Design Issues in the Use of Elastomers in Automotive Tuned Mass Dampers

Allan Aubert & Art Howle
Roush Industries, Inc.

ABSTRACT

The concept of using tuned mass dampers and absorbers to address undesirable vibration responses in vehicles is not new. However, there are several design issues that cause the vibration control performance of real life tuned dampers to be significantly less than that predicted by simple 2-DOF theory.

In this paper, the authors will review tuned damper design theory. Practical issues regarding the use of real life elastomers as spring elements are reviewed, including temperature sensitivity, material damping and nonlinearity. Various elastomers are compared for their effectiveness and applicability to the typical automotive environment. Rules of thumb for tuned damper design are discussed including locations for placement of dampers in automotive structures, tuning for temperature variations, determining the needed mass, and measurements and simulations that can greatly improve the success and timing for tuned damper design.

INTRODUCTION

Practical vehicle designers are constantly faced with the fact that vehicles live in an environment full of sources of vibratory excitation. Common sources of cyclic excitation include engine firing, powertrain inertial forces, rotating imbalances in the driveline and wheels, as well as road inputs. Vehicle structures respond in many different ways due to their natural resonant behavior. Despite the best forethought of designers, unwanted vibration responses can emerge at any stage of a vehicle design. Tuned mass dampers (TMD) and tuned vibration absorbers (TVA) are well known methods to address these issues. Such dampers and absorbers are seldom planned to be included in early design stages. Yet tuned dampers may be the best NVH design alternative and should not be categorized as an undesirable "band-aid" fix.

Table 1 lists some of the many applications of dampers in automobiles.

Table 1: Common tuned damper applications:

- Crankshaft (torsional & linear for bending)
- Axle pinion nose (linear)
- Halfshaft & driveshaft dampers (torsional & linear)
- Transmission extension or transfer case (linear)
- Subframe (linear)
- Frame or body (linear for shake)
- Gear lever or steering wheel (linear)
- Exhaust system (linear)
- Etc., etc., etc.!

A tuned damper adds an additional degree of freedom (M_2, K_2, C_2) to a resonant system as shown in the schematic of Figure 1. The effect on the response of the original system is to split the original resonance peak into two new resonance peaks, which we call side bands. In this diagram, K represents the stiffness element, and C is a viscous damping element. In Figure 2, the blue line shows the frequency response of the original system showing a resonance peak at 120Hz, and the red line shows the response after the TMD is added.

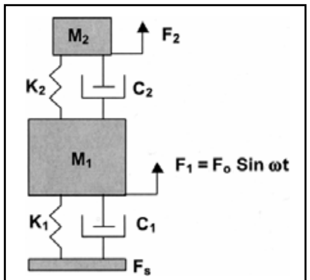

Figure 1: Schematic of dynamic system with TMD

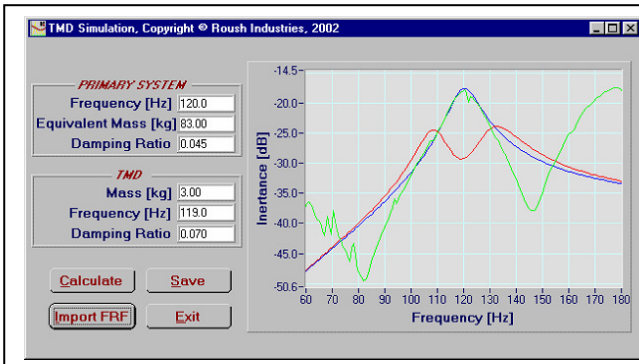

Figure 2: Frequency Response of System Mass (M_1) Before and After Application of Tuned Damper

The spring element on most automotive tuned mass dampers is usually provided by some type of elastomer. The elastomer provides both the spring and damper element indicated in Figure 1. By changing the material type, stiffness and damping properties can be adjusted. Spring stiffness is related to the hardness, or "durometer" of the elastomer element. Figure 3 shows how the stiffness (Young's Modulus) of a typical natural rubber compound shifts with the durometer. Table 2 shows typical cost, temperature and damping properties of several elastomers commonly used for automotive tuned dampers. There are many formulations and blends of elastomers with unique characteristics that can give very good performance in a tuned damper application.

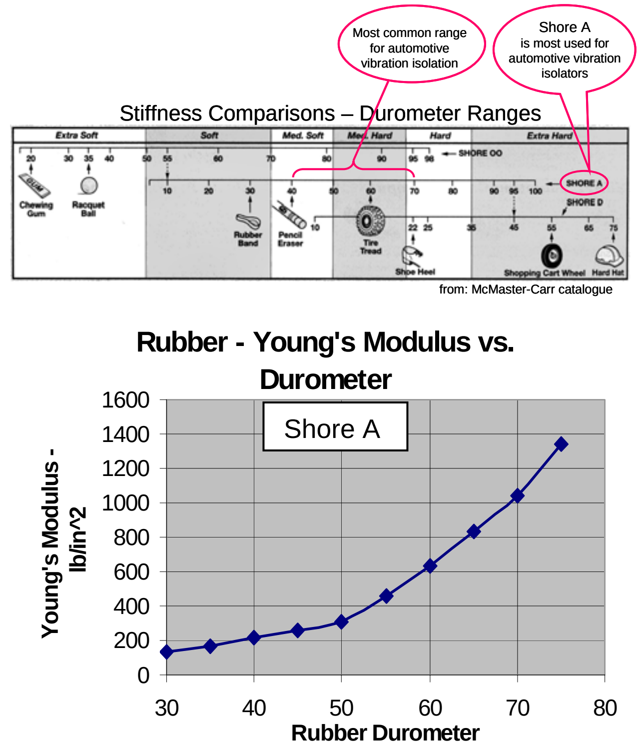

Figure 3: Durometer Definitions

Material	Relative Damping	Temperature Capability Degrees F	Approximate Cost Ratio* (2006)
Natural Rubber (NR)	Reference	-65 to 210	Reference
Isoprene	-25%	-65 to 210	1.0
SBR	+25%	-60 to 215	1.0
Chloro Butyl	200-300%	-50 to 250	0.9
EPDM	+20%	-50 to 290	1.4 - 2
Silicone	-50%	-150 to 400	6 - 8
Silicone - Hi Temp	-50%	? to 600	10 - 12

** Natural Rubber (NR) = $1.80 to $2.25 per Lb. (2006)*

**These Costs Highly Dependent on desired characteristics*

Table 2: Properties of Some Elastomers

PRACTICAL ISSUES OF TUNED DAMPERS

Modulus and damping properties of elastomers vary with the amplitude and frequency of excitation, as well as the operating temperature. As such, the damper environment must be carefully considered during design and development. Figure 4 shows an example of the shift in the response frequency of a tuned damper at various magnitudes of excitation. Figure 5 illustrates how the modulus and damping of an elastomer vary with temperature. Both of these effects must be carefully considered during the design and development phases as they can render a TMD ineffective as an NVH treatment.

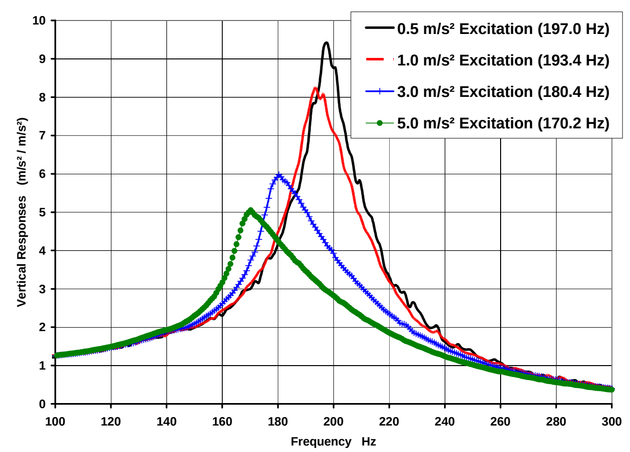

Figure 4: Nonlinear Amplitude response

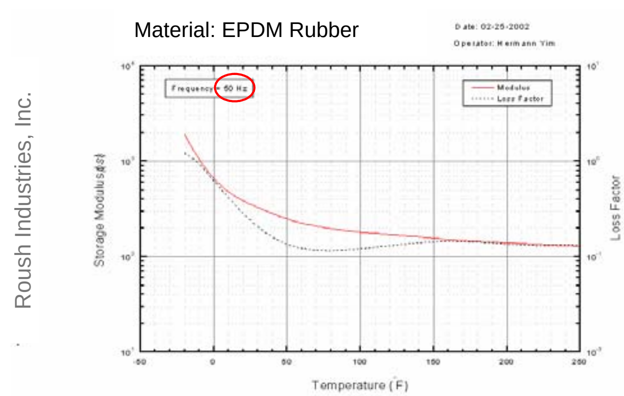

Figure 5: Modulus and loss factor are temperature dependent for most elastomers

Frequency dependence, preload and nonlinear geometric effects of the elastomer spring element must all be taken into account, as they affect the spring rate seen in practice.

The first step in the development of a tuned damper is to identify the system resonance to be treated, and to evaluate the practicality of using a tuned damper to control it. A vibration response is not necessarily due to a resonance. It may, in fact, be an area with high excitation. For example, an engine block will have comparatively high vibration responses. However, most of the response is due to forces of combustion and inertial forces of rotating components moving the mass of the rigid powertrain. A resonance is indicated when the vibration level peaks at a specific frequency that does not change as the excitation frequency is varied. A high response due to high excitation is not likely to be successfully treated with tuned dampers, though very low damping tuned absorbers may be designed to deal with such issues in a very narrow frequency range. Tuned absorbers can be very effective with constant speed devices; however, such devices are outside the scope of this paper.

Mass properties of the resonance, location, and local stiffness are all design concerns for developing tuned dampers that will be covered in the next section.

A PROCEDURE FOR DEVELOPING EFFECTIVE TUNED DAMPERS

Once a resonant response is identified, and the decision is made to treat it with a tuned mass damper, the properties of that resonance must be determined. Key to this is evaluation of the mode shape of the resonance. It is recommended that an experimental or analytical modal analysis, or past experience be used to locate an anti-node, or location of maximum response. This location is most effective in the placement of tuned dampers for reduction of the undesired response. Note that local stiffness of the structure must also be evaluated in order to ensure that the damper is effectively coupled to the system. Stiffness at the attach points should be typically 10X the stiffness of the spring element in the tuned damper in order to achieve best damper performance.

1. First, the frequency response at the chosen location must be measured. This is done using the driving point FRF procedure, resulting in a plot as illustrated in Figure 6. It is a good idea to take similar measurements at all potential locations for placement of the tuned damper.

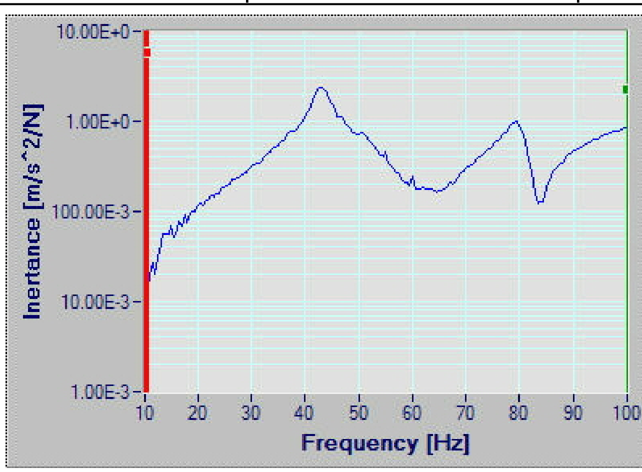

Figure 6: Typical Measured Frequency Response, from a Vehicle Steering System

2. Next, the resonance needing treatment must be identified. In the above example, an undesirable response was observed in the vehicle at 43Hz, indicating the issue to be with the resonance peak seen at that frequency.

3. The peak in the measured data should then be matched against the frequency response of an "equivalent" single degree of freedom (SDOF) system. Such curve fitting routines are commonly available in modal test software. The "Curve Fit" box on the left side of the screen display of Figure 7 shows the resulting SDOF resonance peak frequency, equivalent mass and damping ratio.

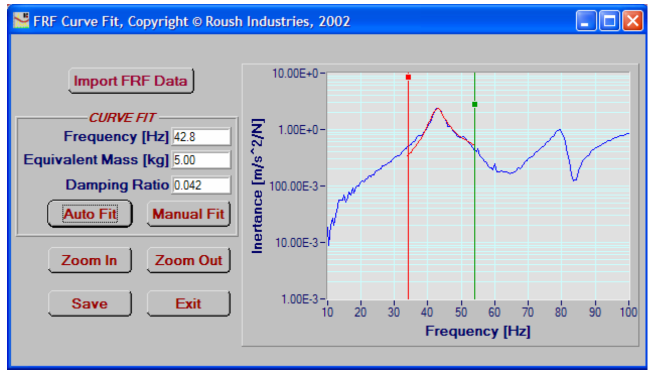

Figure 7: Curve fit Developed from SDOF Parameters that Closely Matches the Measured Resonance at 43Hz

The term "equivalent mass" is used to describe the mass of a single degree of freedom system that, together with the appropriate stiffness and damping, result in a predicted frequency response curve that closely matches the measured frequency response curve. These characteristics are particular to the TMD attachment point measured (preferably an anti-node as mentioned above). As the TMD attachment point is located further from an anti-node, the equivalent mass will increase.

Equations 1a and 1b show an alternate way to determine the equivalent mass at a given point. This method only requires determination of the peak frequency before (f_{n1}) and after (f_{n2}) attachment of a known mass (m_{add}) at the proposed attachment point. The resulting two equations in two unknowns allow calculation of k, the stiffness term, and m_1 = the equivalent system mass.

$$f_{n_1} = \frac{1}{2\pi}\sqrt{\frac{k}{m_1}} \qquad\qquad f_{n_2} = \frac{1}{2\pi}\sqrt{\frac{k}{m_1 + m_{add}}}$$

Equations 1a and 1b: Calculation of Equivalent Mass

4. The mass for the TMD must be chosen. For most automotive applications, a recommended starting point for design of the tuned damper mass is ~1/20th of the equivalent mass at the damper location. In the case being illustrated, we started with M_{TMD} = .25 kg.

The curves of Figure 8 show the effects of choices of TMD mass. Since, in this example, the equivalent mass is 5kg, the .25kg damper mass matches the recommended 1/20th of equivalent mass value. Note that in this figure, as well as in later figures, the dB value of improvement for each tuning option is shown in the legend. One can see that for higher TMD mass the response changes only slightly at the side bands, but the two resulting peaks are split further apart.

This gives a wider effective tuning range. There is far more improvement right at the tuning frequency for higher TMD mass. The choice of TMD mass is therefore dependent on how much improvement in response is needed, and how much insensitivity to mistuning is required. The positive effect of higher mass must be traded off against the negative effects of requiring more package space, and adding cost and weight to the vehicle.

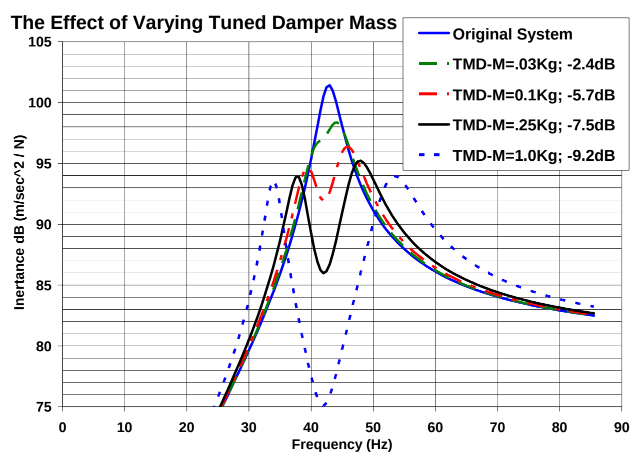

Figure 8: The effect of Varying TMD Mass (in kg) on a System with 5kg Equivalent Mass

Figure 9 illustrates the effects of damping on TMD performance. Damping helps to attenuate the side band responses, but effectiveness at the tuned frequency is traded off. For a constant speed machine, you may want to use less damping. In practice, the damping is usually limited by the elastomer which is selected from a limited set of materials with appropriate characteristics for a tuned damper. The "optimum" damping ratio of a TMD is a function of the ratio of the TMD mass to the equivalent system mass[1].

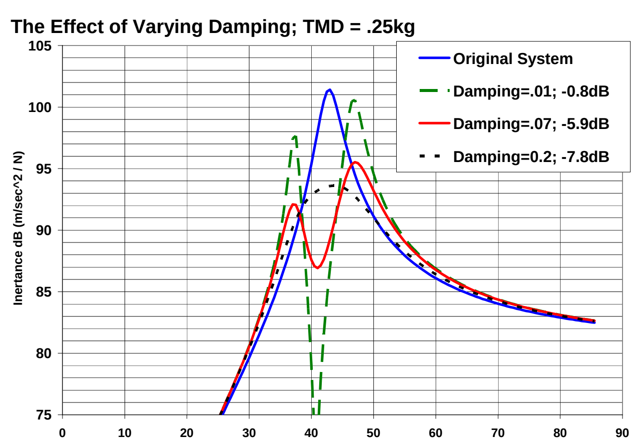

Figure 9: The effect of Varying Damping on TMD Performance

Damping performance often has to be "traded off" against other considerations such as durability, temperature stability, and manufacturing concerns. In addition, the ability to achieve the desired tuning frequency is also degraded by manufacturing variation and the range of temperature encountered by the damper in service.

5. A practical elastomer must be chosen. Figure 10 shows the curve fit resulting from a practical elastomer, a 40 durometer EPDM. This choice must take into account the needed temperature range and tolerance for degradation of performance at the temperature extremes. Consult Table 2 for some options.

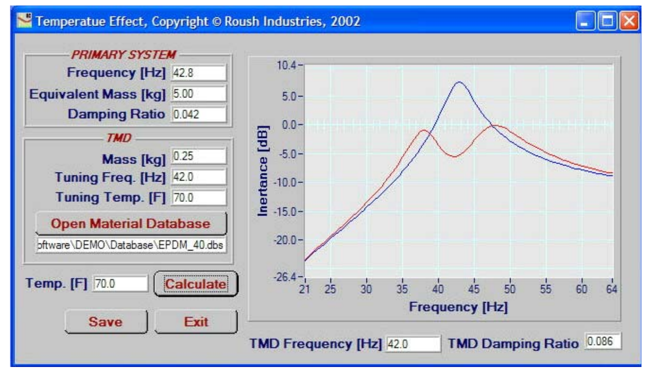

Figure 10: Response of EPDM with 40 Durometer – Giving Close to Optimum Performance

Figure 11 shows the effect of operating the TMD in a range of temperatures commonly encountered by such devices in automotive service. Note that at 30 degrees F the elastomer stiffens and the tuning is shifted higher in frequency, resulting in a reduction to 4dB of improvement. At 120 degrees F, the elastomer softens, and the performance is degraded from the ideal 7.5dB to 5.3dB of improvement. A TMD of higher mass will have a wider tuning band and less degradation.

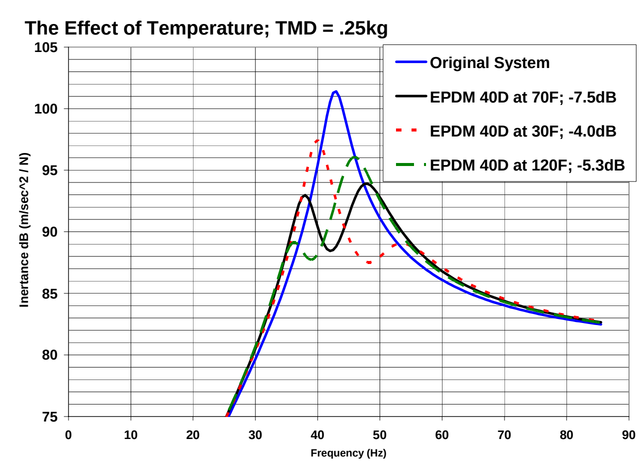

Figure 11: The effect of Varying the Temperature of EPDM in a TMD Tuned at 70F

6. A further design issue in practice is manufacturing variation. How closely can the bench tuning be maintained over hundreds or thousands of dampers? A great deal of effort is required to maintain a tight tuning tolerance. Most variance comes from slight changes in

[1] "Vibration Damping"; Jones, Nashif, Henderson

elastomer composition, age of the uncured batch, and variations in curing time and temperatures during manufacturing. Figure 12 illustrates a spread of about ± 12% for a TMD mass of .25kg (1/20th system mass).

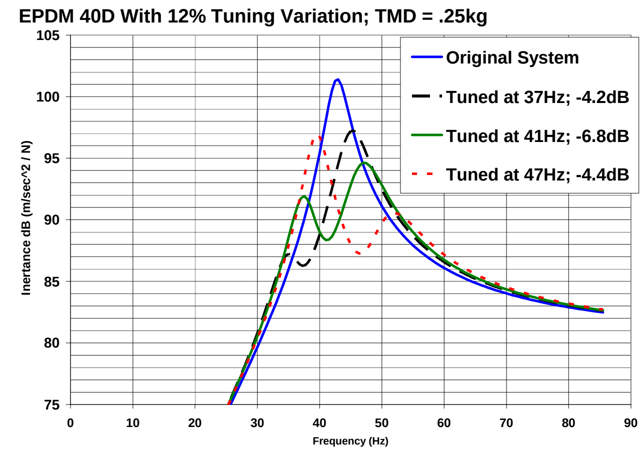

Figure 12: M_{TMD}=.25kg System Response Variation with TMD Tuning Frequency

Figure 13 shows this effect when a TMD mass of .1kg (1/50th of system mass) is selected. Note the greater sensitivity to tuning and less improvement with lower TMD mass selection. Also note that this variation can be from manufacturing, operating temperature, excitation amplitude, and all combinations of these.

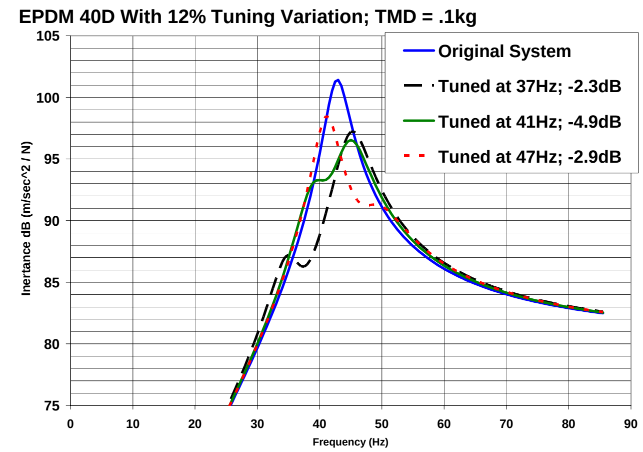

Figure 13: M_{TMD}=.1kg System Response Variation with TMD Tuning Frequency

A robust design will result when mass, elastomer type, operating temperature range, and manufacturing variation have all been addressed.

7. Finally, the TMD is built and tuned on the bench, or on a shaker, and the resonant frequency and damping value verified. After installation on the vehicle, responses should be measured on both the system and the TMD mass during operation. Tuning adjustments are sometimes required in order to optimize damper effectiveness. Caution: It is easy to prove that a damper will not work. The designer must verify that any resulting lack of performance is not due to issues such as a change in the target system between measurement and installation of the tuned mass damper. Proper function must be verified, and not assumed.

CONCLUSION

The above seven step process has been used by the authors in many commercial applications to successfully design and tune tuned mass dampers. The following considerations affecting tuned damper performance have been studied in this paper:

- Tuning frequency and production variation
- Damper mass
- Damping of the elastomer
- Operating temperature

Understanding of the above allows tradeoffs for NVH benefit, weight, and cost to be understood prior to design, hardware fabrication, and vehicle evaluations.

While the use of a TMD is considered by many to be a less than ideal design solution, in many cases it represents the optimum design decision based on cost, weight, function and timing. Many vehicles make extensive use of TMD technology to achieve high levels of NVH refinement.

REFERENCES

1. Jones, David, "Viscoelastic Vibration Damping", John Wiley & Sons, Ltd., ISBN 0 471 49248 5, West Sussex, England, 2001.

2. Nashif, A., Jones, D., Henderson, J., "Vibration Damping", John Wiley & Sons, Ltd., ISBN 0 471 86772 1, New York, New York, USA, 1985.

3. Harris, C.M., Crede, C.E., "Shock and Vibration Handbook", Second Edition, McGraw-Hill Book Company, ISBN 0 07 026799 5, New York, New York, 1976.