Effectiveness and Predictability of Particle Damping on Cantilever Beam

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Abstract— Particle damping technique is a type of impact damping in which multiple auxiliary masses of small size are placed in a cavity that is fixed to the vibrating component. Particle damping can perform at elevated temperatures where most other forms of passive damping cannot. Because of its non-linear energy dissipation along with extreme simplicity, high effectiveness and economic application, it has a wide range of prospective for vibration and noise suppression in a broad range of applications. In past studies, particle damping system has been researched to a large extent, but many factors still need to be considered while implementing particles to damp a vibrating member, such as particle size, packing ratio, cavity size, cavity location, and so on. The objective of this paper is to provide design guidelines for utilizing particle damping and use particle damping to effectively suppress vibrations in simple beams by implementing series of experiments and co-relate the variables to generate imperial relations amongst various damping parameters. Thus, various aspects of particle damping will be studied to correlate the damping parameters with the vibration characteristics of a simple cantilever beam and come up with a generalized equation to provide optimum parameters of particle dampers and beams in various practical applications.

Keywords – Particle Damping, Linear Vibrations, Cantilever Beam, Damping Coefficient, Single DOF.

I. Introduction

This paper aims at studying the application of particle damping technique for noise reduction of a stationary machines used in industries. Particle damping is a unique way of damping with granular particles embedded within small holes in a vibrating component of a machine. The particles absorb kinetic energy through particle-to-wall and particle-to-particle frictional collisions. Due to the ease of application, greater efficiency and low cost, it has a wide range of prospective for vibration and noise suppression in a large range of applications. While particle damping techniques have been researched recently, their successful application is still missing in literature. Particle damping is a sub-type of impact damping where a large number of masses of very small size are placed inside a cavity attached to the vibrating member. Particle damping is effective at elevated temperatures where most other forms of passive damping cannot. Studies in recent years have shown the effectiveness and potential of this technique in practical industrial applications, and also that particle damping is largely a nonlinear damping method whose energy dissipation is based on a combination of loss mechanisms. The effectiveness of these mechanisms changes based on various system parameters. Due to the complex interactions of particle and the cavity walls, a comprehensive design methodology has not been developed which will enable this technology to be implemented without extensive trial-and-error testing. High amplitude resonance in vibrating structures of components in aerospace engine, aircraft propeller and rotating machinery, result in serious problems that can lead to failure of such structural members. For un-damped or very lightly damped structures, the response amplitude increases drastically at resonance frequency. If damping is made available, however, fundamental analysis and experimental data demonstrates that the response is inversely proportional to the damping coefficient (Figure 1). Hence if design stresses need to be calculated, the degree of uncertainties in the damping coefficient is closely equal to that in these stresses. Also, if the damping coefficients are significant, then the effect of these uncertainties in the analyses and the design are reduced.

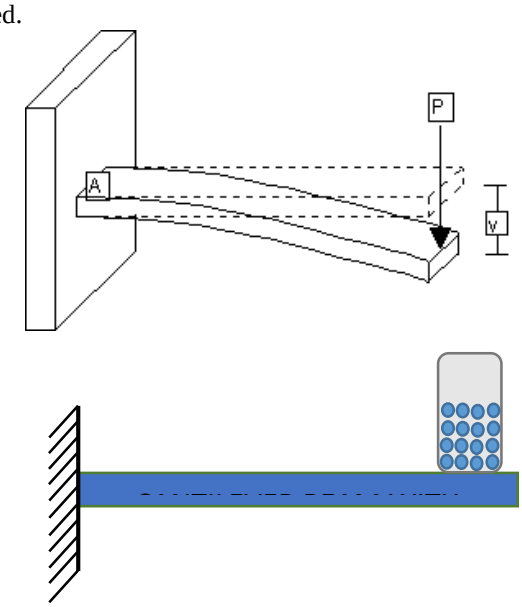


Fig. 1 Schematic of particle damping

II. LITERATURE REVIEW

The literature contains a multitude of papers that describe the application of impact or particle dampers to real structures. These include honeycomb structures, highand low-temperature applications, rotating structures, and others. Panossian sandwiched metallic particles between two copper sheets and reported acoustic attenuation of 5 dB at low frequencies and up to 27 dB at high frequencies. Two of the primary advantages of particle dampers are their insensitivity to temperature fluctuation and their ability to operate at extremely high and low temperatures. Panossian performed experiments with the LOX Inlet Tee on the Space Shuttle Main Engine, where high amplitude vibrations caused cracks to form. Four 1-mm diameter holes were drilled and partially filled with metallic balls of differing size and material. The performance or mass characteristics of the Inlet Tee were not changed, and a 25% increase in damping was recorded.

Particle damping can also be used to augment other vibration control methods. Tuned vibration absorbers are prime candidates for particle damping because the absorbers themselves have a large amount of kinetic energy at their tuned frequency. Collette et al. and Ma et al. applied impact damping to a tuned vibration absorber and attached the combined system to a two-degree-of-freedom system. Ma specifically attempted to reduce the large amount of space a tuned vibration absorber needs to operate. Impact and particle dampers have been used in a wide variety of other applications. This includes antennae, traffic signal structures, hang-gliders, and boring tools. There have been a few facts established about particle dampers. Higher mass ratios lead to more effective dampers. Also, the extreme sensitivity of impact dampers to operating conditions and parameters has been well documented, as well as the corresponding relative insensitivity of particle dampers.

Unfortunately, there is much that is still not known. The relation of the optimum gap size to other parameters such as excitation frequency and amplitude, particle size and shape, mass, etc. is still not understood. A lack of insight also exists about the behaviour of the particles under different conditions. This has greatly hampered attempts to model the damping and mass a PID would add to a system. Furthermore, much of the modelling in the literature was never verified experimentally. In spite of this lack of knowledge, however, impact and particle dampers have been successfully implemented for a wide variety of applications. Many of these applications are probably not optimal but the results have been successful nonetheless, showing that particle dampers can be effective even when applied imperfectly. This indicates that once a more complete understanding of the phenomena is gained, particle damping will be even more effective for a wide range of applications.

The goal is to develop greater insight into the behavior of the particle impact damper, identify and characterize key design variables that influence the effectiveness of the damper, develop a model that will allow prediction of damper effectiveness and implement techniques that will allow for the successful prediction of a structure's response when a particle impact damper is attached. Due to the complex nature of the damper, the majority of the work conducted is experimental. Experiments are designed to give a feel of how particle impact dampers behave and are used to help identify key design variables. Based on this information, later experiments are designed to characterize the effects of these key variables on the damper properties. A new method is developed that allows for the measurement of both the mass and damping properties of particle impact dampers without the supplementary use of a primary structure. This allows the characterization of design variables to be done quickly.

III. THEORY OF PARTICLE DAMPING

1. Particle Damping Theory

1.1. Theory – Viscoelastic Damping

According to this theory the particles damp due to both viscous and elastic property of the material. These particles collide with other similar particles and with those of the cavity wall. Total Energy is conserved only in case of perfectly elastic collisions where the colliding particles are non-deformed. For a purely elastic material, all the energy stored in the component is returned while loading as the load is removed. Also, the displacement of the component responds instantaneously and in-phase to the applied cyclic load. Conversely in case of pure viscous damping energy is completely dissipated after the load is removed. The input stress is wasted in pure damping as the kinetic energy is transferred totally to internal heat energy. These particles can be well represented with Maxwell model. Dashpot shown in the Maxwell model represents viscoelastic damping within the material.

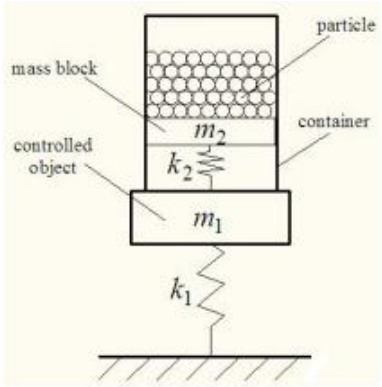


Fig. 2 Dynamic Model of Particle Damper System

1.2. Theory – Frictional Damping

The friction forces act on the colliding particles as they slide against each other and on the walls of the cavity. This process effectively converts the kinetic energy of the colliding particles to the thermal energy that is dissipated to the surrounding. Coulomb damping dissipates energy constantly because of sliding friction. The sliding friction coefficient is a constant that is independent of surface area or displacement or position or velocity of the particles. The system performing Coulomb damping has periodic nature that is oscillating and controlled by the force of sliding friction. Fundamentally, the member in the system is vibrating about the mean equilibrium point. A system being acted upon by Coulomb damping is nonlinear as friction force opposes the direction of motion.

And due to friction the amplitude of the motion decreases and decays with respect to time.

IV. DESIGN METHODOLOGY

1. Determine characteristics of the un-damped system.
2. Determine appropriateness of particle dampers.
3. Select preliminary particle damper configuration.
4. Determine characteristics of the undamped system with adjusted mass.
5. Evaluate damper effectiveness using the particle damper simulation technique.
6. General equation for Logarithmic decrement, $\delta = 1/n \ln (X_o / X_n) = 2\pi\epsilon / \{(1 - \epsilon 2)2\}$
7. FFT analyser is used to determine the frequency & also used to calculate amplitude with respect to time.
8. Accelerometer is used to determine the vibrations.
9. ANSYS or Matlab for software analysis.

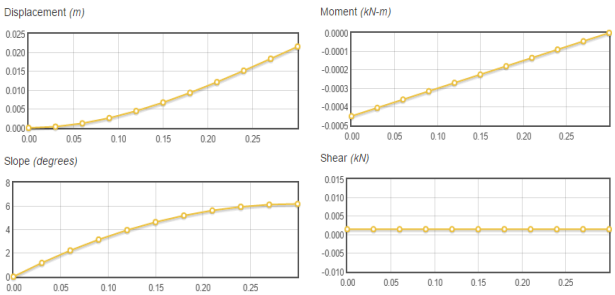


Fig. 3 Damping Parameters Vs Beam Free Length Plots

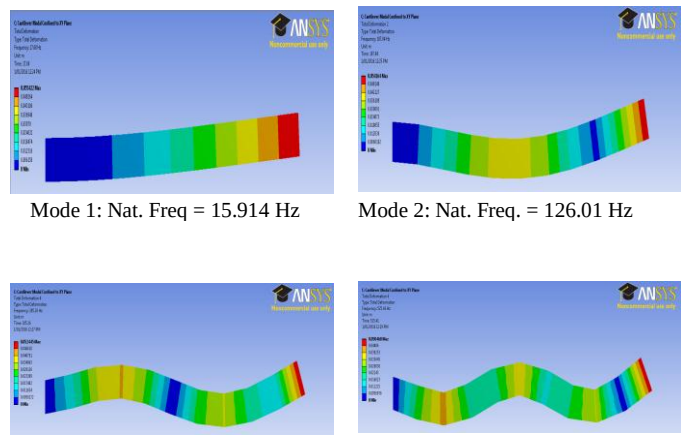
Parameter Selection

1. Cantilever Beam
   - Material — Young's Modulus and Bending Stiffness
   - Cross-section — As per amplitude of vibration
   - Length — As per Euler's buckling criteria
2. Enclosure
   - Diameter — According to width and length of beam
   - Height — As per max/min amplitude of vibration
   - Thickness — Minimum standard sheet thickness
3. Damping Particles
   - Material — High hardness and impact strength
   - Diameter — Vary within standard sizes of results
   - Quantity — As per enclosure dim for 20% packing ratio
4. Excitation Motor
   - Power — Vary as per load conditions
   - Speed — Increase progressively at min step intervals

V. FEA ANALYSIS

The thicknesses of the beams were realized using FEA to tune the natural frequency of the beam-cavity-satellite system to the desired 50-100Hz. In order to gain confidence in and establish accuracy of the FEA results, I first conducted several experiments varying grid density and mesh generator settings then compared the results to ideal hand calculations. As the test part, I modeled a cantilever beam that would be close to what I expected the final beam to be. Then I replaced the particle cavity tip mass assembly with a single point mass at the free end of the beam. The point mass assumption disregards the moment of inertia from the Cavity Cap and sensor board, so my hand calculation should produce a slightly higher natural frequency than the FEA result, but the angular displacement is so small. The mass and center of mass of the tip mass assembly (cavity, accelerometer, fasteners and mounting plate), was found using Solid Works "Advanced Assemblies" toolbox and used to calculate the equivalent beam length and point mass. The tip mass is 21.27g and is located an additional 50 mm from the free end of the beam. Extending the beam leaves 50 mm of beam mass unaccounted for, so its mass was subtracted from the tip mass. Resulting in an ideal beam length (L) of 350 mm with a beam mass (Mb) of 8.91g, and tip mass (Mt) of 21.1g.

Modal Analysis



Mode 1: Nat. Freq = 15.914 Hz — Mode 2: Nat. Freq. = 126.01 Hz

Mode 3: Nat Freq = 334.86 Hz — Mode 4: Nat Freq = 601.74 Hz

Fig. 4 Modal Analysis of Simple Cantilever Beam

Modes of Vibration	Undamped	Chrome damped (5mm)	Steel damped (5mm)	Brass damped (5mm)
Mode 1	15.914	13.316	12.87	15.31
Mode 2	126.01	113.67	110.55	120.68
Mode 3	334.86	313.91	309.94	326.2
Mode 4	601.74	554.83	567.2	590.26

Table 1. Modal Analysis of Various Damping Conditions

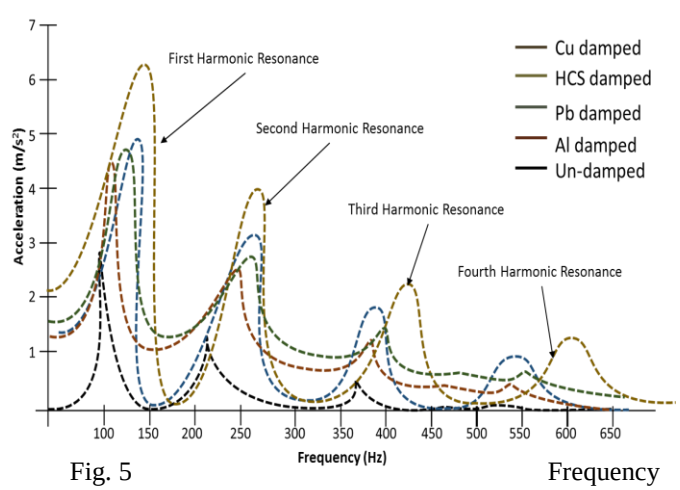


Fig. 5 Frequency

Response Curves for Various Damping Conditions

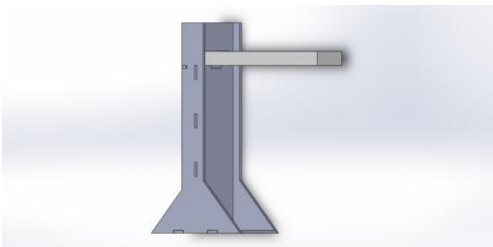
VI. EXPERIMENTAL SETUP

1. Beam material & Specifications

Dobefil 60 + Hardener EH 408 Room temperature curing, filled, off white coloured, epoxy resin system for potting.

Dobefil 60 is an off white coloured filled, solvent less epoxy compound. Dobefil 60 is a specially formulated and processed to obtain void free casting, potting encapsulated components. It contains specially selected and processed fillers incorporated under vacuum mixing & has lower abrasive nature. It is ready to use filled system having low viscosity at processing temperature. Dobefil 60+ Hardener EH 408 is a slower curing system with a low exotherm and flexible cured mass. Hardener EH 408 is a low viscosity polyamide liquid hardener.

The cured mass of Dobefil 60+ Hardener EH 408 shows good mechanical, electrical & chemical properties. Because of special type of fillers in Dobefil 60, the cured mass shows good thermal shock resistance, reduced shrinkage, low



coefficient of thermal expansion & increased thermal conductivity. The combination of Dobeckot 5504, Hardener EH 408 is used in thermal class 120 (class E) electrical components.

Fig. 6 Isometric View of Canitlever Beam Structure

Fig. 7 Experimental Setup with Exitation Motor

Fig. 8 Experiment performed using FFT Analyzer

Experimental Parameters:

1. Cantilever Beam:
   a) Material E = 250 Mpa Hardness = 100 BHN
   b) Cross-section Thk (t) = 10mm Width (w) = 30mm
   c) Length (L) 300mm

2. Enclosure:
   a) Diameter 50mm
   b) Height 50mm
   c) Thickness 5mm

3. Damping Particles:
   a) Material chrome,brass,stainless steel
   b) Diameter 2mm, 3mm, 4mm, 5mm
   c) Quantity Filled for 10%-30% packing ratio

4. Excitation Motor:
   a) Power 1/6 HP
   b) Speed 6000 rpm (to provide 0 – 100 Hz frequency)
5. Particle Variations:

6. Materials of different densities:
   Brass (density = 8.47g/cm^3, shear strength = 280 MPa)
   Chrome (density = 7.8/cm^3, shear strength = 80 GPa)
   Steel (density = 7.85g/cm^3,shear strength = 80 GPa)

VII. RESULTS

Following major observations were plotted during the experiment using FFT Analyzer in order to determine the optimum damping coefficient and the damper parameters for the cantilever beam under consideration:

1. 1. Particle Density vs Damping Ratio
2. 2. Particle Diameter (size) vs Damping Coefficient
3. 3. Packing Ratio vs Logarithmic Decrement
4. 4. Frequency Response Curve
5. 5. Packing Ratio vs Excitation Frequency
6. 6. Beam Stiffness vs Damping Coefficient

Based on the above plots corresponding regression graphs are developed and the below results were tabulated.

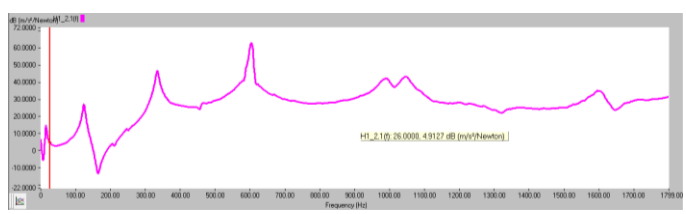


Fig. 9 FFT Analysis: Un-damped Vibrations

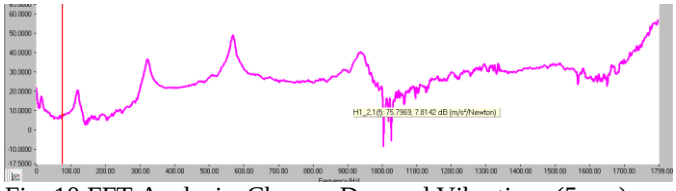


Fig. 10 FFT Analysis: Chrome Damped Vibrations (5mm)

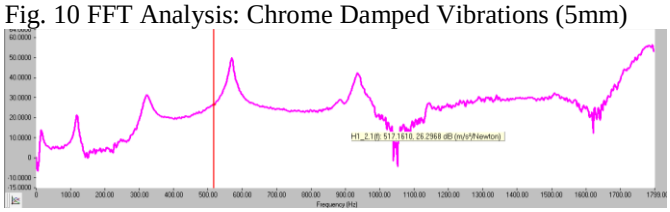


Fig. 11 FFT Analysis: Brass Damped Vibrations (5mm)

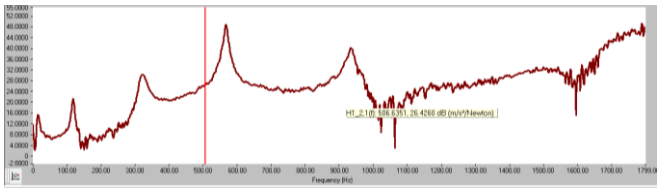


Fig. 12 FFT Analysis: Steel Damped Vibrations (5mm)

Mode No.	Frequency (Hz)	Amplitude (dB)	Damping Ratio (%)
1	15	13.24	12.13
2	124	26.71	2.42
3	337	44.54	1.07
4	604	52.73	0.79

Table 2. Modal Analysis using FFT for Undamped Vibrations

Comparison Chart:

Modes	Undamped		Chrome damped (5mm)		Steel damped (5mm)		Brass damped (5mm)	
	FEA	FFT	FEA	FFT	FEA	FFT	FEA	FFT
1	15.9	15.3	13.3	12.9	12.9	12.4	15.3	14.7
2	126.0	124.8	113.7	110.8	110.6	108.3	120.7	115.3
3	334.9	333.2	313.9	309.5	309.9	305.7	326.2	319.4
4	601.7	598.4	554.8	550.1	567.2	560.6	590.3	578.6

Table 3. Result Comparison Chart for FEA vs FFT Analysis

VIII. Conclusion

As a conclusion we can say that, after analyzing the system for various material and sizes two major conclusions were deduced. Firstly higher the density of the material, lower the damping effect due to inertial forces as predicted by Viscoelastic Maxwell Model. And secondly, higher the size of the particles higher the damping due to larger contact area as predicted by the Friction Based Particle Cavity Model. Thus the existing findings were confirmed and contemporary research done on particle damping has been verified to be in accordance with the current research.

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