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Measurement of Dynamic Parameters of Automotive Exhaust Hangers

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ABSTRACT

Different methodologies to test and analyze the dynamic stiffness (K) and damping (C) properties of several silicone and EPDM rubber automotive exhaust hangers were investigated in this research. One test method utilized a standard MTS hydraulic test machine with a single sine excitation at discrete frequencies and amplitude levels, while a second method utilized an electrodynamic shaker with broadband excitation. Analysis techniques for extracting the equivalent stiffness and damping were developed in the shaker tests using data from time domain, frequency domain, as well as force transmissibility. A comparison of all of the shaker testing methods for repeatability and accuracy was done with the goal of determining the appropriate method that generates the most consistent results over the range of testing. The shaker testing in the frequency domain using a frequency response function model produced good results and the set-up is relatively inexpensive. The hydraulic excitation method, however, is more suitable for large displacements and is ideal to study the variations of K and C with frequency, displacement, temperature and pre-load.

INTRODUCTION

Automotive exhaust hangers are small oval shaped elastomeric elements used to suspend the exhaust system including the muffler to the body of a vehicle. The dynamic forces transmitted to the vehicle body is influenced by the stiffness and damping of the exhaust isolators. Current methods for experimentally determining the dynamic properties of rubber isolators using a hydraulic actuator system are expensive and therefore cost prohibitive for small companies. The goal of this research program is to develop an inexpensive test procedure and analysis technique for estimating the dynamic properties of elastomers. Additionally, we want to compare these results with standard hydraulic excitation test data obtained from MTS 831 elastomer test machine. Also, most applications for viscoelastic isolators involve broadband inputs as opposed to a single sinusoidal input used in the hydraulic actuator system. Hence a methodology that enables measurement of dynamic properties of the elastomer over a wide frequency and amplitude range is useful for computer modeling and simulations including noise path analysis (NPA). To this end, we have attempted to use an electrodynamic shaker to excite the sample with a broadband excitation, and extract the equivalent linear stiffness and damping based on a single-degree-of-freedom model. The results from this study are compared with sine-sweep test data from an MTS test system. The advantages and limitations associated with each method are pointed out.

A review of recent literature indicates that very little work has been done in the area of dynamic characterization of exhaust isolators per se, although a wealth of research papers can be found on general elastomers, mounts and shock absorbers. The reason for this is unclear. Perhaps, the perception may be that the exhaust hangers are relatively small rubber elements that can be modeled as simple springs. Furthermore, since most of these are supposed to be located close to the nodal locations of exhaust system modes, it may be assumed that they do not pose many problems in body NVH. This assumption is not quite valid. In some situations, exhaust hangers can be significant structure-borne NVH paths to the vehicle passenger compartment. There are conflicting requirements placed in the selection of stiffness and damping of exhaust hangers based on durability and NVH. Hence, specific knowledge of the stiffness and damping properties of these bushings is important to accurately estimate the dynamic forces transmitted to the vehicle body.

Fifty different exhaust hangers (10 types of 5 each) of various compounds of silicone and EPDM rubber, some with metal reinforcement bands, were studied in this research. The experimental set-up and analysis techniques using an electrodynamic shaker are first described followed by the same for the hydraulic actuator system. The latter investigation also includes results of the effects of frequency, sample-to-sample variation, input displacement, and temperature on the dynamic properties of the isolator.

SHAKER TESTING METHODOLOGY

The test set-up consisted of an electrodynamic shaker connected through a piano wire to the mounts (See Figures 1 (a) and (b)). A mass of about 10 Kg was applied at the free end to keep the piano wire in tension under the dynamic load and to simulate a pre-load experienced by the exhaust hangers in actual application. The exhaust hanger was held in place by aluminum brackets. The aluminum brackets and the elastomer are assumed to behave as a SDOF lumped mass, spring, damper system in the frequency range studied. The flexible modes of the aluminum bracket were outside the frequency range of interest and hence did not contribute in the analysis procedure.

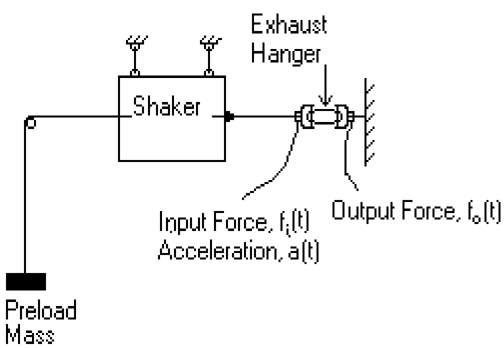


Figure 1(a): A Schematic of the Shaker Set-up.

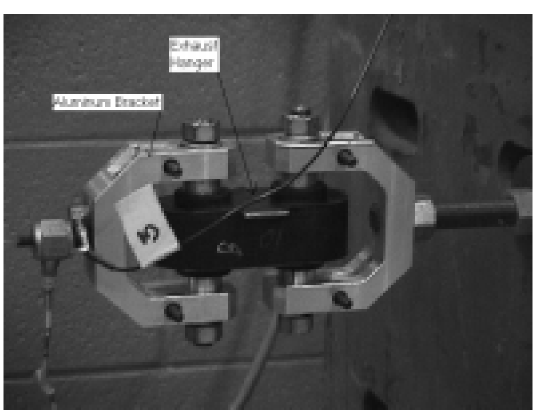


Figure 1(b): A Picture of the Exhaust Hanger.

The input force, $f_i(t)$, input acceleration, a(t), and the output force, $f_o(t)$, were measured using piezoelectric transducers. The initial studies were performed by measuring the two forces and acceleration up to 8000 Hz. Analysis of these spectra revealed that the first peak in the frequency response function (FRF) corresponds to the natural frequency of the hanger stiffness and the bracket-hanger mass. Therefore, further research was focused only on the first peak in the FRF, which, for all samples, was between 50-150 Hz. Consequently, for the later tests, the system was excited using band limited burst-random noise with a bandwidth of 0-250 Hz.

FREQUENCY DOMAIN ANALYSIS - The measured frequency spectrum for the input force, $F(\omega)$, input displacement, $X(\omega)$, and the output force, $F_o(\omega)$, were used for the two SDOF frequency domain analysis methods. The first frequency domain approach used the frequency response function (FRF) to estimate the parameters. The dynamic mass (m), stiffness (K) and damping coefficient (C) were extracted from the measured FRF by fitting a least-squares curve-fit for the equation:

$$\frac{|X(\omega)|}{|F(\omega)|} = \frac{1}{\sqrt{(K - m\omega^2)^2 + (C\omega)^2}} \qquad \textbf{(Eq. 1)}$$

The second frequency domain approach used the force transmissibility between the output and input force to estimate the parameters. The dynamic mass, stiffness and damping were then extracted from the force transmissibility equation:

$$\frac{|F_{out}|}{|F_{in}|} = \sqrt{\frac{K^2 + (C\omega)^2}{(K - m\omega^2)^2 + (C\omega)^2}} \qquad \textbf{(Eq. 2)}$$

In these models, we can also define $\zeta = C/C_c$, the viscous damping ratio, where $C_c = 2m\omega_n$, the critical damping coefficient. The viscous damping ratio ζ, is half the loss factor, η. The derivations of Equations (1) and (2) can be found is any standard vibrations textbook.

TIME DOMAIN ANALYSIS - The time domain method requires time traces for velocity, v(t), and displacement, x(t), in addition to input force, f(t), and acceleration, a(t). The integration of accelerometer data to obtain velocity and displacement information is difficult since time domain integration often becomes unstable unless the acceleration data is closely controlled for drift. For this research, velocity and displacement information was obtained by passing the acceleration data through a high pass filter with a 5 Hz cutoff. This removed any drift in the data and allowed for stable integrations.

An alternative approach is to use a feedback controller to make sure that the sample is excited properly with a constant input during the test. Once appropriate velocity and displacement traces were obtained, dynamic mass, stiffness and damping were extracted by fitting a least-squares curve-fit for:

$$m\ a(t)+C\ v(t)+K\ x(t) = f(t). \qquad \textbf{(Eq. 3)}$$

This time domain model fits the data over the entire frequency range recorded to yield equivalent stiffness and damping valid for that test case. The two frequency domain models fit the parameters in a limited frequency range due to the governing equations. For a single-degree-of-freedom system, the stiffness dominates the response at low frequencies, the damping influences the response near the natural frequency, and the mass usually controls the high frequency region. Therefore, the estimated parameters are treated as constants in the frequency region from which they are extracted.

HYDRAULIC ACTUATOR TESTING METHODOLOGY

This technique is essentially a non-resonant forced vibration method ideal for tests involving low frequencies and large displacements where inertia effects can be ignored. Here the sample is excited at a single frequency, the input displacement and output force are measured. In theory, the dynamic parameters are extracted using Equation (3) without the mass term from time or frequency domain data.

In this research, the hydraulic actuator tests were performed using a MTS 831.50 elastomer test machine. Figure 2 shows pictures of this experimental setup. The elastomer test machine can generally be used for input displacement levels as large as 10 mm and frequencies as low as 1 Hz. Based on the previous shaker characterization work and the limitations of the hydraulic actuator capability, the exhaust hangers were tested using a sinusoidal input ranging from 1 to 200 Hz at various displacement amplitudes and temperatures.

ANALYSIS – The data analysis software in the MTS system has two methods to extract the dynamic parameters that are essentially same. One method is based on DIN Specification 53 513. The DIN analysis method derives force and displacement amplitudes and phase (Φ) from the parameters of a hysteresis loop, formed when force and displacement time data are plotted against each other, as shown in Figure 3. The program estimates the dynamic stiffness K^* as the ratio of the amplitudes of peak-to-peak force (f) and displacement (d).

To determine phase or damping, the program first determines the area of the hysteresis loop by a trapezoidal rule integration. Using the calculated area of the hysteresis loop, the phase is calculated using the equation $\sin(\Phi) = (4a/\pi fd)$, where a = area of hysteresis loop. The loss factor, η, is then calculated according to $\eta = \tan(\Phi)$. The values of K and C are given by:

$$K = K^* \cos(\Phi) \text{ , and } \quad C = \frac{K^*}{\omega} \sin(\Phi) \qquad \textbf{(Eq. 4)}$$

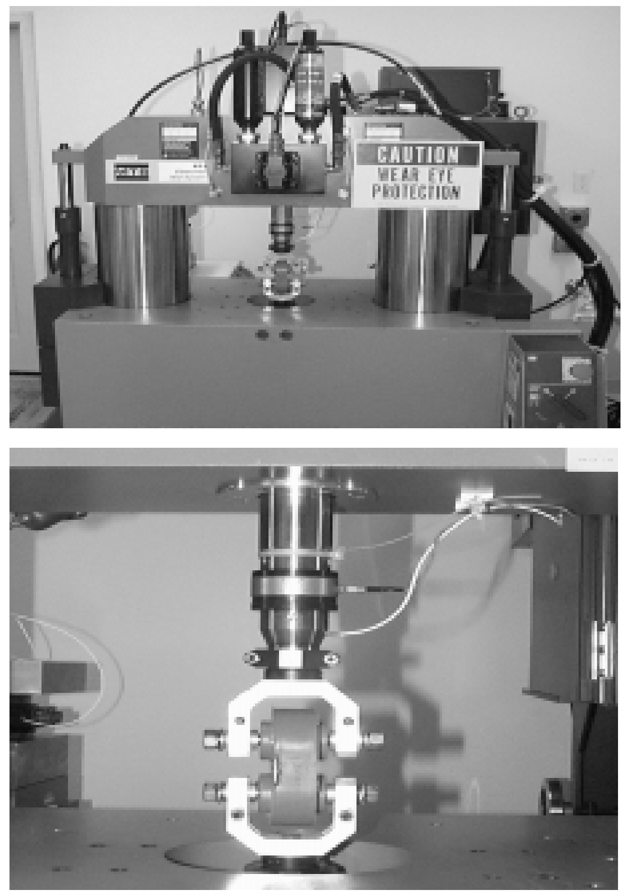


Figure 2: Hydraulic Actuator Test Set-up.

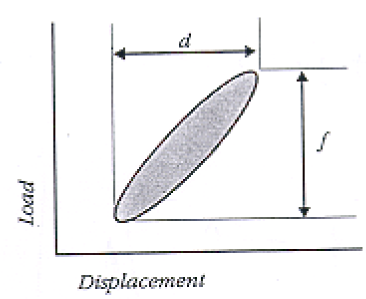


Figure 3: Hysteresis Loop

The second analysis methodology uses the standard MTS sine sweep test and is visualized in Figure 4. Shown in the figure are the output force vector (A_{load}) and input displacement vector (A_{disp}) plotted along with their phases in a real and imaginary format.

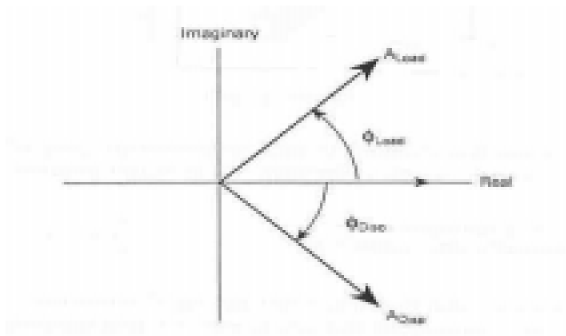


Figure 4: Force and Displacement Vectors.

From this representation, the stiffness (K) and damping (C) of the elastomer can be calculated using equations, $K^* = \frac{A_{load}}{A_{disp}}$, $\Phi = \Phi_{load} - \Phi_{disp}$

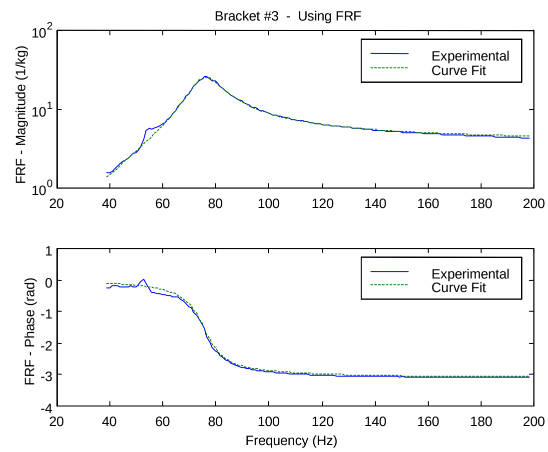
$K = K^* \cos(\Phi)$, and $C = \frac{K^*}{\omega} \sin(\Phi)$.

It should be noted that this analysis is limited to a single frequency excitation. This analysis method is the current industry standard in the elastomer characterization. This method, however, can be extended to all frequencies by realizing that K* and the phase are simply the magnitude and angle of the frequency response function between the output force and input displacement. Note that this assumes that the inertia effects can be ignored in the frequency range of interest.

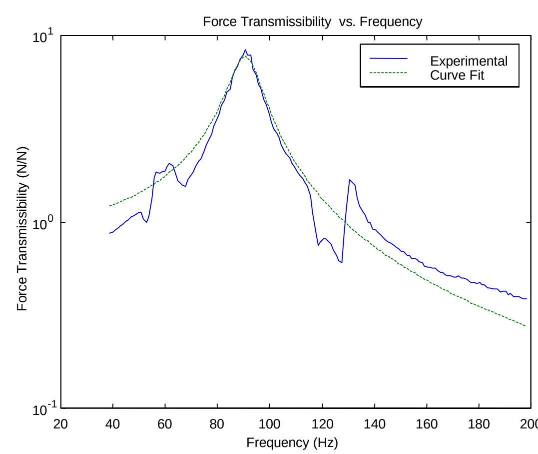
Before discussing the results, it is appropriate to note some fundamental differences in the nomenclature used in the literature--that are often confusing--to define stiffness and damping of rubber isolators. Terms like, static, quasi-static, dynamic, equivalent linear, effective, etc. are used to represent stiffness while damping ratio, loss factor, loss angle, tan-delta, quality factor, etc. are all used for damping. This is because of the various mathematical models used in the analysis. It should be noted that the stiffness value obtained from the SDOF system shaker set-up is considered constant & does not vary with frequency. This may be considered as effective or equivalent linear stiffness. The stiffness, K^*, obtained from a non-resonant single frequency test is called the dynamic stiffness and is related to K and C by $K^* = \sqrt{K^2 + (C\omega)^2}$. This value obviously is frequency-dependent. The static stiffness is obtained from a pure static load-deflection data and it can be linear or non-linear with the displacement amplitude. The MTS 831 machine utilizes a quasi-static test procedure to estimate the static (or quasi-static) stiffness.

SHAKER TEST RESULTS AND DISCUSSION

Figure 5 shows typical FRF and force transmissibility plots along with curve-fits for an exhaust hanger with metal insert. Figure 6 shows time domain raw data and results from the curve-fit. A summary of all shaker test results for one particular sample (type 3) is given in Table 1.



(a)



(b)

Figure 5: Frequency Domain Curve Fits, a) Using Frequency Response Function, b) Using Force Transmissibility.

As evident from the results of Table 1, the parameters extracted from the time domain correlate well with the parameters extracted from the frequency domain. The results from other specimens of hangers (not presented here) yielded similar results. In all of these tests, the r.m.s values of input force and hanger displacements were about 8 N and 0.1 mm, respectively.

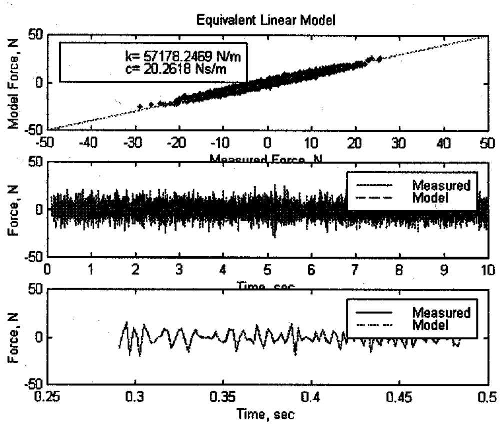


Figure 6: Time Domain Curve-fits.

Table 1: Summary of Shaker Test Results.

Sample	Parameter	Time Domain	Frequency	Force
			Response Function	Transmissibility
5152	M (Kg)	0.253	0.257	0.251
Compound	C (Ns/mm)	20.262	20.485	18.754
SKT 1044 w/	K (N/m)	57178	58402	57095
Insert	ωd (Hz)	75.416	75.577	75.697
	Loss Factor (%)	16.851	15.294	17.128

HYDRAULIC ACTUATOR TEST RESULTS

STATIC TEST - Figure 7 shows a typical static load vs. deflection curve obtained from the MTS machine during static tests. The static stiffness of the sample is linear in the entire displacement range tested and is about 40,000 N/m. However, this was only true for some samples with metal bands. For some other samples with metal bands, dual stiffness rates were observed. This bi-linear stiffness feature --one for small displacements, the other higher stiffness under large displacements--may be advantageous from a design point of view. Most of all samples without the metal insert exhibited non-linear static load-deflection patterns indicating predominantly cubic non-linearity. The estimation of a single static stiffness value under such conditions becomes very difficult since the stiffness changes with displacement and rate of loading. For most part, however, the static stiffness can be assumed to be linear in the narrow range of displacements typically encountered in mid-to-high frequency NVH.

SAMPLE-TO-VARIATIONS - Figures 7 shows the variation of dynamic stiffness (K^*) and damping (loss factor) of six hangers from the same type tested under identical conditions. The standard deviation (σ) for the stiffness of the six specimens was about 6%. Almost all the variation comes from specimen 3C, which has a stiffness of about 20,000 N/m less than the other five specimens over the entire frequency range. Close examination of the specimen revealed that the drop in stiffness was due to a crack in the metal reinforcement band. The standard deviation without the cracked specimen was less than 1%. The mean stiffness over the entire frequency range was 84,000 N/m, where as stiffness at 1 Hz was about 60,000 N/m.

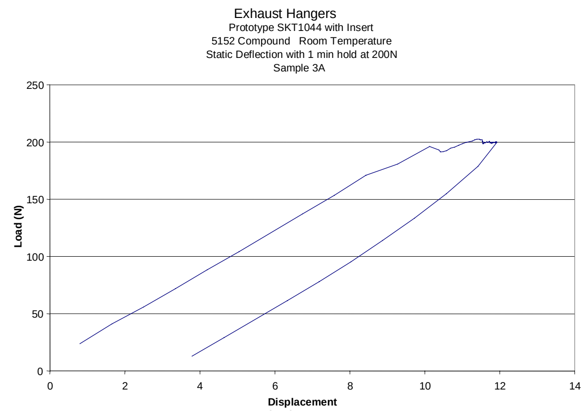
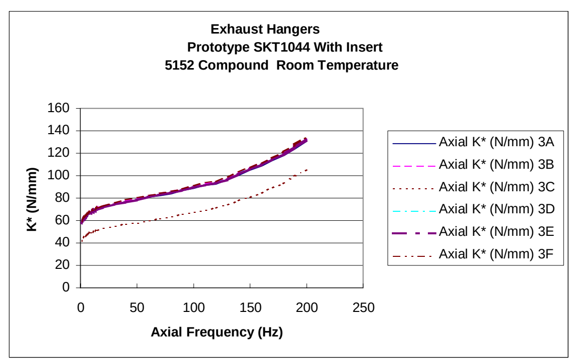
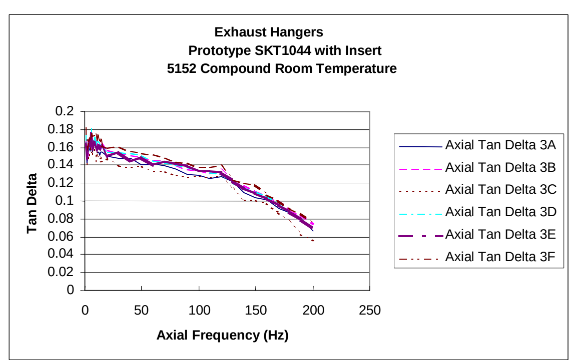


Figure 7: Sample Static Test Result.



a)



b)

Figure 8: MTS Test Data Showing Sample-to-Sample Variations- a) Dynamic Stiffness, b) Loss Factor.

The loss factor is fairly constant over the frequency range, with a slight decrease as frequency increases. The mean loss factor was about 13.6% with a standard deviation of σ = 2.67%.

EFFECTS OF METAL INSERT – Figures 9 (a) and (b) show the variation of dynamic stiffness and damping of two exhaust hangers made of the same rubber compound with and without a metal insert. The stiffness of type 10A with the metal insert is more than double the stiffness of type 9A without the insert. Also, some samples with metal insert showed bi-linear stiffness behavior. There is also a small increase in the loss factor due to the addition of the metal band as shown in Figure 9(b).

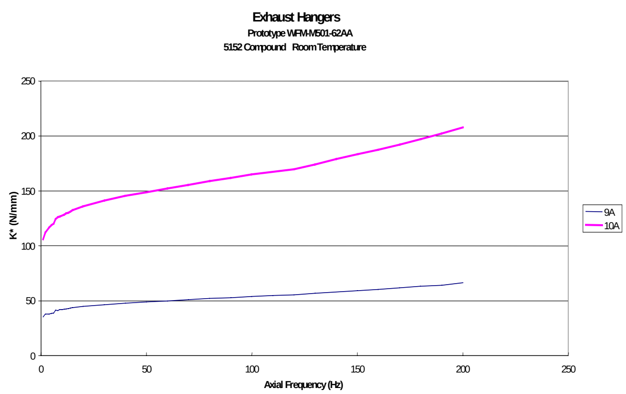


Figure 9(a): Stiffness vs. Frequency for Samples With and Without Metal Insert.

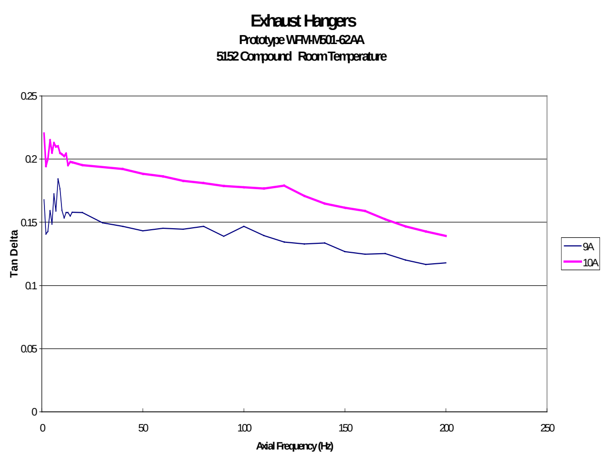


Figure 9(b): Loss Factor vs. Frequency for Samples With and Without Metal Insert.

Sample type 3 and type 8, shown in Figure 10, are identical except for the material. Type 3A is made of EPDM rubber whereas type 8A is made of silicone. The stiffness values of these hangers were very close, but the loss factor of all EPDM samples were almost twice those of the silicone samples.

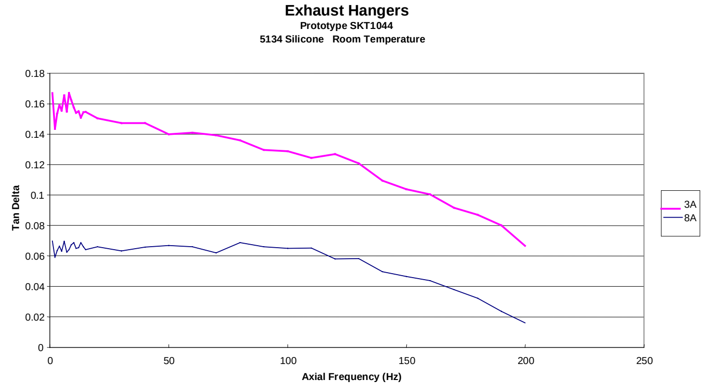


Figure 10: Loss Factor of EPDM vs. Silicone Samples.

EFFECTS OF TEMPERATURE – Figures 11 (a) and (b) show the variation of K^* and η for exhaust hangers tested by varying the temperature from –40 to 200 °F. Frequency sweeps up to 200 Hz were also conducted at each temperature. These results show that stiffness increases with frequency, but decreases with temperature. A large jump in the stiffness is seen near -20°F, which corresponds to the glass transition temperature (T_g) for the rubber material. The loss factor shows a significant increase near $T_{g,}$ which is consistent with typical viscoelastic behavior. The estimated damping is maximum for temperature well below room temperature. Exhaust hangers are typically exposed to temperatures well above the room temperature where the damping is minimum. This is actually good news since reduced damping translates to lower values of force transmitted to the vehicle body near the operating frequencies and temperatures.

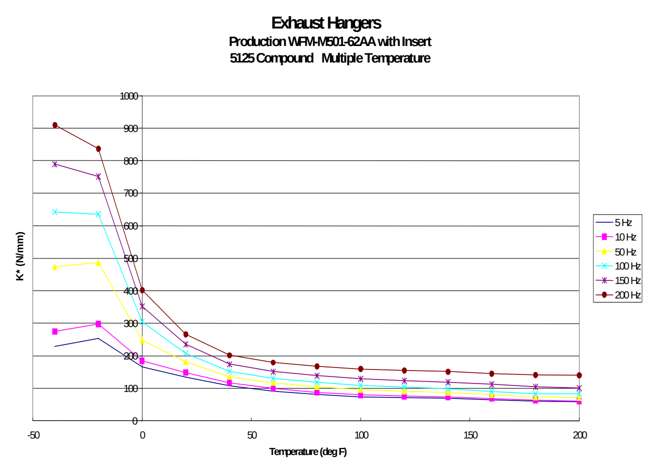


Figure 11 (a): Variation of Dynamic Stiffness with Temperature.

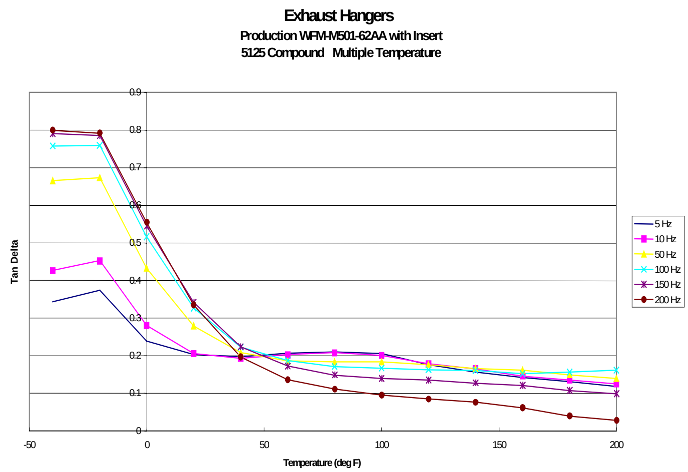


Figure 11 (b): Variation of Loss Factor with Temperature.

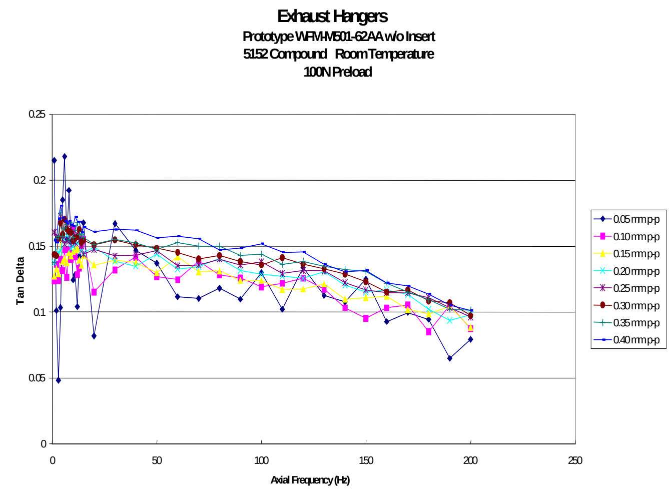


Figure 12 (b): Variation of Loss Factor with Displacement.

EFFECTS OF DISPLACEMENT – Figures 12 (a) and (b) show the variation of K^* and η with eight different peak-to-peak displacement levels. The dynamic stiffness shows a slight decrease in value with increase in displacement. The loss factor shows a slight increase with an increase in displacement.

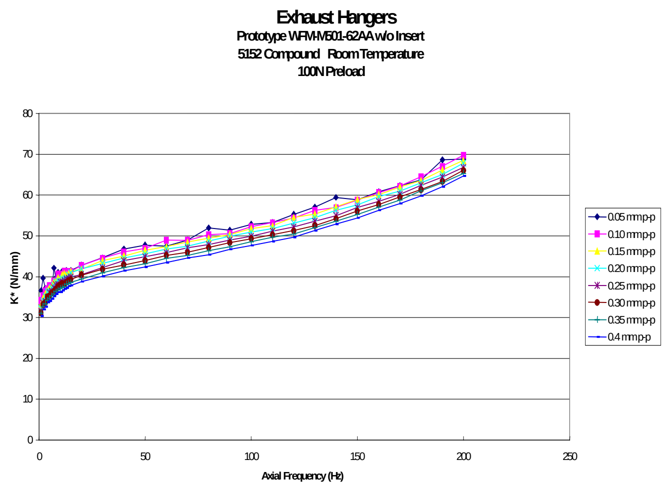


Figure 12 (a): Variation of Dynamic Stiffness with Displacement.

COMPARISON WITH SHAKER TEST DATA - When comparing the MTS results with shaker test data, it is important to realize the differences in the mathematical models used in the two approaches for parameter estimation. There is an increase of dynamic stiffness with frequency as seen in Figures 8 (a) and 9 (a). This is typical for all MTS test data. The reason for this is that the dynamic stiffness has an inherent frequency dependence given by, $K^* = \sqrt{K^2 + (C\omega)^2}$. So, for lightly damped elastomers, K and K^* will be approximately same even though the frequency term appears as a square. For cases where damping is relatively significant, K will be close to K^* for low frequencies. For all samples tested in this research, K and K^* were equal for frequencies equal to or less than 1 Hz. Another reason to keep in mind is the fact that inertia effects at high frequencies are largely ignored in the non-resonant forced vibration methods as in MTS analysis. Finally, the damping itself will exhibit a frequency dependency in many elastomers as noted in Figures 8 (b), 9 (b) and 10. Now onto the comparison.

For the type 3 sample with metal insert, the MTS data indicates a low frequency dynamic stiffness of 60,000 N/m, compared to the 57,000-58,000 N/m obtained from the three analysis methods used in the shaker tests. Similarly, the extracted loss factor from the frequency domain shaker tests closely match the MTS loss factor result near the natural frequency. The MTS data indicates a loss factor at 75 Hz of 14-15%, which is close to the 15-17% range obtained from the all of the shaker tests. For most samples tested in this study, the linear stiffness estimated from shaker tests closely matched the low frequency dynamic stiffness from the MTS testing. This is a significant result. In NPA, and most body and chassis NVH computer models, the exhaust isolators are usually represented as a linear-springdashpot model. For such models, the equivalent linear K and C values can quickly be obtained from a simple shaker test that is relatively inexpensive compared with the hydraulic set-up.

CONCLUSIONS

The dynamic parameters of various exhaust isolators were measured using two different excitation methods. The shaker set-up with analysis using time or frequency domain data produces equivalent stiffness and damping results that are useful for linear analysis. The shaker testing is less time-consuming and is relatively inexpensive compared with a hydraulic set-up. The hydraulic excitation method, however, is more suitable for large displacements and is ideal to study the variations of dynamic stiffness and damping with frequency, displacement, temperature and large pre-loads. The actual choice of the testing method depends on the requirements and intended use of the dynamic properties of the elastomer.

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