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# Passive vibration damping of cantilever plate

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**Abstract.** Vibration control of mechanical systems divided into two types active and passive, this case study illustrates other approach of passive control of the cantilever plate by using double layers of the viscoelastic material (Dyad 606) one in the upper face of the Al substrate (5051) and the second one in the lower face. This layers are affected by axial uniform distributed force by applying tension load, the damping level depend on the value of this axial force where the vertical component of this force consistently the other way of the vibrating element that reduce vibration amplitude in this direction by about (95%) of the original amplitude without damping layers. Computational analysis using ANSYS program are used firstly to verify that the results from the software are compatible with that obtained from analytical and experimental evaluation by comparing the values of natural frequencies. Then using ANSYS model to calculate the reduction occurs in the lateral vibration with changing of the axial loads on the viscoelastic plates. The benefits of this model are enhancing the reliability and reducing the cost.

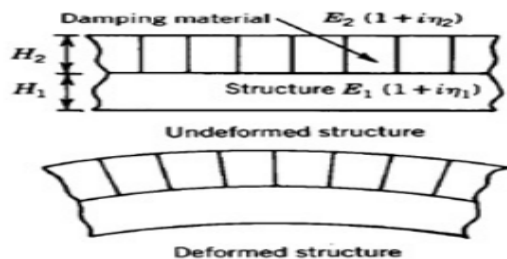
## 1. Introduction

Various techniques are used in damping process of the undesirable vibrations that may cause damaging of the structures and machines. Damping systems are divided into two main types active and passive [1], at the first, active vibration control uses many kinds of sensors and actuators to generate forces in the right direction and time to sustain the unwanted frequencies that cause vibration[2], this type of vibration control have higher cost and low reliability[3]. On the other hand, passive vibration control doesn't use any sensor or actuator that require external power units and depend only on the damping properties of the used material[4].

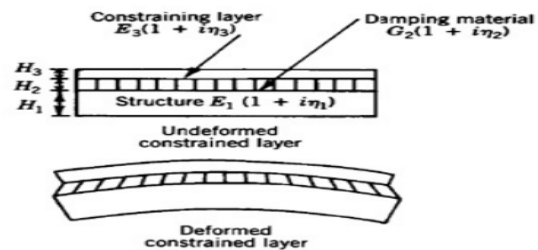
There are two main types of passive damping, the first one is unconstrained layer damping (UCLD)[5] by adding a completely bonded damping layer on the surface of the vibrating structure, as shown in figure 1 but this method cannot be used in many mechanical structures due to lower damping ratios[6]. The second type In 1959, Kerwin, et al [7] first did a fundamental work in what is called passive constrained layer damping (PCLD), where the damping layer putted in between of the base material layer and constrained



layer that may be made of the same material of the base layer or different material as shown in figure 2. This



**Figure .1** Un constrained layer damping.



**Figure .2** Constrained layer damping.

type of damping reduces the efficiency of the system and increasing the net cost[8].

In our work we study other approach of damping system that used two layers of damping material one of them on the upper surface of the base vibrating element and the second one on the lower side to achieve higher damping characteristic, low cost, simplicity and enhanced reliability[9]

Damping material having different types according to the level of vibration energy dissipation or loss factor ( $\eta$ ) [10] that classified into:

- Conventional materials (that having low loss factor).
- Composite materials, which exhibited higher viscoelastic properties than conventional materials and having superior specific stiffness and strength[11].
- Viscoelastic materials, for which the correlation between stress and strain depend on time so that the behavior of this type of material depend on the loading history and that have highest loss factor comparing with the composite materials[12].

In our damping system we use viscoelastic material type (Dayd 606) in both damping layers

## 2. Calculating of natural frequency

The natural frequency is the frequency at which a system tends to oscillate in the absence of any driving or damping force. when an object vibrates at frequency equivalents  $s$  to natural frequency, its vibration amplitude increases significantly which could lead to irreparable damage. Determination of natural frequency could be estimated with several methods.

### 2.1. Numerical investigation

The process of developing the finite element model starts in the "Engineering Data section" in ANSYS workbench package version R15.0 [14], by assigning the mechanical properties of all materials used in building the FEM, The materials used are discussed in details in the next section.

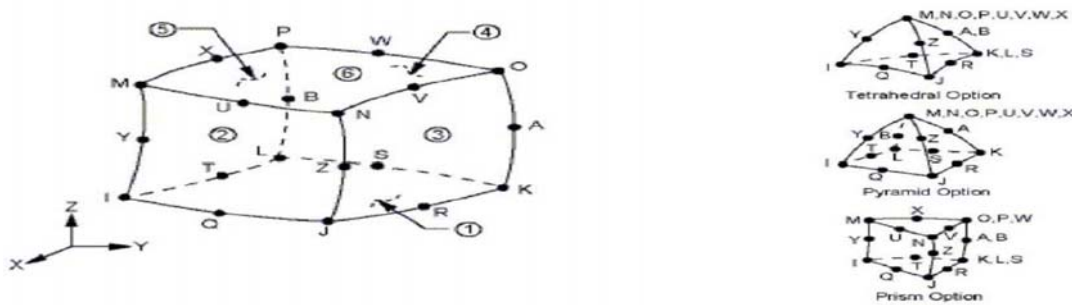
**2.1.1. Mechanical properties of FEM materials.** An aluminum (AL 5051) rectangular plate and polymer (dyad 606)

Table 1 shows the mechanical and geometrical properties of the plain plate and the damping material used in the (PTLD) treatment

**Table 1.** Mechanical and geometrical properties of the plain plate.

Layer	Thickness (mm)	Length (mm)	Width (mm)	Density ( $k_g/m^3$ )	Posion Ratio	Young's Modulus (Pa)
Aluminum alloy	1.2	300	50	2700	0.33	$70 \times 10^9$
Damping material	0.5	300	60	1120	0.49	$20 \times 10^6$

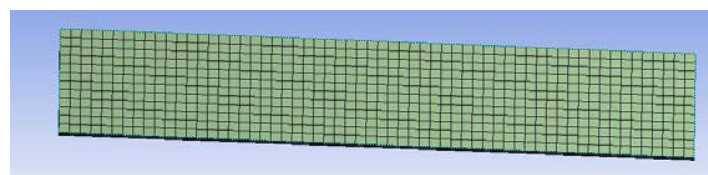
2.1.2. *Meshing of the plate FEM* .Meshing process is started by applying mesh tool to the developed 3-D model using fine mesh. The number of elements is decided after a convergence study made for the ANSYS model using mesh sensitivity analysis. The convergence isobtained at 5 mm element size. The element type is "solid 186" – 20 node homogenous structural solid, figure 3 shows the solid 186 element configuration. It is a higher order 3-D 20-node solid element that exhibits quadratic displacement behavior. The element is defined by 20 nodes having three degrees of freedom per node: translations in the nodal x, y, and z directions. The element supports plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities[15]. The geometry, node locations, and the element coordinate system for this element are shown in figure 4



**Figure. 3** Solid 186 element configurations.

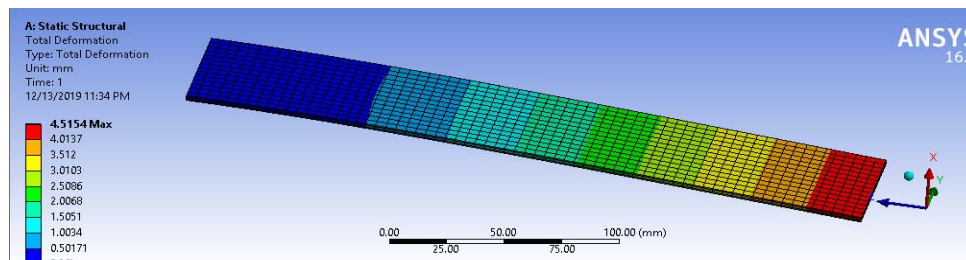
**Table 2.**Number of elements and nodes used for the plate mesh.

MODEL TYPE	NUMBER OF ELEMENT	NUMBER OF NODES
The plate	720	5403
The plate with polymer	2160	16209



**Figure. 4** The plate FEM with mesh.

*2.1.3. Static analysis.* Static structural analysis tool is performed to apply the steady inertia loads such as plate gravity, and used to determine the plate displacements, stresses, strains, etc... a cantilever boundary conditions are applied to the FE model. Figure 5 Shows the plate tip displacement due to gravitational load



**Figure. 5** The plate tip displacement due to gravitational load.

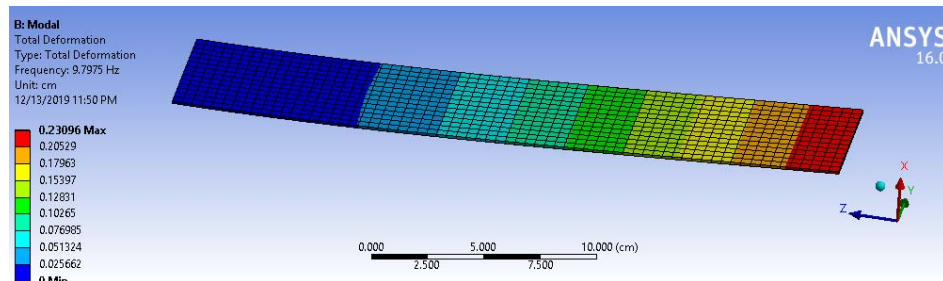
#### *2.1.4. Modal analysis*

Modal analysis is used to determine the vibration characteristics (natural frequencies and mode shapes) of the cantilever plate and the plate with polymer. It considered as a starting point for more detailed dynamic analysis, such as transient dynamic analysis, harmonic response analysis, or spectral analysis.

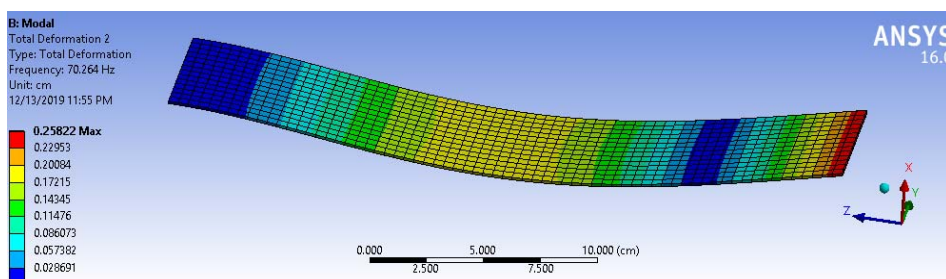
When frequencies of dynamic load match one of natural frequencies, then resonance takes place, which is considered from the main critical parameter of dynamic structure design. Mode shapes hereinafter called modes, and natural frequencies (NF) of the plate and the plate with polymer are obtained by modal analysis tool. The plate and the plate with polymer mode shape are shown in figures 6, 7, 8, 9. The values of natural frequencies of the plain plate and the plate with polymer layers are tabulated in Table.3

**Table 3.** The plate and the plate with polymer mode shape.

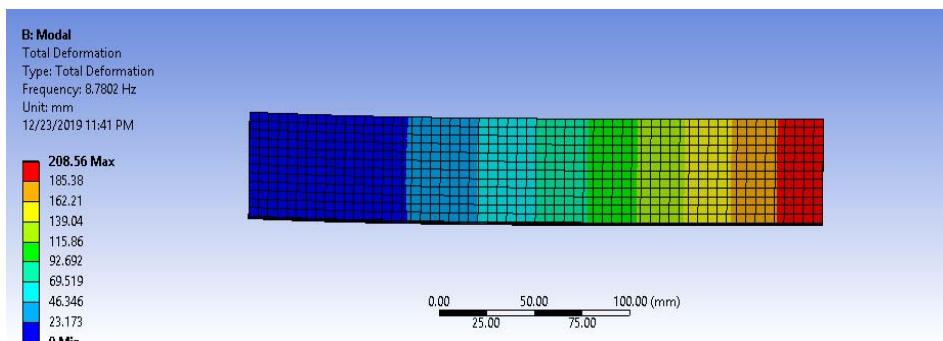
Mode Shape	Mode Shape Type	Plain Plate	Plate with Polymer
1	Bending	9.7975	8.7
2	Bending	70.267	60.3



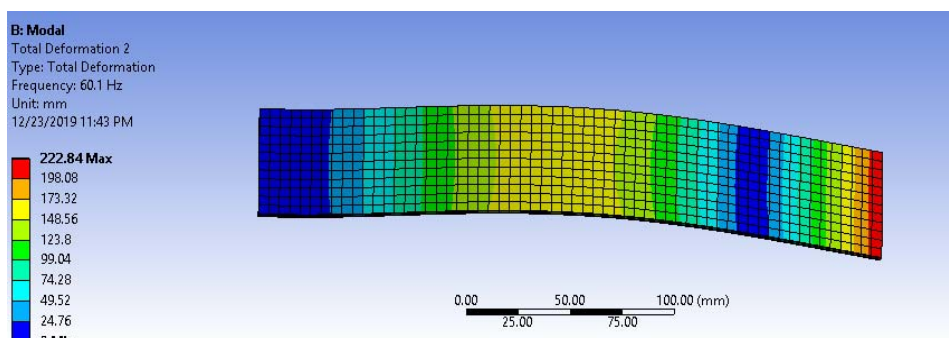
**Figure. 6** First mode shape of plain plate.



**Figure. 7** Second mode shape of plain plate.



**Figure. 8** First mode shape of plate with polymer.



**Figure.9** Second mode shape of plate with polymer.

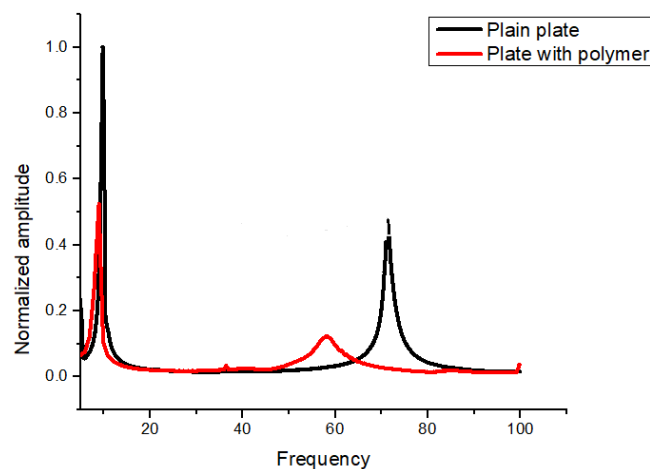
### 2.2. Experimental investigations

The experimental verification of finite element analysis is conducted. The experimental system layout is shown in figure 10. In this experiment, the plate is clamped from one end onto a shaker 4808 with fixation support, and the second end is free.

Two accelerometers type PCB PIEZOTRONICS are attached to the plate by a suitable adhesive, one is fixed on the plate free end to measure the output response, and the other is fixed on the shaker fixation support to measure the input excitation. The shaker B&k type 4808 is driven by power amplifier B&k type 2712, which amplifies the sine sweep, signal (0-200) Hz coming from waveform generator. Chasses of NI compact DAQ-9188 is used for data acquisition system and PC with Lab View program is used to acquire and analyze the data. The most suitable technique used for characterizing the plate material properties in the frequency range (from 1 up to 1000 Hz) is the frequency response technique[13]. FRF results are processed by comparing the measured acceleration values taken by the accelerometer mounted on the plate free end (output) with the measured acceleration results taken from the accelerometer mounted on the shaker support (input). To obtain exact experimental results, ten repeated measurements are recorded for each experimental set up and the average values are considered as shown in figure 11.



**Figure.10** The experimental setup



**Figure.11** First and second natural frequency with and without polymer

### 3. Finite element modeling verification

The theoretical results are used to verify the FE model by conducting a comparison between the plainplate and the plate with polymer numericallyand experimentally. The comparison between the resonant frequencies of the numerical simulation and the experimental result are shown in table 4.

The results are in good agreement with an 4 % maximum error. The contrast among FE and exploratory outcomes can be credited to the accompanying:

- Distinction in material properties among FE and actual plate.
- Difference in boundary conditions; in FE model, the end is fixed, where in testing it is clamped and there may be still some relative freedom.
- -Difference in the portrayal of the accelerometer; in FE model, it is a concentrated mass, where in testing it is stuck to the plate.
- -Effect of environmental conditions during tests (temperature...etc.), which are not accounted for in simulation.
- In the FE model, just the shaker impact is signified (the environmental vibrations are not represented).

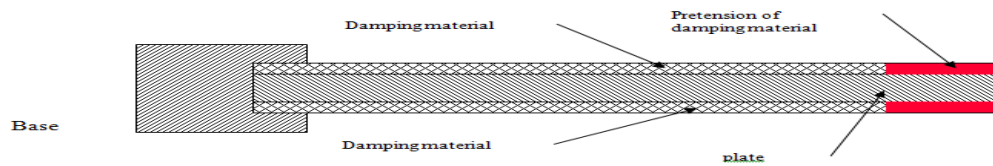
**Table 4.** The plate resonant frequencies(Hz)(numerically/experimentally)

MODE SHAPE	THE PLAIN PLATE		THE PLATE WITH POLYMER	
	Numerical	Experiment	Numerical	Experiment
1	9.7975	9.89	8.7	8.72
2	70.264	71.1	60.1	59.8

### 4. Casestudy

The main purpose of this technique is to illustrate numerically the fundamentals associated with damping the vibration of plate using a new method of passive control layer damping treatment. The developed new model is based on the idea of generating uniformly distributed forces on two layers of the damping materialexperiencing cyclic motionis shown in figure 12. The lateral components of the exerting forces always affecting in direction opposing the angle of rotation of plate. These lateral components have a complex form due to the inherent damping of the elastic layer bonded to the structure fibers. It will be demonstratednumerically that a more than 95% reduction in the peak of the fundamental bending mode of a cantilevered plate can be obtained by virtue of this new technique.

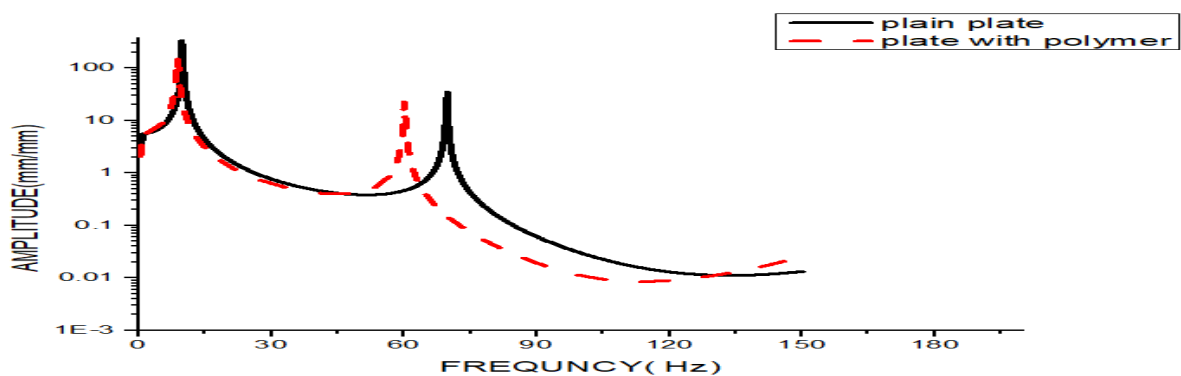




**Figure. 12** Plain plate with Pretension Layer assembly.

#### 4.1 Harmonic response analysis

A harmonic response tool is applied to study the plate FRF for the first two bending modes. It is carried out for the plate with /without the damping material. A transverse excitation force of 1 N with sine sweep signal varies from 0 to 200 Hz is applied at the plate structure. FRF of a structure otherwise called a transfer function is a ratio of the output signal to the input signal in frequency domain. Figure 13 shows the amplitude of the plain plate and plate with polymer at different resonant frequencies or called amplitude FRF of the plate and the plate with polymer, hereinafter called the plate and the plate with polymer FRF.



**Figure. 13** The plain plate and the plate with polymer (FRF).

### 5. Results and discussion

The response of the generated exerting force on the reaction of the Plate/PTLD system is assessed. The attenuation percentage is calculated for different Pretension forces values, where, the comparison is made with respect to the response of the plain plate as a fixed reference. The attenuation percentage is determined from the accompanying equation.

$$\Delta_n^t(\omega_n, L) = \frac{|\alpha'(\omega_n, L)| - |\alpha(\omega_n, L)|}{|\alpha(\omega_n, L)|} 100\% \quad [16]$$

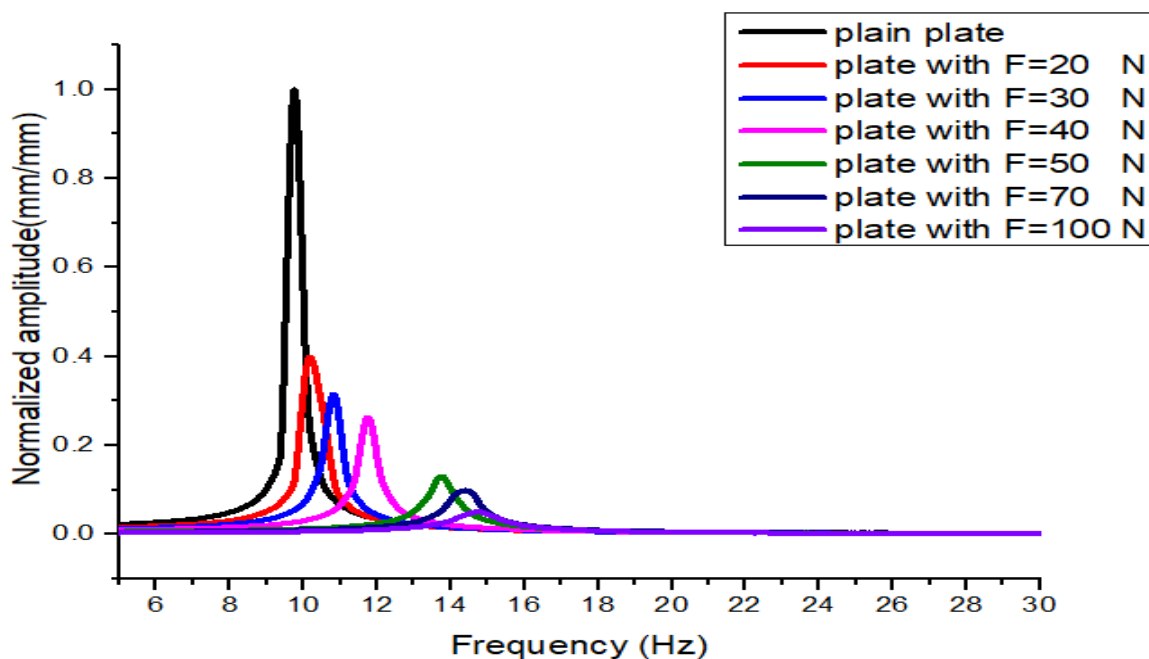
The performance of the system without the effect of the forces should be study, at this moment, the system response is that response of the Plate/UCLD system, and the damping produced is due to the direct shear

strains in the damping layers, in addition to the damping of the base plate. Figure 13 gives the comparison between the Plate with polymer and the corresponding plain Plate.

Figure 14 gives the comparison between the magnitudes of the frequency response functions of the system at different values of the Pretension force within the elastic range of the damping material. These figures indicate the high damping efficiency of the new approach. Table 5 gives the relation between the value of the Pretension force and the attenuation percentage of the new model at 1<sup>st</sup> resonance.

**Table 5.** The attenuation percentage at different Pretension force.

Pretension force [N]	Attenuation ratio[%]
0	52.3%
20	61.9%
30	69.3%
40	74.3%
50	86.8%
70	91.2%
100	95.2%



**Figure. 14** Plain plate compare to plate with polymer and pretension polymer.

## 6. Conclusion

A new passive control model used for suppression of the lateral vibrations of a flexible base excited cantilevered Plate is illustrated in the present paper, This new approach, which is called pre-tensioned layer damping (PTLD), is a sort of artificial damping techniques, which is based on the theory of energy dissipation from vibrating systems. The devolved model has a simple, reliable and inexpensive solution

for the Plate /PTLD system. At which the major advance in the attenuation of the vibration amplitude has been discussed. the damping ratio is enhanced without the usage of complications control devices that used in case of Active Constrain Layer Damping (ACLD) It is noticed that the higher the elastic Pretensionforces in the pre-tensioned layer damping material that increase the attenuation percentage and the damping efficiency.

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