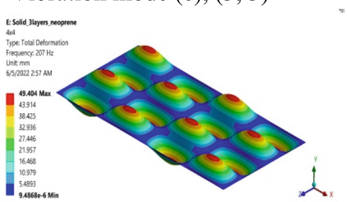
Vibration mode (d), (2, 3)



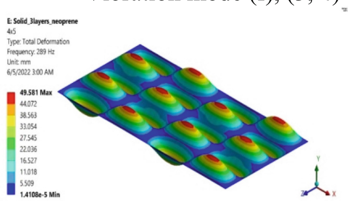
Vibration mode (e), (3, 3)



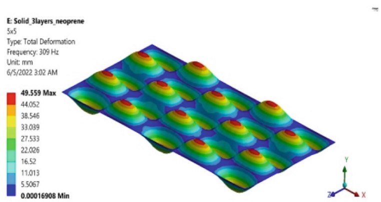
Vibration mode (f), (3, 4)



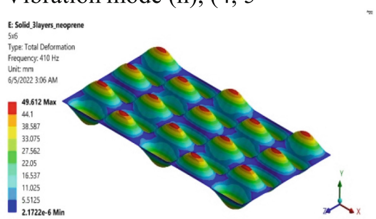
Vibration mode (g), (4, 4)



Vibration mode (h), (4, 5

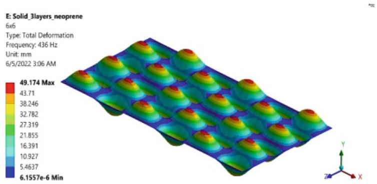


Vibration mode (i), (5, 5)



Vibration mode (j), (5, 6)

Fig. 12. Vibration patterns with damping layer—neoprene rubber



Vibration mode (k), (6, 6)

Fig. 12. (*continued*)

5 Conclusion

The modal analysis is done on the isotropic aluminium plate without the application of a damping layer and with the application of an optimised damping layer by using silicon rubber and neoprene rubber separately. The FEA results of the modes and natural frequencies of a simply supported plate with damping layer the same as aluminium material are compared with analytical results of an isotropic aluminium plate, and it is observed that the FEA results are very close to the analytical values.

Now these analytical values of natural frequencies of isotropic aluminium plate are compared with the FEA results of the application of viscoelastic materials, silicon rubber and neoprene rubber separately, used as damping layer, and it is observed that the application of damping material in the isotropic aluminium plate structure, the natural frequencies are decreased due to damping added passively to the plate without any changes in cross-sectional area and dimensions.

Finally, in both cases, the natural frequencies of the plate structure are decreased with the application of the damping layer, except for the first two modes. Further, the vibration response analysis needs to be done for the same plate structures with and without a damping layer.

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