Design and Development of Passive Damper

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***Abstract*—There are various types of viscoelastic materials are available for the absorption of vibrations known as antivibration passive damper, but these viscoelastic materials has some disadvantages such as less life, degradation rate is high, melting point is less. To avoid these disadvantages, instead of single material use. The composite materials as combination of two viscoelastic materials are use as per their different properties. The comparative analysis of these viscoelastic materials as per the various parameters such as maximum load withstand capacity, vibration displacement, sound level, compressive strength. On these parameters the best passive damper is choose for low, medium and high load applications. The DC speed variable motor is use for this testing by FFT Analyser.The test of vibration displacement, sound level are check by FFT Analyzer.**

***Key words*— Transmissibility, FFT, UTM.**

I. INTRODUCTION

In most cases vibrations are undesirable. Vibration of cars and carriages, motors and machine tools, oil and gas platforms, buildings and constructions in a zone of seismic activity, undesirable vibrations of laboratory tables (especially optical), setups, etc. In all these cases an object has to be isolated from the source of vibrations.

A vibration isolator typically comprises a resilient element attached to a mounting plate at each end. Resilient elements include rubbers, elastomers, polymers, metal springs, corks, felts and air bags. The dynamic properties (stiffness and damping) of the isolator determine the level of the transmission of vibration through an isolator. Vibration is isolated by placing properly chosen isolation materials between the vibrating body and the supporting structure. The effectiveness of isolation is measured in terms of the force or motion transmitted at the point of exposure from the source. The less force or motion transmitted the greater is the isolation. It also helps in protection of health through vibration and structureborne noise insulation means it help in environment protection.

Commonly, the solution to the vibration isolation problem is to use antivibration mounts or viscoelastic coatings on structural elements to enhance the damping. These solutions lead to weight and cost penalties and the effectiveness of such measures diminishes with increasing levels of vibration, making increasingly stringent noise and vibration targets more difficult to meet.

In recent years a number of active and passive vibration isolation solutions have been proposed and tested. Some of the more successful solutions include Hexapod or Stewart Platforms, dual-chamber pneumatic springs, passive D-strut systems, zero-spring-rate mechanisms, voice coil actuators, open/closed cell foam designs, smart structures and shape memory alloy isolators. Perhaps, one of the most effective designs has been the soft ride multiFlex whole-spacecraft vibration isolation system which consists of a series of identical isolator elements. An alternative approach to the above-mentioned methods is to design structures with inherent vibration isolation properties. Therefore, a generic structural design capability is required through which beneficial vibration characteristics can be built into structures, while their ability to achieve highly accurate measurements or alignment is retained, along with their static load carrying capacity.

Rubber as an elastic medium is probably the most universal material used vibrational damping. Instead use of rubber as single material the other viscoelastic materials such as polyeurathe, epoxy glass, cork and felt are use in combination for better absorption of vibration. Its special properties render it particularly suitable for damping and springing elements.

J. C. Snowden [1] discussed the rubber like materials, their internal damping and role in vibration. The internal damping and the dynamic elastic moduli of rubber like materials depend upon frequency and temperature is described in detail. They relate to the low damping materials natural rubber, natural rubber reinforced with carbon black, and SBR rubber; and to the high-damping materials Thiokol RD rubber, butyl rubber reinforced with carbon black, and plasticized polyvinyl acetate. The influence upon the transmissibility curves of nonlinear rubberlike behavior is discussed briefly. Results show the case of high-damping rubbers, the simple and more complex theories are found to be in good agreement. In the case of rubbers with relatively small damping, the effectiveness of the isolators at high frequencies is overestimated by the simple theory although, at these frequencies the transmissibility of the isolators will normally remain a small quantity. D. I. G. Jones [2] proposed techniques for measuring damping properties of thin viscoelastic layers, for performing the difficult task of measuring the complex modulus properties of viscoelastic materials when they are available only in the form of thin layers, such as damping tapes, attached securely to thin sheets of metal. The method involves measuring the response of a simple beam with several layers of the damping tape attached

to the entire surface and comparing the results with prior tests on the same configuration with another viscoelastic material having known properties. Results are verified by comparison with data obtained by using a resonance technique. It is shown that the present approach is simple and effective to use, and that the limitations are simple to understand and avoid. Wu et al. [3] developed hybrid vibration absorber with virtual passive devices, vibration control scheme integrating a passive mass–spring resonator and a linear actuator. A control algorithm is devised to convert the actuator into an additional set of virtual mass-spring structure of programmable characteristic frequency. The relative motion between the primary body and the reaction mass is measured, as well as the acceleration of the reaction mass. This hybrid dynamic vibration absorber is capable of neutralizing a harmonic disturbance regardless of the detailed dynamics of the primary structure and other passive elements. Stability analysis leads to a simple, explicit stability criterion. Distribution of the counter-disturbance force between the active and passive devices is analyzed, and the transient performance is also investigated. Real-time experiments as well as numerical simulations are conducted to confirm the effectiveness of the proposed scheme. Fan et al. [4] investigated experimental study of the effect of viscoelastic damping materials on noise and vibration reduction within railway vehicles, interior noise and vibration reduction has become one important concern of railway operating environments due to the influence of increased speeds and reduced vehicle weights for energy efficiency. Three types of viscoelastic damping materials, bitumen based damping material, water based damping coating and butyl rubber damping material, were developed to reduce the vibration and noise within rail way vehicles. It is found that the reduction effect of damping treatments depends on the running speed. The unweighted root mean square acceleration is reduced for the carriage treated by bitumen based as well as water based damping materials and water based damping material, respectively. The first two materials reduce vibration in a wider frequency range of 63–1000Hz than the last. It turns out that the damping treatments of the first two reduce the interior noise level by 5–8dBA within the carriage and the last damping material by1–6dBA.However, the specific loudness analysis of noises shows that the noise components between 125 and250Hz are dominant for the overall loudness, although the low frequency noise is noticeably decreased by the damping materials. The measure of loudness is shown to be more accurate to assess reduction effect of the damping material on the acoustic comfort. Rao [5] presented applications of viscoelastic damping for noise control in automobiles and commercial airplanes. Passive damping technology using viscoelastic materials to control noise and vibration in vehicles and commercial airplanes is described. Special damped laminates and spray paints suitable for mass production and capable of forming with conventional techniques are now manufactured in a continuous manner using advanced processes. These are widely used in the automotive and aerospace industry in a variety of applications to reduce noise and vibration and to improve interior sound quality. Passive damping using viscoelastic materials is simpler to implement and more cost-effective than semi-active and active techniques. Du et al. [6] discussed control of internal resonances in vibration isolators using passive and hybrid dynamic vibration absorbers. They discussed methods to improve isolator performance by controlling Internal Resonances (IRs), also referred as wave effects, in vibration isolators. The IRs are associated with the isolators' internal elastic motions that are due to the inertia existing in practical vibration isolators. It is well known that the IRs degrade the isolator performance as predicted by ideal massless isolator models. This degradation could be as high as 20–30 dB in the force transmissibility at the IR frequencies and 10–20 dB in the overall noise radiation from the foundation in the audible frequency range. Ibrahim [7] studied recent advances in nonlinear passive vibration isolators, the theory of nonlinear vibration isolation has witnessed significant developments due to pressing demands for the protection of structural installations, nuclear reactors, mechanical components, and sensitive instruments from earthquake ground motion, shocks, and impact loads. In view of these demands, engineers and physicists have developed different types of nonlinear vibration isolators. Comprehensive assessment of recent developments of nonlinear isolators in the absence of active control means. It does not deal with other means of linear or nonlinear vibration absorbers. Specific types of nonlinear isolators are discussed, including ultra-low-frequency isolators. For vertical vibration isolation, the treatment of the euler spring isolator is based on the post-buckling dynamic characteristics of the column elastic and axial stiffness. Nonlinear viscoelastic and composite material springs, and smart material elements are described in terms of material mechanical characteristics and the dependence of their transmissibility on temperature and excitation amplitude. Pan and Zhang [8] proposed new method for the determination of damping in occurred composite laminates with embedded viscoelastic layer method for the determination of damping incocured composite laminates with embedded viscoelastic layer is developed based on mode superposition and modal strain energy method. The calculated damping value is not modal loss factor but a combination of damping from the contributing modes. The dynamic mechanical properties of the viscoelastic material occurred with composites were investigated and were substituted in the present method for calculating the damping incocured composites.

The rubber pad or any viscoelastic material is used as the vibration absorber, but it get attached with C.I. metal. The rubber pad has some disadvantages while using as vibration absorbers, so that the new vibration absorbers is the need for future work. The single viscoelastic material as passive damper has many disadvantages so it need to be overcome by the combination of those. To create different combination of viscoelastic materials, suitability for damping and loads.

Design and fabricated of experimental set up for vibration analysis. To create combination of viscoelastic materials for the reduction of vibration. Determine load deflection curve identified under different loadings. To scientifically designed

combination of resilient material that isolates shock and vibration and reduce acoustic noise. Determine deflection for gradually applied load by using universal testing machine (UTM) for different combinations.

II. DESIGN AND EXPERIMENTATION

A. Vibration Isolation

The essential properties of an isolator are natural frequency (developed by the spring rate or stiffness) and an energy dissipating mechanism known as damping. In some types of isolators the stiffness or natural frequency and damping properties are contained in a single element such as elastomers, cork, rubber mats, etc. Other types of isolators may have separate means of providing stiffness and damping as is the case with air springs (pneumatic isolators) and coil steel springs which are relatively undamped until used in conjunction with auxiliary damping elements such as orifice flow restrictors and viscous dampers.

The purpose of damping in an isolator is to reduce or dissipate energy as rapidly as possible. Damping is also beneficial in reducing vibration amplitudes at resonance. Resonance occurs when the natural frequency of the isolator coincides with the frequency of the source vibration.

The ideal isolator would have as little damping as possible in its isolation region and as much as possible at the isolator's natural frequency to reduce amplification at resonance. Damping however can also lead to a loss of isolation efficiency.

The various types of isolating materials are :

1) Rubber Pad

The rubber is generally used for vibration isolation, where load is light and frequency of vibrations is high. The type of loading may be compression or shear in nature. Its properties are influenced by heat, gasoline, oil etc. Therefore, it cannot be used at high temperature in the presence of gasoline and oil. Rubber pad forming is a metal pressing process which is used to press steel metal with the help of die and rubber block.

2) Felt

Felt is used as a vibration isolator for heavy loads and low frequency vibrations. Felt is highly resilient, retaining its strength. Wool Felt is chemical resistant. Synthetic Felt and Blended Felt can be treated for flame resistance.

3) Cork

These are obtained by condensation polymerisation of epichlorhydrin and polyhydroxy compound. Epoxy resin is cross linked plastic by addition of a hardener.

4) Polueurathane

PU are formed by chemical reaction between a di/poly isocyanate and a diol or polyol, forming repeating urethane groups, generally, in presence of a chain extender, catalyst, and/or other additives.

B. Force Transmissibility (T_r)

The values of Force and Deflection are taken from Universal Testing Machine.

1) Polyurethane and rubber pad

Table 1 shows the data for Polyurethane and rubber pad.

TABLE I
DATA FOR POLYURETHANE AND RUBBER PAD

Damping factor (ξ)	0.058
Static deflection (δ)	1 mm
Force (F)	1055 N
Speed (N)	1000 rpm
Stiffness (K)	10550 N/m
Mass (M)	10 kg

$$\omega = \frac{2\pi \times 1000}{60}$$

ω =104.7 rad/s

K= Stiffness in N/m;

$$K = \frac{F}{\delta}$$

$$\omega_n = \sqrt{\frac{K}{M}}$$

ω_n = 32.48 rad/s

$$T_r = \frac{F_T}{F_o}$$

$$T_r = \frac{\sqrt{1 + (2\xi \frac{\omega}{\omega_n})^2}}{\sqrt{\left[1 - (\frac{\omega}{\omega_n})^2\right]^2 + \left[2\xi \frac{\omega}{\omega_n}\right]^2}}$$

Where;

ω =Angular velocity,

ω_n =Natural frequency,

ξ =Damping factor

F_T =Transmitted Force,

F_0 =Impressed Force

T_r = 0.1138

Here, $\frac{\omega}{\omega_n}$ = 3.22 > $\sqrt{2}$

So that transmissibility is better.

1) Polyurethane and epoxy glass

TABLE II
DATA FOR POLYURETHANE AND EPOXY GLASS

Damping factor (ξ)	0.0625
Static deflection (δ)	1 mm
Force (F)	1055 N
Speed (N)	1000 rpm
Stiffness (K)	10550 N/m
Mass (M)	10 kg

$$\omega = \frac{2\pi \times 1000}{60}$$

= 104.7 rad/s

$$\omega_n = \sqrt{\frac{K}{M}}$$

$$\omega_n = 32.48 \text{ rad/s}$$

$$T_r = \frac{\sqrt{1 + (2\xi \frac{\omega}{\omega_n})^2}}{\sqrt{\left[1 - (\frac{\omega}{\omega_n})^2\right]^2 + \left[2\xi \frac{\omega}{\omega_n}\right]^2}}$$

$$T_r = 0.1148$$

Here, = 3.22 > $\sqrt{2}$

So that transmissibility is better.

3) Cork and felt

TABLE III
DATA FOR CORK AND FELT

Daming factor (ξ)	0.04
Static deflection (δ)	1 mm
Force (F)	7000 N
Speed (N)	1000 rpm
Stiffness (K)	7000 N/m

$$\omega = \frac{2\pi \times 1000}{60}$$

=104.7 rad/s

$$\omega_n = \sqrt{\frac{K}{M}}$$

$$T_r = \frac{\sqrt{1 + (2\xi \frac{\omega}{\omega_n})^2}}{\sqrt{\left[1 - (\frac{\omega}{\omega_n})^2\right]^2 + \left[2\xi \frac{\omega}{\omega_n}\right]^2}}$$

$$T_r = 0.07136$$

Here, $\frac{\omega}{\omega_n}$ = 3.96 > $\sqrt{2}$

So that transmissibility is better.

TABLE IV
COMPARATIVE ANALYSIS OF COMPOSITE MATERIAL ON TRANSMISSIBILITY

Sr. No.	Composite Material	Transmissibility T_r
1	Polyurethane and Rubber pad	0.0150
2	Epoxy glass and Polyurethane	0.0157
3	Cork and Felt	0.0125

D. Experimental Study

Using UTM to verify the practical applicability of the passive damper developed. The experimental setup is shown in Fig. 2. The change in natural frequency of real life structure due to the presence of the unbalanced force. It is generally observed that theoretical analysis of structure have some error with respect to actual structure, as model of structure generated for vibration and structure borne insulation through the most modern materials such as Cork, Felt, Rubber Pad, Glass Fibre, Epoxy Resin. The combination of different viscoelastic materials are used for to minimise the vibration and noise of respective machine. The size of each composite material is of 147×177×6 mm.The instruments used for experimental analysis are accelerometer, charge amplifier, Fast Fourier Transform (FFT) analyzer and related accessories. The FFT analyzer used have measuring range 10-200 dB, amplitude stability + 0.1 dB, frequency limit 1 Hz to 20 KHz.

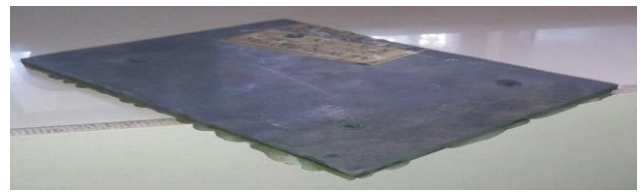

Fig. 1: Combination of Polyeurathane and Rubber pad.

Fig. 2: Combination of Cork and Felt.

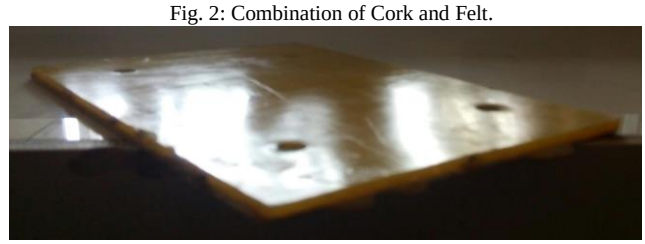

Fig. 3: Combination of Polyeurathane and Epoxy glass.

1) Load displacement test

Fig. 4: Test of deflection and maximum load of composite materials on UTM under compression.

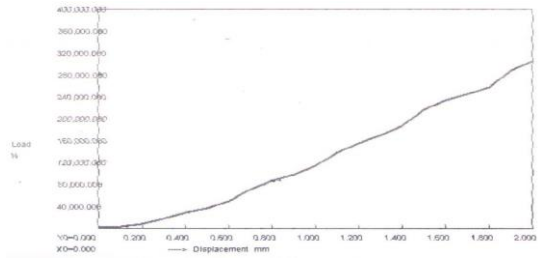

Fig. 5: Graph of Load Vs Displacement of Polyeurathane rubber pad material of test on UTM

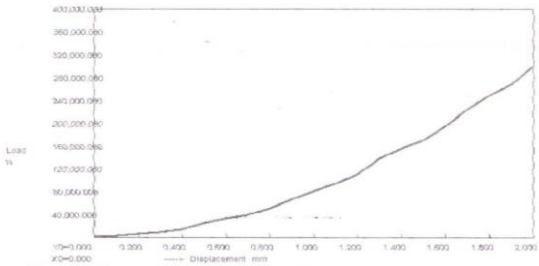

Fig. 6: Graph of Load Vs Displacement of Polyeurathane epoxy glass material of test on UTM.

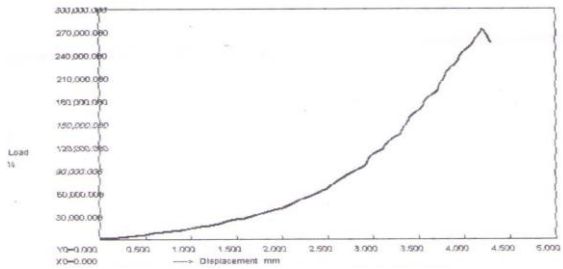

Fig. 7: Graph of Load Vs Displacement of Cork felt material of test on UTM.

TABLE V
COMPOSITE MATERIALS TESING ON UTM

Sr.No.	Composite material	Compressive strength in MPa	Maximum force in N
1	Polyeurathane and Rubber pad	14.782	319300
2	Polyeurathane and Epoxy glass	14.384	310920
3	Felt and Cork	13.578	293280

E. Experimental set up and procedure

Experimental set up was designed and fabricated for creating unbalance fault for vibration analysis. For the vibration measurement and analysis, four channel FFT analyzer (Make: Adash; Model: VA4Pro) is used. Vibration and noise analysis using scientifically designed combination of resilient material. The experimental set up is check that all the parts are on their good condition. The Speed variable DC motor of 1HP is used in this experiment. The disc of MS with diameter and thickness of 90 mm and 20 mm respectively. The 12 number of holes of 10mm diameter are created for unbalanced. The additional unbalanced load of 20 gm is attached to obtained the better vibration in the motor.

In case 1, taking reading on without damper and without unbalanced mass, for that electrical plug is switched on. The speed is keep on 1000 rpm. As one accelerometer placed on the motor for measuring the vibration level of motor and second is put on the base of motor for measuring the vibrations transfer to the base. The same procedure is follow in case 2 and case 3 of unbalanced without damper and unabalanced with damper respectively. This process is follow with polyeurathane-rubber pad, polyeurathane-epoxy glass and cork-felt. Compare this material combination on vibration absorb basis and gives the result.

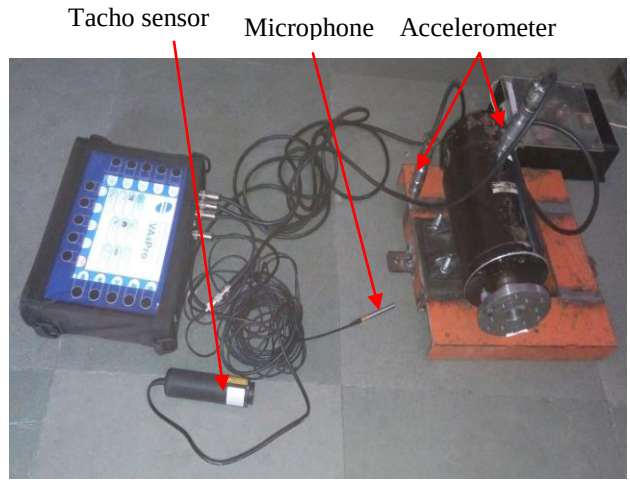

Fig. 8: Actual photograph of Experimental set up.

III. RESULTS

The test of DC speed variable motor with different composite viscoelastic materials by FFT Analyzer. The vibration displacement and sound level are obtained by accelerometer.

A. Vibration analysis using FFT analyzer

1) Case 1

1.1) Without damper and without unbalanced mass of motor speed 1000 rpm.

Fig. 9. Vibration reading of channel 1 & 2 of without damper and unbalanced mass.

1.2) Without damper and with unbalanced mass of motor speed 1000 rpm.

Fig. 10: Vibration reading of channel 1 & 2 of without damper and with unbalanced mass.

1.3) Without damper and without unbalanced mass of motor speed 2000 rpm.

Fig. 11: Noise level reading of channel 1 & 2 of without damper and with unbalanced mass.

Fig. 12: Vibration reading of channel 1 & 2 of without damper and without unbalanced mass.

1.4) Without damper and with unbalanced mass of motor speed 2000 rpm.

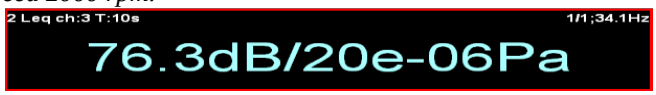

Fig. 13: Noise level reading of channel 1 & 2 of without damper and with unbalanced mass.

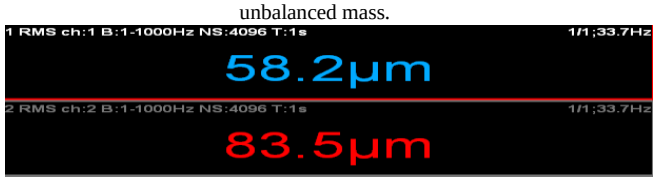

Fig. 14: Vibration reading of channel 1 & 2 of without damper and with unbalanced mass.

2) Case 2

2.1) With Polyeurathane - rubber pad damper and without unbalanced mass of motor speed 1000 rpm.

Fig. 15: Experimental noise level reading of channel 1 & 2 of with Polyeurathane rubber pad damper and without unbalanced mass.

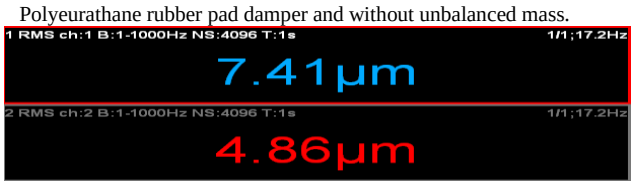

Fig. 16: Experimental time domain channel 1 & 2 of with Polyeurathane rubber pad damper and without unbalanced mass.

2.2) With Polyeurathane rubber pad damper and with unbalanced mass of motor speed 1000 rpm.

Fig. 17: Experimental noise level reading of channel 1 & 2 of with Polyeurathane rubber pad damper and with unbalanced mass.

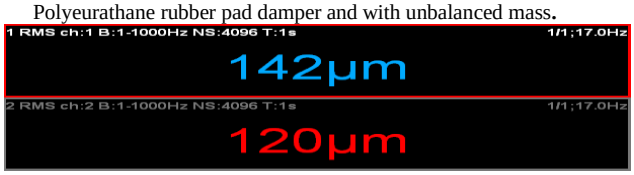

Fig. 18: Experimental time domain channel 1 & 2 of with Polyeurathane rubber pad damper and with unbalanced mass.

2.3). With Polyeurathane rubber pad damper and without unbalanced mass of motor speed 2000 rpm.

Fig. 19: Experimental noise level reading of channel 1 & 2 of with Polyeurathane rubber pad damper and without unbalanced mass.

Fig. 20: Vibration reading channel 1 & 2 of with Polyeurathane rubber pad damper and without unbalanced mass.

2.4) With Polyeurathane rubber pad damper and with unbalanced mass of motor speed 2000 rpm.

Fig. 21: Experimental noise level reading of channel 1 & 2 of with Polyeurathane rubber pad damper and without unbalanced mass

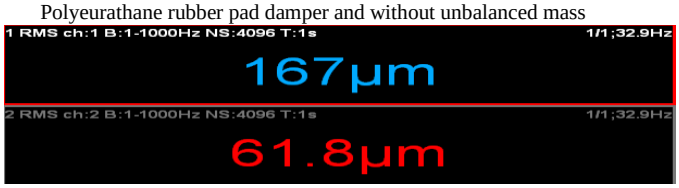

Fig. 22: Vibration reading channel 1 & 2 of with Polyeurathane rubber pad damper and with unbalanced mass.

3) Case 3.

3.1) With Polyeurathane epoxy glass damper and without unbalanced mass of motor speed 1000 rpm.

Fig. 23: Experimental noise level reading of channel 1 & 2 of with Polyeurathane epoxy glass damper and without unbalanced mass.

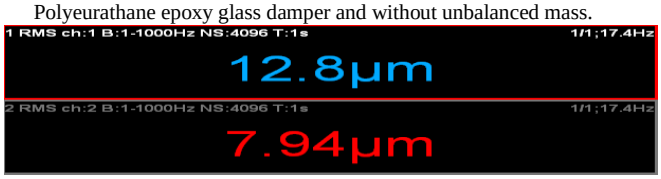

Fig. 24: Experimental time domain channel 1 & 2 of with Polyeurathane epoxy glass damper and with unbalanced mass.

3.2) With Polyeurathane epoxy glass damper and with unbalanced mass of motor speed 1000 rpm.

Fig. 25: Experimental noise level reading of channel 1 & 2 of with Polyeurathane epoxy glass damper and with unbalanced mass.

3.3) With Polyeurathane epoxy glass damper and without unbalanced mass of motor speed 2000 rpm.

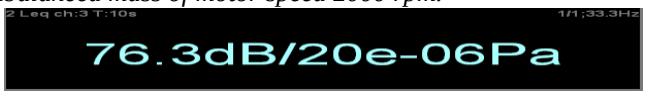

Fig. 26: Experimental noise level reading of channel 1 & 2 of with Polyeurathane epoxy glass damper and without unbalanced mass.

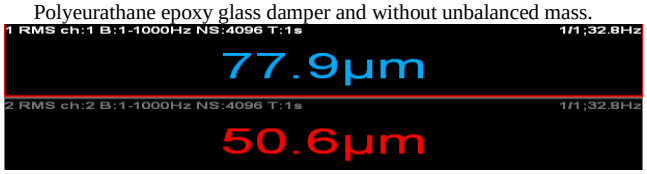

Fig. 27: Experimental channel 1 & 2 of with Polyeurathane epoxy glass damper and without unbalanced mass.

3.4) With Polyeurathane epoxy glass damper and with unbalanced mass of motor speed 2000 rpm.

Fig. 28: Experimental noise level reading of channel 1 & 2 of with Polyeurathane epoxy glass damper and with unbalanced mass.

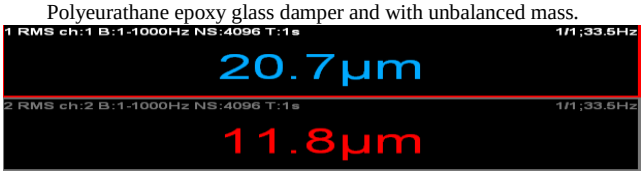

Fig. 29: Experimental time domain channel 1 & 2 of with Polyeurathane epoxy glass damper and with unbalanced mass.

D. Case 4

4.1) With Cork felt damper and without unbalanced mass of motor speed 1000 rpm.

Fig. 30: Experimental noise level reading of channel 1 & 2 of with Cork felt damper and without unbalanced mass.

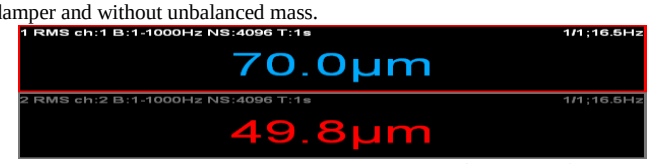

Fig. 31: Experimental time domain channel 1 & 2 of with Cork felt damper and without unbalanced mass.

4.2) With Cork felt damper and with unbalanced mass of motor speed 1000 rpm.

Fig. 32: Experimental noise level reading of channel 1 & 2 of with Cork felt damper and with unbalanced mass.

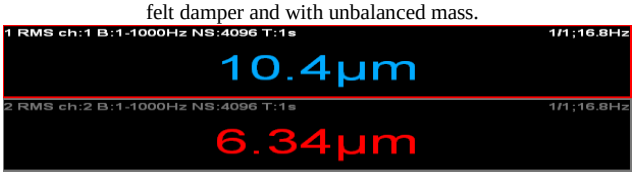

Fig. 33: Experimental time domain channel 1 & 2 of with Cork felt damper and with unbalanced mass.

4.3) With Cork felt damper and without unbalanced mass of motor speed 2000 rpm.

Fig. 34: Experimental noise level reading of channel 1 & 2 of with Cork felt damper and without unbalanced mass.

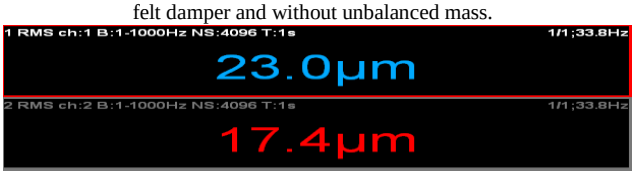

Fig. 35: Experimental time domain channel 1 & 2 of with Cork felt damper and without unbalanced mass.

4.4) With Cork felt damper and with unbalanced mass of motor speed 2000 rpm.

Fig. 36: Experimental noise level reading of channel 1 & 2 of with Cork felt damper and with unbalanced mass.

Fig. 37: Experimental time domain channel 1 & 2 of with Cork felt damper and with unbalanced mass.

TABLE VI
COMPARATIVE ANALYSIS OF COMPOSITE MATERIAL ON VIBRATION REDUCTION BASIS

Sr. No.	Composite material	Sensor	Displacement (µm) (With unbalanced Mass)	Vibration Reduction Displacement (µm)
1	Polyurethane and Rubber pad	Motor	167	105.2
		Base	61.8	
2	Epoxy glass and Polyurethane	Motor	20.7	8.9
		Base	11.8	
3	Cork and Felt	Motor	60	23.5
		Base	36.5	

TABLE VII
COMPARATIVE ANALYSIS OF COMPOSITE MATERIAL ON TRANSMISSIBILITY AND SOUND LEVEL BASIS

Sr. No.	Composite Material	Transmissibility (T_r)	Sound level dB(A)
1	Polyurethane and Rubber pad	0.0150	70.68
2	Epoxy glass and Polyurethane	0.0157	79.7
3	Cork and Felt	0.0125	79.5

The testing of displacements of motor vibration are shown in table no.4 also the sound levels taken simultaneously for each reading are shown in table no.5.The transmissibility also shown of each composite viscoelastic material in the same.

IV. CONCLUSION

As per the comparative study of different passive composite dampers on different parameters such as; load withstand capacity,transmissibility,sound level, vibration displacement.

It can be concluded that these materials are use as for different load conditions such as, for Low load applications (upto 293 KN) Cork felt is suitable. The Polyeurathane Epoxy glass can be better for the medium load applications (upto 310 KN). High load (upto 319 KN) can be sustain by Polyeurathane Rubber pad better than other two composite viscoelastic materials.As the sound level (dB) of each composite material is checked.The Polyeurathane rubber pad create the less noise than others.It suggest that it has the better damping ability rather than two.

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