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Effect of damping material thickness on vibration analysis in pretension layer damping process

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Abstract. Vibration reduction is a critical necessity in many fields of engineering, technology, and industry. As a result, there is a need for vibration management. To restrict or adjust the system's vibration response, a variety of techniques are employed. In recent years, there has been a lot of enthusiasm for the easy implementation of these vibration-control structures. The damping properties of viscoelastic materials tend to be excellent. Damping is determined by the material's ability to dissipate energy. To minimise the vibration of vibrating surfaces, viscoelastic materials are commonly used. In this study, viscoelastic material (Dyad 606) is applied on the AL plate in the form of free layer damping and pretension layer damping. First aluminium structure using free layer damping and another one using pretension layer damping by applying tension load (two layers of damping plates, one on the top surface of the Al alloy and the other on the lower face). Passive vibration damping of the plate is achieved. These layers are influenced by axial uniform distributed load. The damping behaviour of the AL Plate is discussed in relation to the thickness variation of the pre-tension damping material. The results show that the loss factor and the attenuation percentage of the structure with pretension layer damping are increased as compared to free layer damping. It is also found that the most effective thickness of the damping material to increase the damping capacity of the framework is around half the base plate thickness

1. Introduction

In different branches of engineering, technology, and industry, the issue of reducing the amount of vibration in a system arises. Vibrations that cause harm have a major effect on task execution, performance, and precision. [1]. As a result, vibration control is essential. The vibration response of the device is restricted or adjusted using a variety of techniques. In the last few years, there has been a lot of interest in bringing these vibration-control systems into operation. [2]. First and foremost, if an unwanted vibration problem needs to be regulated, it is desirable and always appropriate to comprehend its entire nature. It can then be determined that if an additional damping mechanism is to be successful, the increased damping must be much greater than the original damping. [3]. Then it's time to figure out if passive or active control strategies are the most effective way to solve the problem. To make the vibrational system less sensitive to its vibrational environment, passive control necessitates adjusting the rigidity, mass, and damping of the system. If one or more structural resonances overpower the unwanted vibration and acoustics, they can be effectively balanced by increasing the system's damping. For most applications, four types of noise and vibration management may be used: absorption, barriers and enclosures, structural vibration damping, and vibration insulation. [4]. the increased damping must be incomparably greater than the initial damping to be appropriate for an additional damping device. The most popular method of improving damping is to introduce highly damped polymeric material into the structure at key locations. The structure must interact with the polymer in such a way that the polymer dissipates as much energy as possible. In the common method, two forms of vibration damping and acoustic control treatments are provided [5]. The first one is referred to as thorough damping. This procedure is also known as the unregulated or free treatment of layer damping. When the structure is flex-mounted, the treatment is coated on one or both sides of the structure to cause tension-compression deformation of the damping material. But, due to the lower damping ratios, this technique cannot be used in a number of mechanical structures. [6]. The second type of damping is referred to as a shear type. The damping type is more effective for a given weight than the prolonged damping treatment. This efficiency is, however, balanced by more study and application complications. The therapy is identical to that of the unrestrained layer, with the exception of another layer limiting the viscoelastic content. Therefore, the extra layer will limit the viscoelastic material and deform the structure in shear when the structure is flexed [7]. Such damping decreases device performance and increases net costs.

Other damping systems with two layers of damping material, one on the top surface of the base vibrating component and the other on the bottom part, are investigated, to improve damping properties, reduce costs, simplify, and improve reliability [8].Damping material thickness is an effect of the vibration damping efficiency so that it is tested and the most effective thickness is clarified.

Damping is the dissipation of energy in the cyclical stress of a material or system. In any mechanical device, there are four major types of damping [9]:

- Viscous damping
- Magnetic damping
- material damping
- dry friction damping

2. Scope and objective

Structures that are both functional and lightweight are becoming increasingly necessary (Electric vehicles, engines, aircraft, boats, car engines, manufacturing equipment, and so on are examples of such products.). This criterion resulted in light structures shifting and/or vibrating at higher frequencies, producing higher temperatures and, as a result, increasing unwanted noise and oscillation levels. As a result, stronger vibration damping materials had to be found [10]. The aim of the optimum design of pretension layer damping is to maximise structural damping due to vibration modes' modal loss factors [11]. Here the maximization of damping is reported in the first free vibration mode. The damping intensification of pretension layer damping, and has been carried out as a thickness optimization of the polymeric sheet.

2.1. Damping material thickness

The vibration damping behaviour of the plate varies as the thickness of the damping material changes. However, it only increases dramatically near the damping material's glass transition temperature due to a substantial shift in the damping layers loss factor. [12]. If the damping layer is too thin or too thick, the damping effect is diminished. Maximize damping by increasing the thickness and/or rigidity of the material [13]. Since cost-effectiveness is often assumed when the damping film is too thin, the thickness of PTLD (which reflects the cost) is often considered a penalty. The following four types of plates are considered for analysis by taking into account the above references:

a) PTLD with VEM 0.3 mm.
b) PTLD with VEM 0.6 mm.
c) PTLD with VEM 0.9 mm.
d) PTLD with VEM 1.2 mm.

3. Numerical work

In ANSYS edition R15.0 [14], the method for building a finite element analysis begins in the 'Engineering data file.' In the second paragraphs, the materials used are discussed in depth, for example, by assigning material properties to all of the materials that were used in the FEM's design. Mechanical properties of FEM material. Base plate (AL 5051), and the damping layer (dyad 606) [8].

Table 1. The basic plate's mechanical and structural properties, as well as the damping material used in the (PTLD) procedure.

Layer	Thickness (mm)	Length (mm)	Width (mm)	Density (kg /m3)	Passion's Ratio	Young's Modulus (Pa)
AL	1.2	300	60	2660	0.33	70×109
DYAD 606	0.5	300	60	1120	0.49	20×106

3.1. Meshing of the plate FEM

The meshing method begins with the use of a fine mesh mesh tool on a shaped 3d geometry. After a mesh sensitivity analysis convergence study for the ANSYS model, the number of items is calculated. Convergence is done at a 5 mm element size.

The element's shape is "solid 186," which is a 20-node uniform structural solid. Figure 3 illustrates the solid 186 item's configuration. It's a polynomial displacement function with a 20-node higher-order 3-D solid feature. There are 20 nodes in the element, each with degrees of freedom. Descriptions of nodes in the x, y, and z directions. The factor supports plasticity, hyperelasticity, creep, stress stiffening, large deflection, and large strain capabilities. [15].This element's geometry, node locations, and element coordinate scheme are shown in Figure 2.

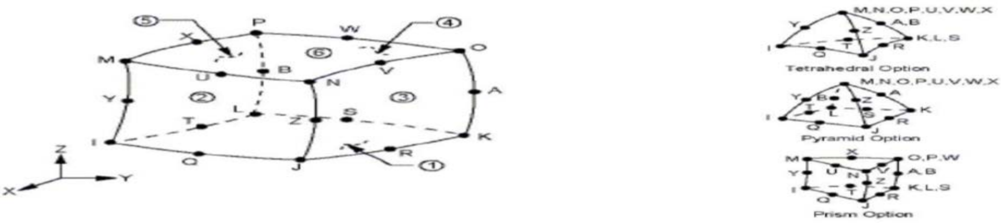

Figure 1. Solid 186 element configurations

Table 2.elements and nodes number used for the structure mesh.

model type	element number	nodes number
AL Alloy	722	5406
AL/UCLD	2161	16210

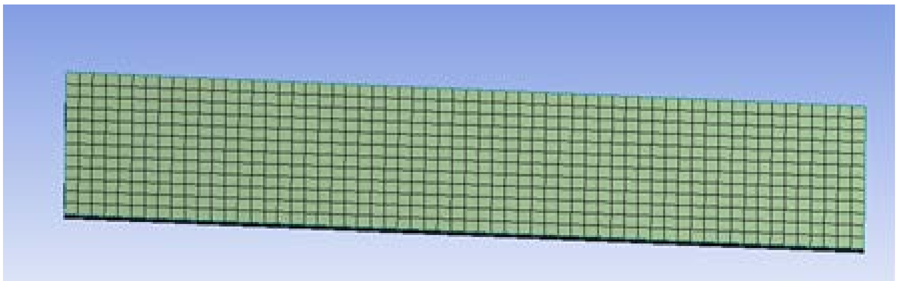

Figure 2. The final mesh of the plate FEM

3.2. Static analysis

In the application of continuous loads of inertia, such as plate gravity, the static structural analysis method is used to measure flat movements, stresses, pressures, and so on... Cantilever boundary conditions must be added to the FE model. Figure 3. Shows the plate tip displacement's gravitational strain.

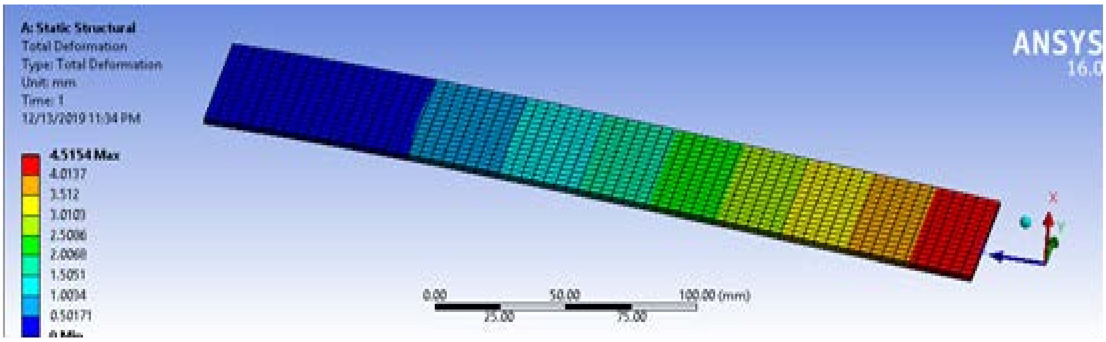

Figure 3. Gravitational load causes plate tip displacement.

3.3. Modal analysis

The vibrational characteristics of the platform and the platform using a polymer are classified using a modal analysis. (natural frequencies and modal shapes). It's been used as a jumping off point for more dynamic studies, such as transient, harmonic, or spectral dynamic analyses. Resonance occurs when dynamic load frequencies match one of the natural frequencies, which is the most important parameter in dynamic structure design. The modal analysis approach involves the following mode shapes, also known as modes, as well as the platform and polymer plate's Natural Frequencies (NF). The platform and the platform in polymer mode are depicted in the figures. The mode shape of the plate and the UCLD plate are shown in Table 3. The FRF of the current plate at various resonant frequencies is shown in Figure 7.

Table 3 .AL plate and the UCLD Plate mode shape.

Mode Shape	Type	AL Plate	UCLD Plate
1	Bending	9.7975 (Hz)	8.7 (Hz)
2	Bending	70.267 (Hz)	60.3 (Hz)

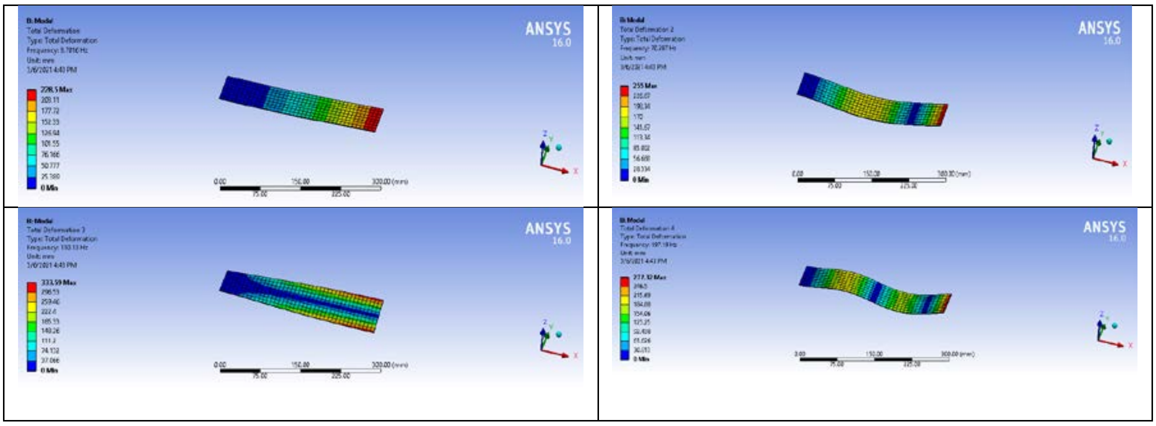

Figure 4.first four mode shape of AL plate

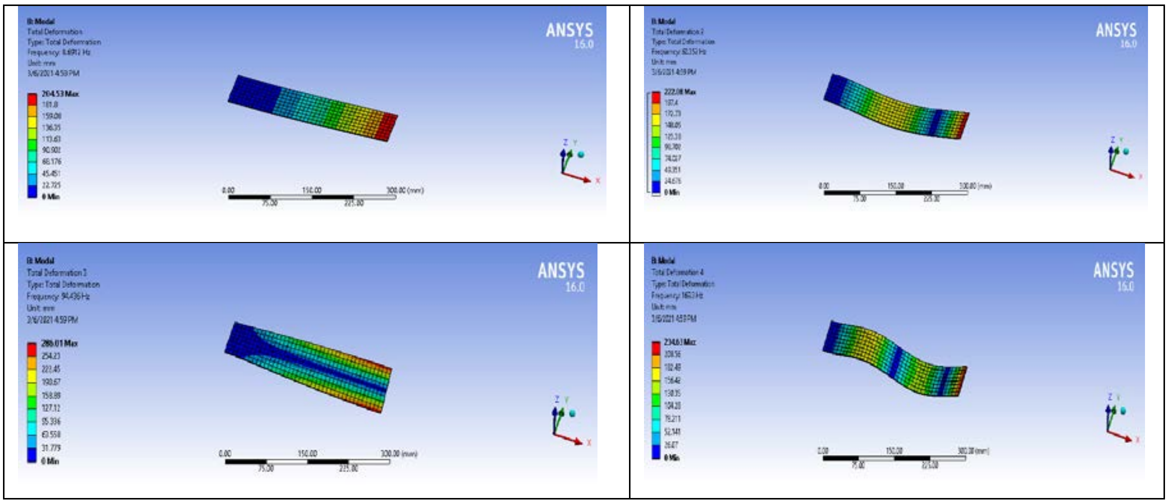

Figure 5. First four mode shape of UCLD

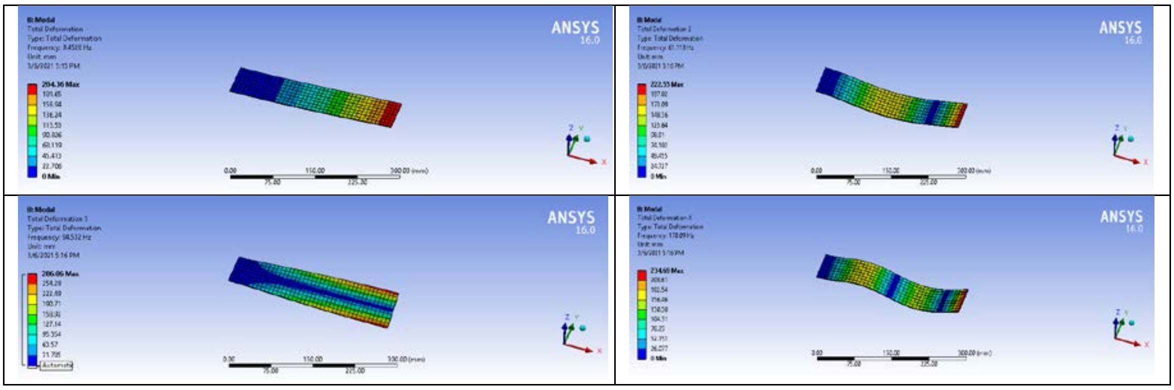

Figure 6.first four mode shape of PTLD f=5 N

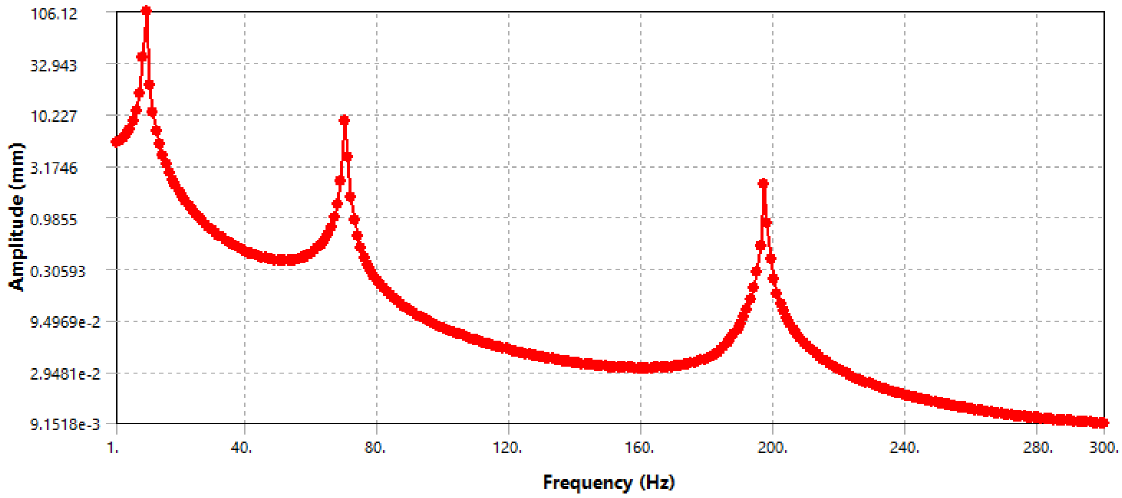

Figure7. Frequency response function (FRF) for the AL plate.

3.4. Verification of finite element modelling

AL plate was previously examined experimentally and analytically in comparison [8].

1. To account for weight due to gravitational acceleration (g = 9.81m/s2) and accelerometer mass modelled at the free-end mass level, the first step was to use the identical plate dimension as the reference by applying the same amount and distribution of the load applied obtained by the reference.
2. Table 4 compares the frequencies of the simulated results to the reference result, which can be referred to as the second step of testing.

Table 4. The comparison between the frequencies of the simulated work results and the reference results

No. of modes	Kind of mode shape	Resonant frequency (Hz)	
		Current model	Reference model
1	Bending	9.797	9.76
2	Bending	70.267	70.312

3.5. Parametric study:

3.5.1. Aluminum Plate with damping layer without pretension forces. The numerical analysis distinction between vibrational modes and FRF was shown in the previous paragraph, which had a good deal between the two models FE. The main goal of this study is to determine the smallest FRF amplitude that is proportional to the best damping layer thickness. We use two layers of damping material (DYAD 606) in this model, one on the top side of the base plate (AL 5056) and the other on the bottom side, to investigate the effect of damping material thickness on the FRF amplitude. Table 5 provides the results of the first natural frequency relation between the resonating frequencies and the loss factor for the numerical analysis. The comparison of various thicknesses of the vibration control layer effect of the plate is shown in Figure. The percentage attenuation is determined by the related equation.

$$\Delta_n^t(\omega_n) = \frac{|\alpha(\omega_n)| - |\alpha^t(\omega_n)|}{|\alpha(\omega_n)|} 100\%$$

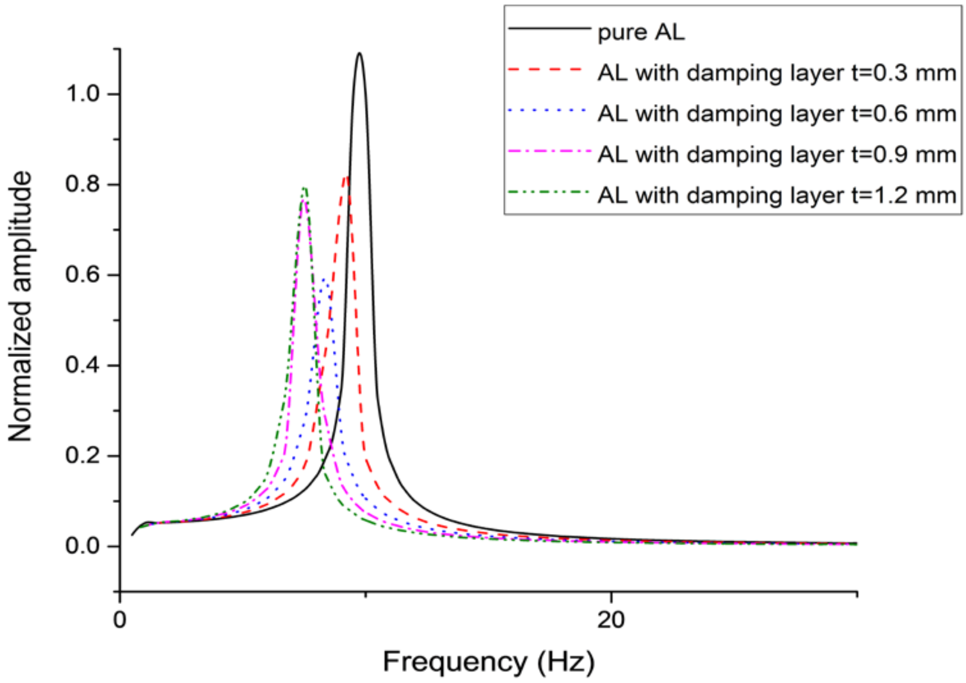

Figure 8. The comparison between various thicknesses of the damping layer effect without pretension force on the control of vibration of the plate

Table 5. The comparison between the first natural frequency and loss factor of the numerical simulation work results for the four models without pretension layer damping F=0 N.

No.	Plate type	Natural frequency (Hz)	Modal loss factor	Attenuation ratio[%]
1	UCLD PLATE -1	9.19	0.011	20%
2	UCLD PLATE -2	8.75	0.0204	41%
3	UCLD PLATE -3	8.112	0.0159	24.32%
4	UCLD PLATE -4	7.75	0.0125	22.1%

3.5.2. Aluminum Plate with damping layer with pretension forces. In this case, we demonstrate how to adjust the damping material thickness of the vibration control of the plate by using two layers of polymer plates (Dyad 606), which are influenced by an axial uniform distributed force when tension is applied, as shown in figure (11)[8]. With varying damping thickness, we apply a 5 N pretension force to achieve the smallest vibration amplitude. Figure 9 illustrates how the thickness of a damping material influences the vibration control of a flat. Table 6 compares the frequencies and the loss factor of the numerical simulation work's results for the first natural frequency.

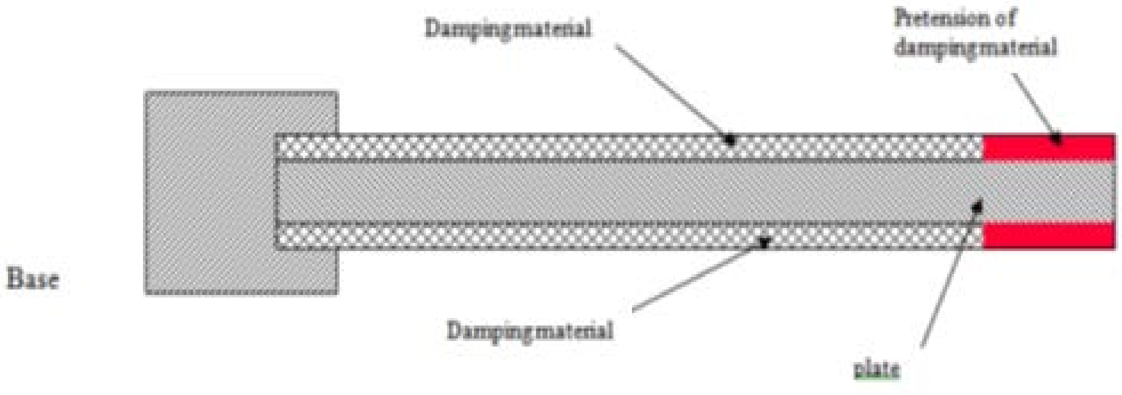

Figure 9. Base plate with pretension layer damping assembly

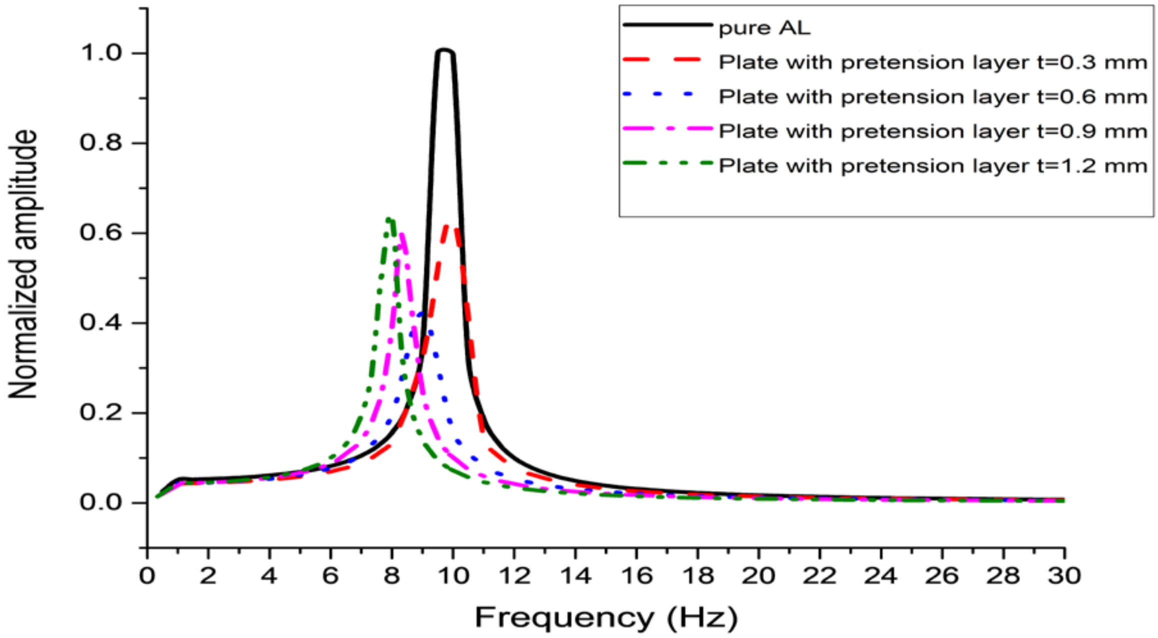

Figure10.the comparison between various thicknesses of the damping layer effect with pretension force F=5 N.

Table 6. The comparison between the first natural frequency and loss factor of the numerical simulation work results for the four models for pretension layer damping F= 5N

Sr. No.	Plate type	Natural frequency (Hz)	Modal loss factor	Attenuation ratio [%]
1	PTLD PLATE -1	9.85	0.021	41.2%
2	PTLD PLATE -2	9.2	0.0298	59.5%
3	PTLD PLATE -3	8.78	0.0195	38%
4	PTLD PLATE -4	8.5	0.0185	39.5%

4. Conclusion
The impact of a proper damping thickness on the loss factor and attenuation percentage of the PTLD plate is discussed in this paper. The loss factor results are obtained to the PTLD -1, PTLD-2, and PTLD-3, PTLD-4, from a numerical analysis using the ANSYS programme. It is notes that the most important increasing of loss factor in the PLTD-2. The amplitude of the vibration in plate-2 decreases as the modal loss factor increases. Vibration damping is enhanced when the damping layer thickness is about half of the plate thickness, whether pretension damping is used or not. The thickness ratio of the damping material layer to the plate must be calibrated to achieve the best damping in the vibrating plate.

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