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A Comparison of the Effectiveness of Elastomeric Tuned Mass Dampers and Particle Dampers

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ABSTRACT

This paper presents a study and comparison of two methods commonly used to treat unwanted vibration in vehicles. Laboratory work was done to measure and compare the effectiveness of common designs for practical tuned mass dampers (TMDs) and particle dampers under a wide range of conditions. The relative strength and weaknesses of the two approaches are compared in their abilities to treat vibration in a system due to resonant modes and forced response. The effectiveness of each method is investigated as a function of the weight of the treatment, amplitude and temperature effects.

INTRODUCTION

Auxiliary mass dampers are commonly used to reduce the vibration of vehicle systems. The two most common types of auxiliary mass dampers are the tuned mass damper (TMD) and the particle damper (also know as the shot damper).

This paper explores the relative performance of TMDs and particle dampers from a mass perspective. Added mass is an increasingly important issue as the manufacturers of vehicles strive for improved fuel economy.

The TMD consists of a mass mounted to the vehicle through an elastomeric spring. The TMD is tuned to have a natural frequency near the frequency where the vehicle system has the greatest response (usually a natural frequency of the vehicle system).

The elastomeric springs are usually SBR (synthetic butyl rubber) or EDPM. These materials are chosen for their low cost, relative insensitivity to temperature changes, durability, and aging characteristics. They generally have lower damping than is desirable for maximum TMD effectiveness.

Particle dampers are most commonly used to reduce the vibration of exhaust systems. The particle damper is a container filled with metal particles or spheres mounted to the vehicle through a simple spring such as a metal cantilever beam. Damping is produced by the relative motion of the fill material. Particle dampers are temperature insensitive, but their response is highly nonlinear making design more difficult.

SYSTEM DESCRIPTION

To compare the performance of TMDs and particle dampers, the simple system shown in Figure 1 is used. The vehicle system is represented as a mass/spring/damper system and the auxiliary mass damper is represented as a mass/spring system. While this system may appear overly simple to represent realistic systems in automobiles (such as the engine and transmission), complex systems can be decomposed into individual modal responses, so this analysis is generally applicable to more complex systems.

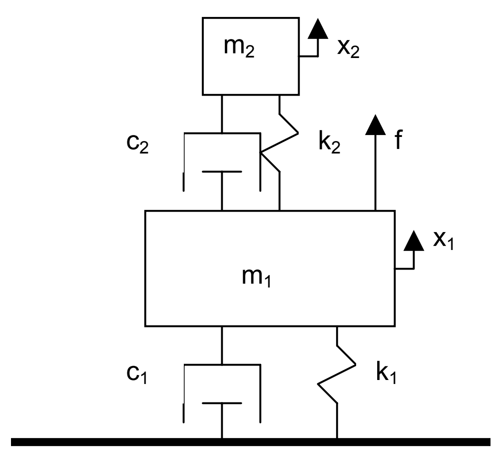


Figure 1 – Model of Base Structure with Damper.

A vehicle system with parameters commonly found in practice (for example controlling powertrain bending) is used, but the formulations are applicable to systems of any size. The vehicle system has a mass of 100 kg, a natural frequency of 100 Hz, and a damping ratio of 5%.

TUNED MASS DAMPER

The spring and damping functions of the TMD are both performed by the elastomer. Consistent with the literature on elastomers, a complex representation of the stiffness is used per the equation below.

$$\hat{k}_2 = k_2(1 + j\eta) \quad (1)$$

Where k_2 is the stiffness, η is the loss factor (equal to twice the damping ratio), and j is the square root of minus one.

The equations of motion for the system are:

$$f = m_1\ddot{x}_1 + c_1\dot{x}_1 + (k_1 + \hat{k}_2)x_1 - \hat{k}_2 x_2 \quad (2)$$

$$0 = m_2\ddot{x}_2 + \hat{k}_2 x_2 - \hat{k}_2 x_1 \quad (3)$$

At a single frequency, ω, Equations (2) and (3) become:

$$F = \left[-\omega^2 m_1 + j\omega c_1 + (k_1 + \hat{k}_2)\right]X_1 - \hat{k}_2 X_2 \quad (4)$$

$$0 = \left(-\omega^2 m_2 + \hat{k}_2\right)X_2 - \hat{k}_2 X_1 \quad (5)$$

Where ω is the frequency in radians per second.

PARTICLE DAMPER

The mass and damping functions of the particle damper are both performed by the fill material and its container; therefore, the mass will be represented as a complex quantity per the equation below:

$$\hat{m}_2 = m_{2r} + m_{2i} j \quad (6)$$

Where m_{2r} is the real part of the mass, and m_{2i} is the imaginary part of the mass (representing damping). For this study, the complex mass as a function of frequency and vibration amplitude was measured for motion in the vertical direction by performing swept-sine shaker measurement. The characteristics of the particle damper are nonlinear. At low vibration amplitudes, there is little relative motion of the particles, so the real mass is equal to the static mass (container plus particles) and damping is small. As vibration amplitude is increased, relative motion of the particles increases. The real mass decreases and damping increases. As the vibration amplitude increases further, the particles become very active. The real mass approaches that of the empty container, and the damping decreases.

The imaginary part of the mass, m_{2i}, can also be related to the more commonly known damping coefficient by the equation:

$$c_2 = -\omega m_{2i} \quad (7)$$

At a single frequency, the motion of the dynamic system is still described by Equations (4) and (5) with $\hat{m}_2$ substituted for m_2. To solve the equations, it is necessary to assume a vibration amplitude X_2 to estimate the correct values for m_{2r} and m_{2i}, solve the equations to determine a better estimate of X_2, and repeat the process at each frequency until the solution converges. Experimentally determined values for m_{2r} and m_{2i} are used. The analysis assumes that the damper is controlling a structure whose vibration is dominated by a single frequency component.

COMPARISON OF EQUAL MASS TMD AND PARTICLE DAMPER PERFORMANCE

For the purpose of comparison, the performance of one TMD and two particle dampers are compared at different excitation levels of the base vehicle structure. One particle damper is filled with lead spheres, and the other is filled with steel powder. All the dampers have the same mass of 3 kg. It is assumed that the weight of the mounting brackets and springs for the three dampers would be comparable.

One particle damper is filled with lead spheres with a diameter of 1.5 mm. The container is a cylinder with an inner diameter of 36 mm oriented horizontally. The mass of the container is 0.744 kg, and the mass of the fill material is 2.256 kg. This design is optimized using guidelines for particle dampers design given in [1]. Lead is chosen because this material is most commonly used for exhaust system dampers.

The other particle damper is filled with iron powder (Fe-116 –30/+80 mesh). The same container shape is used for both particle dampers (but longer for the iron powder due to its lower density). The mass of the container is 1.376 kg, and the mass of the fill material is 1.624 kg. Iron powder is used because previous experiments showed improved performance over lead spheres.

Note that different materials could be used particle dampers. The most important design variable is the material density, but secondary design variables include particle size, shape, and material. Special care would be required for the iron powder used here to prevent corrosion.

The measured values of m_{2r} and m_{2i} for the particle dampers as a function of excitation level are shown in Figure 2. Note that the iron powder is, in general, superior to the lead spheres because the real part of the mass and the imaginary part of the mass are higher over a wider amplitude range.

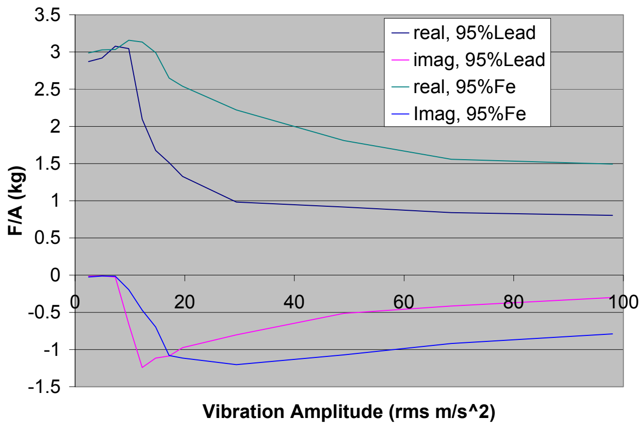


Figure 2 – Measured Real and Imaginary Masses for the Particle Dampers.

Figure 3 shows the effectiveness of two TMD designs in controlling the vibration of the base structure excited by a sinusoidal force of 10 N,rms. The damping value of η = 0.25 is optimal, and the damping value of η = 0.1 is representative of materials typically used in automotive damper construction. The damper spring is optimized to produce the lowest peak acceleration of the base structure. The uncontrolled base structure has a peak acceleration of 1 m/s^2,rms. The peak values with the η = 0.1 and η = 0.254 dampers are 56.9% and 47.5% of the uncontrolled peak acceleration, respectively. Because the TMD is assumed to be a linear system, the percentage improvements are the same at all force levels.

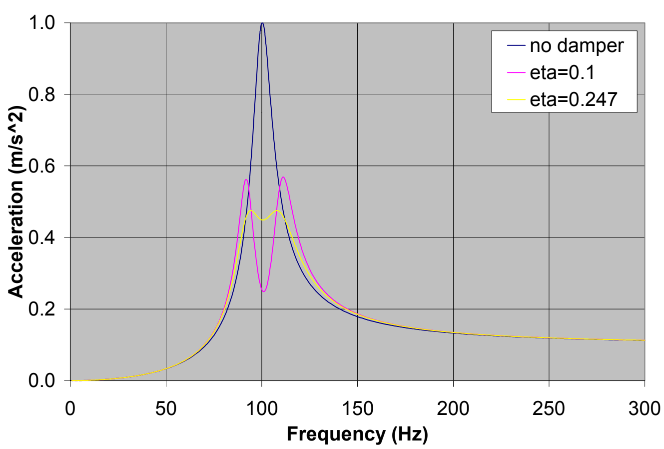


Figure 3 – Effectiveness of TMDs.

Figure 4 shows the effectiveness of the particle damper with lead fill material in controlling the vibration of the base structure with an excitation force of 10 N,rms. The ratio of controlled amplitude to uncontrolled amplitude is 90.8%. Further examination of the damper performance shows that the excitation level of the damper is too low for optimal performance. Figures 5, 6, and 7 show the effectiveness of the same particle damper with excitation forces of 20, 30, and 40N applied to the base structure. The resulting performance ratios are 67.2%, 60.6%, and 69.8%, respectively. The effectiveness of the damper is a strong function of the vibration level of the base structure. For this structure, the uncontrolled peak acceleration for the 30 N force is 3 m/s^2 which would be a value that an automotive powerplant might experience, but this value is higher than would be found anywhere on the body.

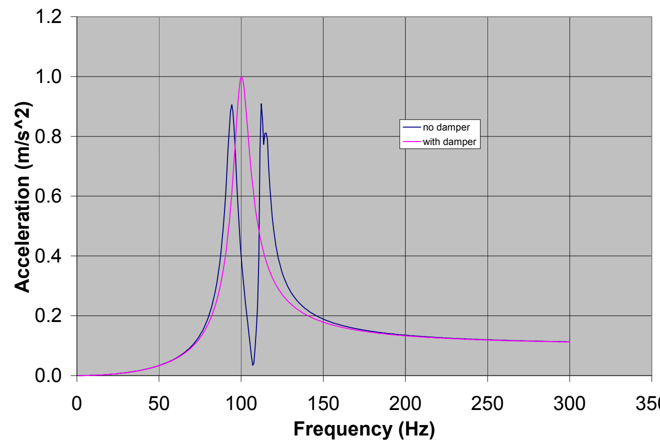


Figure 4 – Effectiveness of 95% Lead Filled Particle Damper with Structural Excitation of 10 N.

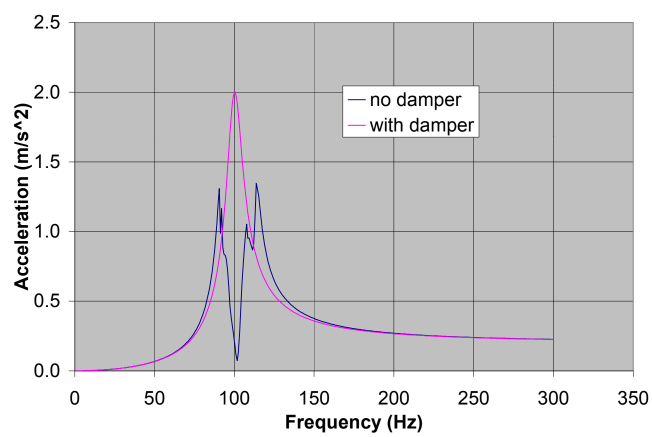


Figure 5 – Effectiveness of 95% Lead Filled Particle Damper with Structural Excitation of 20 N.

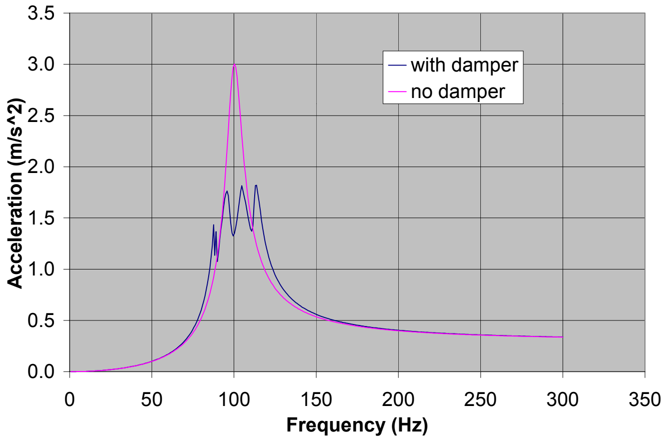


Figure 6 – Effectiveness of 95% Lead Filled Particle Damper with Structural Excitation of 30 N.

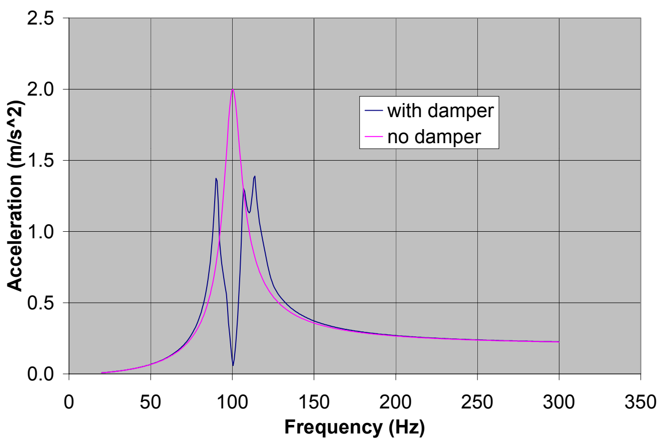
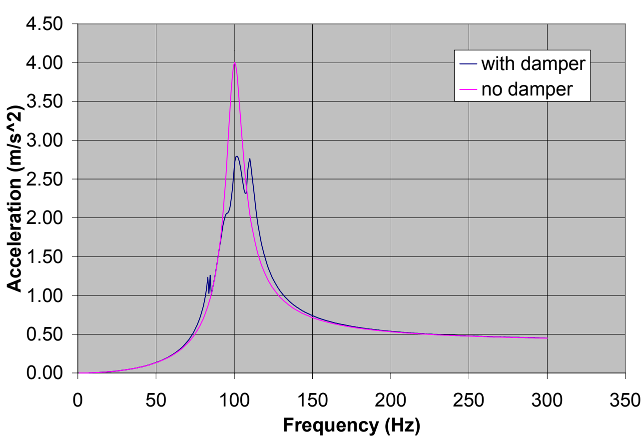


Figure 7 – Effectiveness of 95% Lead Filled Particle Damper with Structural Excitation of 40 N.

Figures 8, 9, 10, and 11 show the effectiveness of the damper with iron powder fill material with excitation forces of 20, 30, 40, and 60 N applied to the base structure. The performance ratios are 69.6%, 56.7%, 52.2%, and 59.5%, respectively. The iron powder fill material performs better than the lead fill material. Also,



the performance does not degrade as much at higher excitation levels. The best performance (52.2%) is better than the TMD with η = 0.1 (56.9%), but worse than the TMD with η = 0.25 (47.5%)

Figure 8 – Effectiveness of 95% Iron Powder Filled Particle Damper with Structural Excitation of 20N.

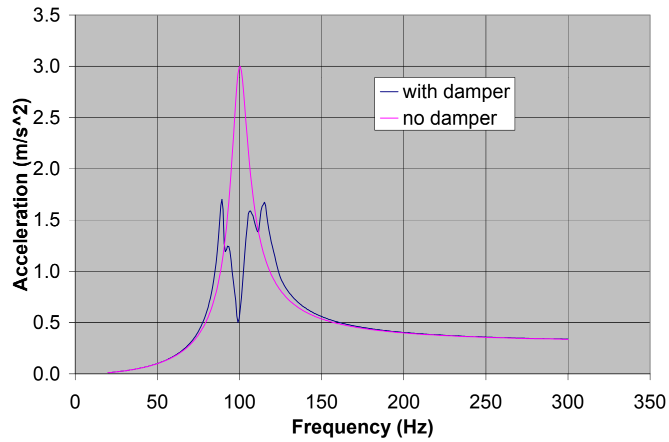


Figure 9 – Effectiveness of 95% Iron Powder Filled Particle Damper with Structural Excitation of 30N.

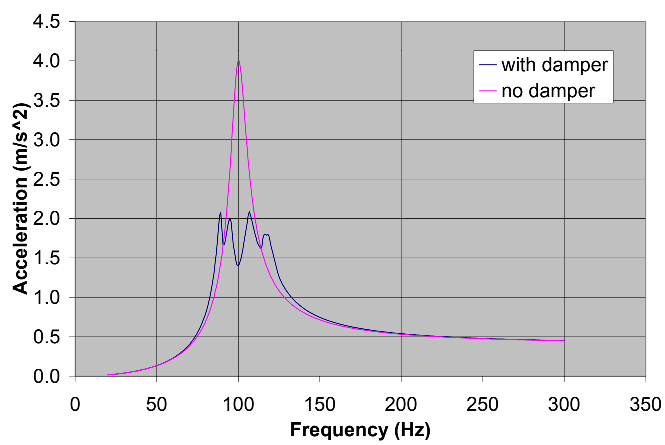


Figure 10 – Effectiveness of 95% Iron Powder Filled Particle Damper with Structural Excitation of 40N.

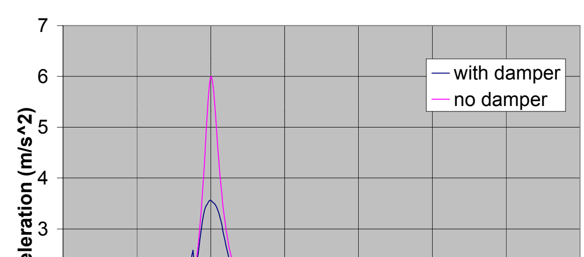


Figure 11 – Effectiveness of 95% Iron Powder Filled Particle Damper with Structural Excitation of 60N.

The effectiveness of the damper on a modified base structure (damping increased to 10%) is shown in Figure 12. The force amplitude was 70 N. The performance ratios are 74.7%, 66.1%, and 72.7% for the TMD with η = 0.1, the TMD with η = 0.26 (optimum), and the iron powder filled particle damper. The performance of the particle damper was comparable to the TMD with typical damping, but poorer than the performance of the TMD with optimum damping. Note that to improve performance, the particle damper is modified by increasing the container mass to 2.190 kg and decreasing the fill mass to 0.810 kg. This produces a very slight improvement in performance, but the required excitation level is increased.

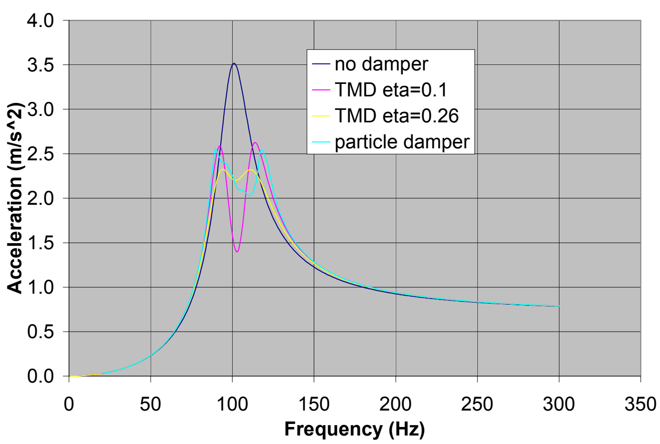


Figure 12 – Effectiveness of TMD and Iron Filled Particle Damper on a Modified Base Structure (10% Damping).

TMDs become less effective when the operating temperature is not the same as the design temperature. For example, when the temperature of EDPM rubber is reduced from room temperature to –18C the stiffness can increase by a factor of 2.5. Figure 13 shows the effect of this stiffness increase of the optimized TMD (η = 0.25) applied to the original base structure (5% damping). The performance goes from 47.5% to 91.6%. Particle dampers are not temperature sensitive like TMDs.

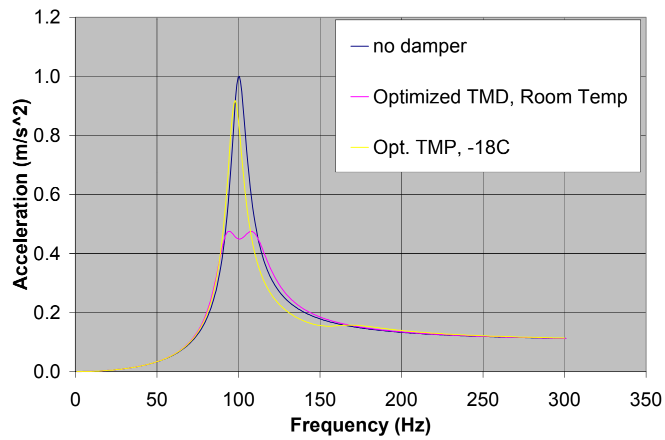


Figure 13 – Effect of Temperature on the Optimized TMD.

CONCLUSION

Tuned mass dampers and particle dampers of equal mass where found to have similar performance potential.

The relative performance of two particle dampers and one tuned mass damper of equal mass were compared for vibration control of a base structure. The best performance was achieved with the tuned mass damper with optimized damping (controlled/uncontrolled vibration of 47.5%). The same TMD with commercially practical damping values had a performance of 56.9%.

The best performance of a particle damper filled with lead spheres was 60.6%, and the best performance of a particle damper filled with iron powder was 52.2%. This is slightly better than the performance of a practical TMD of the same mass.

Particle damping is effective over a range of excitation, but poor control is seen when the excitation is too high or too low.

The performance of particle dampers is comparable to typical tuned mass dampers of equal mass, but poorer than optimized TMDs. The advantages of a particle damper are that it is temperature insensitive, and not as likely to degrade in performance with time.

Because the particle dampers are nonlinear, it was necessary to perform studies with specific structures at specific forcing amplitudes; therefore, the results and the conclusion will vary under different conditions.

REFERENCES

1. G. Tomlinson, “Recent Developments in Particle Dampers”, Sixth National Turbine Engine High Cycle Fatigue Conference, March 5-8, 2001, Jacksonville, Florida, USA, University of Sheffield, UK.

CONTACT

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