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Measurement of Dynamic Properties of Automotive Shock Absorbers for NVH

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ABSTRACT

This paper describes a project on the dynamic characterization of automotive shock absorbers. The objective was to develop a new testing and analysis methodology for obtaining equivalent linear stiffness and damping of the shock absorbers for use in CAE-NVH low- to- mid frequency chassis models. Previous studies using an elastomer test machine proved unsuitable for testing shocks in the mid-to-high frequency range where the typical road input displacements fall within the noise floor of the elastomer machine. Hence, in this project, an electrodynamic shaker was used for exciting the shock absorbers under displacements less than 0.05 mm up to 500 Hz. Furthermore, instead of the swept sine technique, actual road data were used to excite the shocks. Equivalent linear spring-damper models were developed based on least-squares curve-fitting of the test data. The type of road profile did not influence the stiffness and damping values significantly for the range of amplitudes and frequencies considered. Finally, sensitivity of the vehicle level responses to the shock absorber rate change was studied, to finalize whether or not an upgrade to the existing shock absorber test procedure is necessary.

INTRODUCTION

The shock absorber is one of the most important elements in a vehicle suspension system. It is also one of the most non-linear and complex elements to model. There are two approaches to model shocks: analytical (or physical) modeling based on physical and geometrical data, and parametric modeling based on experimental data. The physical models attempt to calculate the shock absorber force as a function of displacement, velocity and acceleration from a system of differential equations. The internal pressures must be measured or predicted numerically and the system geometry in terms of areas, port diameter, architecture of the valve assemblies, etc. must be known to use these models. An iterative procedure is generally used to solve the differential equations. An exhaustive review of physical models to date is presented by Duym, et. al [1].

A comprehensive physical model was developed by Lang [2], later condensed and validated by Morman [3]. Lang's model has more than 80 parameters, is computationally complex and is not suitable for comprehensive vehicle simulation studies. Morman's model has been shown to be useful for studying the effects of design changes for a particular shock. Reybrouck [4] has developed a physical model, which has 14 parameters, valid for frequencies up to 20 Hz, but has limited appeal for the analysis of shock absorbers for NVH applications.

Simplified models using springs and dashpots in various combinations have been built. Attempts have been made to include non-linearities due to hysteresis and backlash, which lead to a set of non-linear differential equations requiring numerical solution [5]. Hence these models have limited use in total vehicle CAE studies.

The parametric modeling approach involving development of an input/output relation of the shock absorber based on experimental data is ideal for CAE simulations. In this approach, a shock absorber is characterized by **a "black-box"** system for a specific range of test conditions. The shock absorber is subjected to a known input and the output force is measured. A model is then developed from these measurements, which describes the input-output relationship. The parameters of the model may or may not have any physical meaning, but are strongly correlated with measurements. If the parameters do not have any meaning, then the model is sometimes called "nonparametric". One limitation of parametric modeling approach is that the model is valid only within the boundary of test conditions. This means, a model that has been developed using smooth road test data may not be accurate for use under rough road conditions. A parametric model from experimental data using system identification techniques has been developed by Alanoly [6].

A nonparametric model based on a **restoring force surface** mapping has been developed [7-10]. The model considers the force to be a function of displacement and velocity. Although, this model is limited to single frequency excitation, it serves as a useful tool for identifying the non- linearities in the system.

A comprehensive physical model of the shock absorber is necessary to study the effects of design changes and to tune the shock absorber to obtain the desired performance. The vendors have used physical models in the design stage. If the objective, however, is to characterize the performance of the shock absorbers for CAE simulations and benchmarking, the parametric modeling approach similar to the one presented in this paper is appropriate. *It should be noted that the parametric models are valid only within the range of test conditions.*

The objectives of this project was to develop a testing and analysis methodology for obtaining equivalent linear stiffness and damping of automotive shock absorbers for use in CAE-NVH low-to-mid frequency chassis models. The first task involved developing a suitable testing procedure including fixtures for exciting the shock absorber with a random input corresponding to different driving conditions. The second task included development of a data analysis procedure to extract equivalent linear dynamic properties from the measured data. Finally, a sensitivity analysis was conducted using a vehicle level CAE model to study the effects of stiffness and damping changes on the predicted interior sound pressure level.

TEST PROCEDURE

An electrodynamic shaker was used for exciting the shock absorbers under displacements less than 0.05 mm up to 500 Hz. Furthermore, instead of the swept sine technique as used in MTS, actual road data were used to excite the shocks. This enables the development of both non-linear as well as equivalent linear parametric models from the measured data.

Figure 1 shows a picture of the experimental set-up. As seen, the shock absorber is fixed at the tube end using a U-shaped clamp to a massive plate on a test bed. The rod end of the shock is connected to a shaker (50-lbf shaker from MB dynamics) through an impedance head (PCB Model No. 288C01). The impedance head has an accelerometer and a force transducer, both integrated into the same unit for measuring the input displacement and output force. The LMS Time Waveform Replicator (TWR Revision 3.4 under TMON) software and DIFA Scadas II (with QDAC) front-end hardware were used to generate, apply and control the input to the shaker in order to reproduce road excitations in the lab.

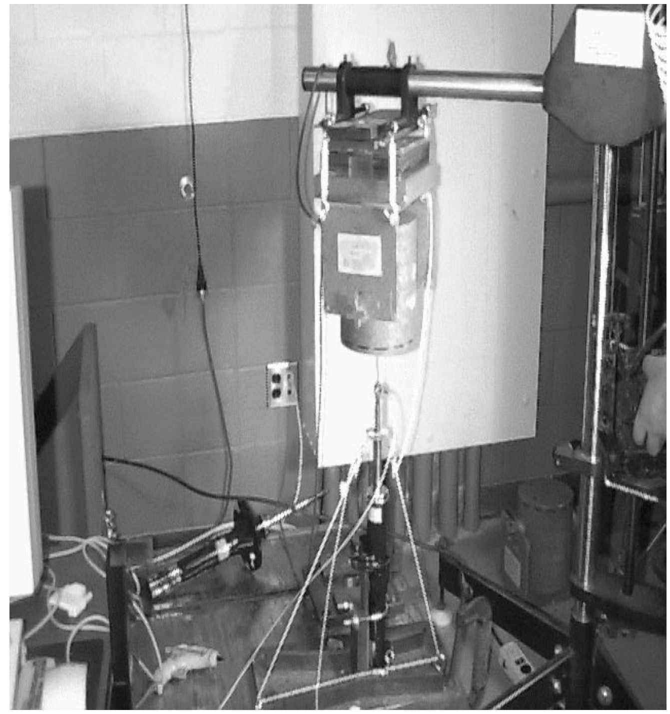

Figure 1. Experimental Set-up.

For testing under actual stoke lengths (pre-loads), a pair of thin cords and a thin aluminum plate was used as shown in the picture. The aluminum plate (size= 2 X 2 X 1/8 inch) with a hole was bonded to the rod end near the step, and a pair of thin cotton cords (about 3 mm in diameter) was attached to the plate using S-hooks. The other ends of the cords were fixed to the bottom plate. First, the cords were tied by pushing the rod to its approximate stroke length, and then the exact stroke length was adjusted and maintained by using turnbuckles in the middle of the cords.

Based on trials with other materials we decided to use the cotton cord for pre-loading the shock. Two shock absorbers were tested under the following five input data: a) Smooth Road @50 MPH, b) Rough Road @ 30 MPH, c) Spindle Shaker Lab Test with Hydraulic Shaker On d) Spindle Shaker Lab Test with Hydraulic Shaker Off, and e) Random white noise excitation (with r.m.s value of 0.005 mm). For cases a, b, and c, the relative accelerations to simulate the road input in the lab were calculated from the synchronized time history record of accelerations at the top and bottom of the shock absorber measured during road tests.

DATA ANALYSIS

A linear spring-damper model of the form $f(t) = K\,x(t) + c\,v(t)$, where x= input displacement, v= input velocity, and f (t)= output force was developed based on test data in the time domain. The term K is the spring stiffness (N/m) and c=viscous damping coefficient (N.s/m). All curve-fitting and plotting were done using MATLAB software.

In the frequency domain, for a simple harmonic excitation, the above model is interpreted as:

$F(\omega) = K\,X(\omega) + j\,c\,\omega\,X(\omega)$, where ω is the frequency in rad/sec. The input-output relationship in the frequency domain is: $F(\omega) / X(\omega) = K_R + j\,K_I$ where K_R is the real part and K_I is the imaginary part of the dynamic stiffness.

We can also write $F(\omega) / X(\omega) = K_M\,e^{j\phi}$, where K_M is the magnitude of the Dynamic Stiffness and ϕ is its phase at the frequency of excitation. Values of K_M and ϕ at various frequencies of interest can be obtained from the above linear model. It should be noted that K & c are treated as constants (independent of displacement amplitude & frequency) in the time domain, while the complex dynamic stiffness is a function of frequency if the excitation is assumed as simple harmonic. Care must be exercised in the correct use and interpretation of these models, as they are not applicable for all cases.

RESULTS & DISCUSSION

Table 1 is a summary of all results. It shows a total of eight test cases. Figures 2 through 6 refer to test No. 1, for the front shock absorber under rough road excitation. Figure 2 shows the raw data: input displacement and output force in the time domain before post-processing. The r.m.s. Values of the measured displacement and force in this case are 0.025 mm and 1.30 Newtons. Figure 3 shows subplots of the power spectral densities (PSD) of the input and output time histories. Note the rapid decrease in the original PSDs with increase in frequency. The sampling frequency for all road data was 2000 Hz, hence data up to half its value are theoretically useful. The filtered response, however, shows data only in the range 25-300 Hz. This is the frequency range in which we were able to generate valid control algorithms in all our tests without either over-loading or under-loading the shaker. The shaker displacements were either too large (below 25 Hz) or too small (above 300 Hz) outside of this frequency range.

Table 1. Summary of Results
Equivalent Stiffness (K) and Damping (c) for each test case:
Frequency Range: 25-300 Hz

Test No.	Shock Absorber	Excitation	RMS Disp., mm	RMS Force, N	Stiffness K, N/mm	Damping c, Ns/mm
1	Front	Rough Road	0.025	1.30	32.73	0.170
2	Front	Smooth Road	0.013	0.91	35.19	0.231
3	Rear	Rough Road	0.031	1.64	24.16	0.200
4	Rear	Smooth Road	0.019	1.15	25.42	0.231
5	Rear	Spindle Shaker Test with Hydraulic Shaker on	0.018	1.35	25.27	0.282
6	Rear	Spindle Shaker Test with Hydraulic Shaker off	0.006	0.68	24.26	0.410
7	Front	Lab Test- Random White Noise, No pre-load (Full Extension)	0.005	0.80	85.67	0.335
8.	Front	Lab Test- Random White Noise, with pre-load	0.005	0.57	31.76	0.383

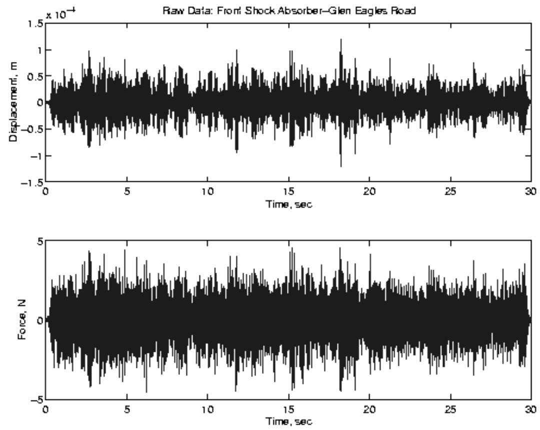

Figure 2.   Input & Output Time Histories

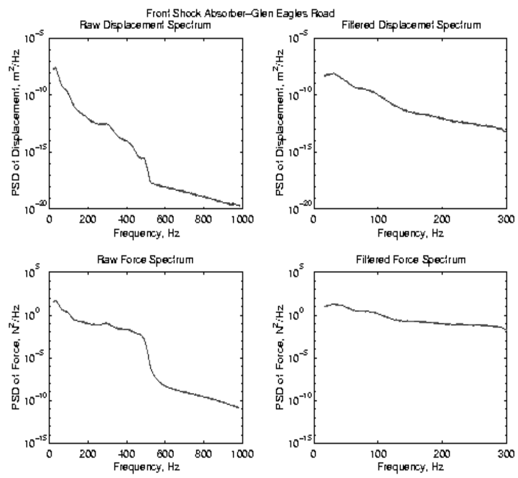

Figure 3.   PSDs of Input & Output

Figure 4 shows the force vs. displacement and force vs. velocity curves obtained by plotting the filtered time histories. The contribution of many frequencies to the stiffness and damping of the shock absorber as well the presence of strong non-linearities are quite evident from these shapes. In fact, one can easily extract the bi-linear damping exhibited by most shocks under low frequencies from these hysteresis loops. For linear systems with no "memory" the shape of force vs. displacement and force vs. velocity would be a simple ellipse for a single sine wave excitation.

Next, a comparison of the measured vs. model force is shown in Figure 5 in two different formats. The model force is generated from the curve-fitting constants. The accuracy of the model varies with each test. It is seen that an ideal linear model is one in which all the dots lie on the straight line in Figure 5. Finally, Figure 6 plots the magnitude and phase of the dynamic stiffness as a function of frequency useful for linear frequency domain CAE analysis.

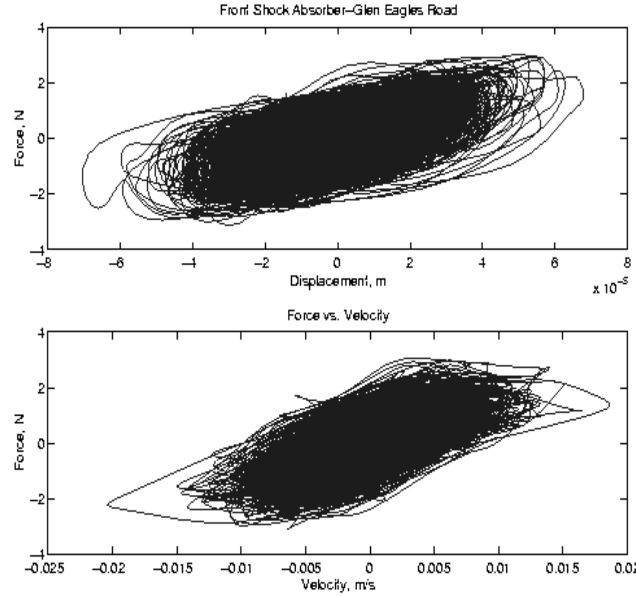

Figure 4.   Force Vs. Displacement & Force Velocity Plots

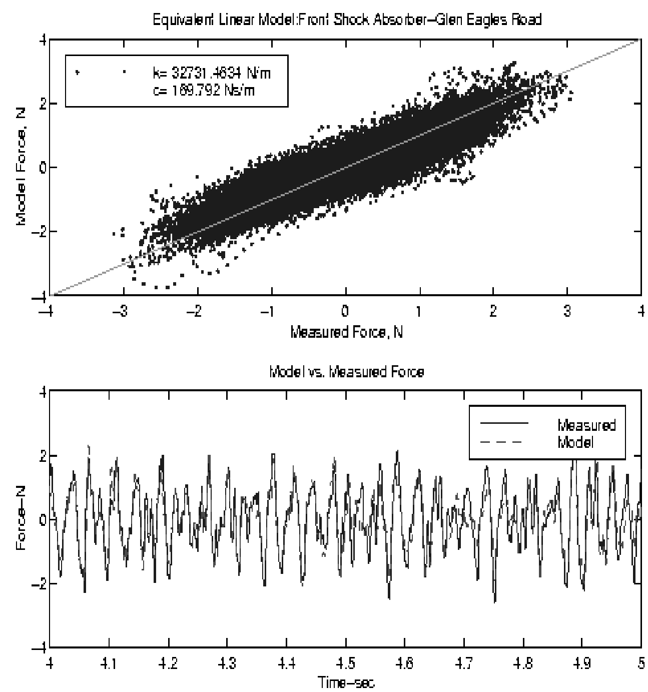

Figure 5.   Equivalent Linear Model

The last two tests (Test Nos. 7 & 8 in Table 1) need some explanation.   These test were conducted on the front shock absorber using a random white noise excitation (r.m.s value= 0.005 mm in the frequency range 25-300 Hz). Test No. 7 was with no pre-load that is with full extension of the rod.   The results of Table 1 and the accompanying plots lead us to the following general conclusions:

- The shock absorber can be modeled as a linear spring-damper system for the range of amplitudes and frequencies considered. The model fits especially well for the smooth road excitation.
- The stiffness and damping values appear to not change much with the type of road excitation. This may be because of the very low levels of displacements considered in this work.
- The front shock absorber is considerably stiffer than the rear shock absorber for all excitations. This could be due to the differences in the internal "tuning" of the shock.

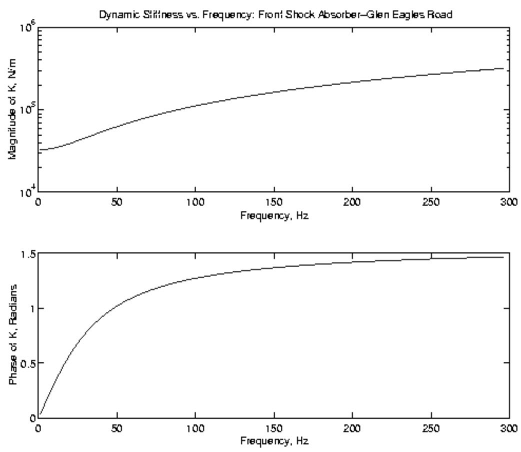

Figure 6. Dynamic Stiffness & Damping vs. Frequency

CAE ANALYSIS AND CONCLUSIONS

CAE analysis was conducted using a total vehicle NVH CAE model. We compared vehicle responses of the two models, the baseline model whose shock absorber rates were based on an earlier test procedure, and a modified model, whose shock absorber rates were based on the new test procedure.

The shock absorber rates for the baseline model were K=464 N/mm and C=1.75 Ns/mm. The shock absorber rates for the modified model were K=32.73 N/mm, C=. 17 Ns/mm for the front shocks, and K=24.16 N/mm, C=. 200 Ns/mm for the rear shocks.

The comparison of the vehicle model prediction of the interior sound for the baseline and the modified shock rates are shown in Figure 7. Frequency range of analysis is 1-250 Hz. The model is most sensitive to the change in the shock absorber rates below 50 Hz. By viewing the forced response animation of the vehicle, wheel hop/ tramp modes dominate vehicle responses below 50 Hz. As a result, shock axial motion is active making vehicle sensitive to the changes in the shock rates. At some higher frequency bands, e.g., 145-155 Hz, shock axial motion is still active, but the lateral motion of the strut/ shock has the larger contribution to the interior responses, making vehicle responses less sensitive to the shock rate changes. For the most frequencies above 50 Hz, there is very little or no difference between the two models.

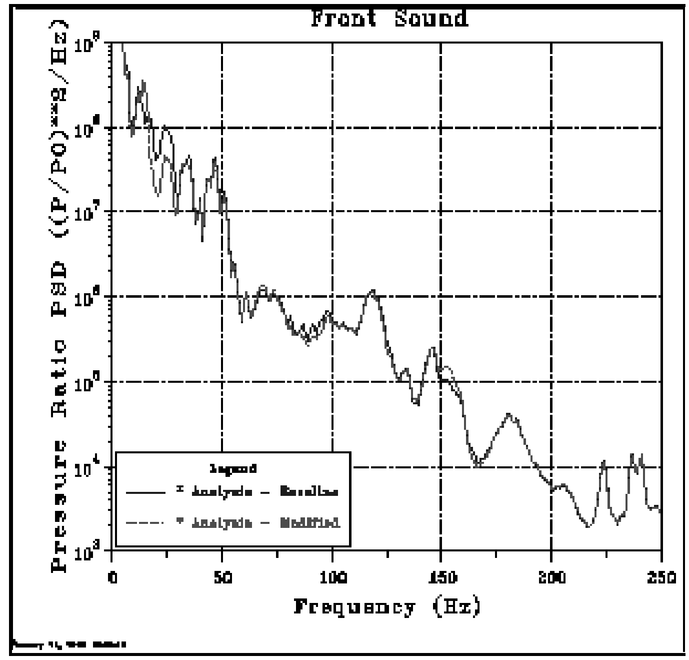

Figure 7. Sensitivity of Vehicle Model to Different Shock Absorber Rates

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