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Development of a Nonlinear Shock Absorber Model for Low-Frequency NVH Applications

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ABSTRACT

This paper discusses the development of a nonlinear shock absorber model for low-frequency CAE-NVH applications of body-on-frame vehicles. In CAE simulations, the shock absorber is represented by a linear damper model and is found to be inadequate in capturing the dynamics of shock absorbers. In particular, this model neither captures nonlinear behavior of shock absorbers nor distinguishes between compression and rebound motions of the suspension. Such an inadequacy limits the utility of CAE simulations in understanding the influence of shock absorbers on shake performance of body-on-frame vehicles in the low frequency range where shock absorbers play a significant role.

Given this background, it becomes imperative to develop a shock absorber model that is not only sophisticated to describe shock absorber dynamics adequately but also simple enough to implement in full-vehicle simulations. This investigation addresses just that. The developed model is nonlinear and is constructed using control-force data of shock absorbers. While the model maintains simplicity without increasing vehicle model size, it describes shock absorber behavior both in compression and rebound. The shock absorber model is implemented in full-vehicle simulation of a full-size pickup truck, and the vehicle shake and impact harshness performances are evaluated. Numerical results show the influence of using a nonlinear model in lieu of a linear model. Moreover, a parametric study with respect to input excitation level shows that for large displacements of suspension, nonlinear damping plays a significant role in controlling the response. The nonlinear model also captures the frequency dependency of shock absorber characteristics; this offers considerable promise in analytically tuning shock absorber characteristics for different frequencies of operation.

INTRODUCTION

The shock absorbers are primarily designed to dissipate energy stored in suspension due to external disturbances by providing damping in the suspension. Such a characteristic is vital for improved ride comfort, vehicle control as well as to reduce the impact of modal resonance. The shock absorbers produce damping force in response to suspension motions, and are designed to have different damping for jounce (compression) and rebound motions. It is customary to have less damping for compression motion than that of rebound motion so that less force is transmitted to the vehicle when it encounters bump-type disturbances. By comparison, more damping is provided for rebound motion in order to dissipate energy stored in the suspension system quickly (Ref. 1). In general, the suspension of body-on-frame vehicles goes through large displacements even at frequencies relatively higher than that of ride motions. By virtue of their design, the shock absorbers exhibit nonlinear variation of control forces with respect to velocity under large displacements. In summary, shock absorbers are tuned to have different damping in compression and rebound, and exhibit nonlinear and frequency dependent control-force variations. These shock absorber characteristics significantly influence vehicle shake and impact harshness performances, which are low frequency NVH (Noise, Vibration and Harshness) issues occurring at frequencies less than 20Hz, of body-on-frame vehicles.

Traditionally, in full vehicle CAE (Computer Aided Engineering) simulation models, the shock absorber is modeled as a linear viscous damper using a set of linear spring and dashpot. Such a representation neither captures frequency dependency nor distinguishes between compression and rebound motions. Accordingly, considerable work has been carried out in developing shock absorber models; for a detailed review of these models, see Refs. 2 and 3. In general, these models fall in two categories. The first category consists of analytical models that are based on physical and geometrical data; these models predict the generated control force numerically using physical construction details of shock absorbers. Hence, the use of such models in full vehicle simulations can become cumbersome as well as computationally prohibitive since the control forces are predicted by solving a large number of algebraic and differential equations in an iterative manner. Alternatively, parametric models are developed and implemented in a few full vehicle simulations (Ref. 2). These models are based on input/output relation of shock absorbers and use experimental data. The shock absorbers are tested for known inputs and output force data are generated. Then models are developed from these measurements by using either system identification or least square estimation techniques. Rao and others (Ref. 2) proposed a model using such an approach in which the shock absorber is modeled in terms of its stiffness and damping, which are considered to be linear. To characterize the shocks, small amplitude excitations are used over a frequency range up to 300Hz. Furthermore, they demonstrated its utility in full vehicle simulations of road noise response. Although this model is simple to implement in full vehicle simulation, it has limited appeal for operating conditions in which the suspension goes through large displacements as encountered by body-on-frame vehicles.

The model developed in the present investigation focuses on capturing the dynamics of shock absorbers for large displacement events. It advances the state of the art in that it is a nonlinear model and distinguishes between compression and rebound motions of the suspension. Given this background, we have two objectives:

1. Develop a nonlinear shock absorber model that is accurate and simple enough to implement in full vehicle NVH simulations; and
2. Demonstrate the practicality of the developed model by simulating shake and impact harshness performances of a large pick-up truck.

NONLINEAR SHOCK ABSORBER MODEL

The nonlinear shock absorber model uses control force vs. velocity data, which are generated by testing shock absorbers for known inputs. In the present study, the shock absorbers are subject to pure sine wave for a frequency sweep in MTS machine. A peak-to-peak displacement of 2mm is given at one end and force measurements are carried out at the other end of the shock absorber. It is noteworthy to mention that the developed model uses only the force vs. velocity data and is independent of the type of excitation used in characterizing shock absorbers. A detailed exposition on the impact of type of input excitation used in characterizing shock absorber is given in Ref. 3. Figure 1 shows typical force-velocity data for the shock absorber used in this study. As shown, there are two distinctive force curves: one for compression and the other for rebound. For this shock absorber, the rebound curve is nearly linear and the compression curve is highly nonlinear. These curves are parameterized using a regression curve-fitting technique to obtain a nonlinear expression of the form $F = cV^E$, where F is the force, c is a coefficient, V is velocity across shock and E is an exponent. Thus, c and E represent the parameters of the nonlinear shock absorber model. For the shock absorber tested, the coefficient c is 1.86 and exponent E is 1.04 for the rebound curve, and similarly, the parameters are 11.67 and 0.55, respectively, for the compression curve. By comparison, in a linear model, the force is given by the linear expression $F = c_d V$, where c_d is the damping coefficient, which is same for both compression and rebound curves.

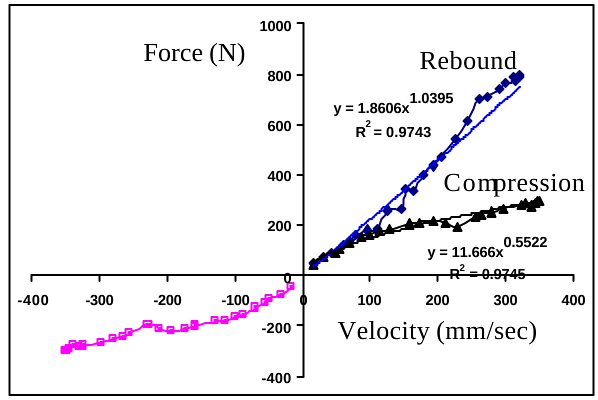

Figure 1: Force vs. Velocity Curves

The nonlinear model is implemented in MSC/NASTRAN using CBUSH1D and PBUSH1D card entries; for details, see Ref. 4. The element has axial stiffness and damping as shown in Fig. 2. It needs local coordinate system to be defined at coincident grids. The element supports large displacement. The total elemental force is a combination of damping force and stiffness force, and is nonlinear. Hence, the use of this element requires SOL129, which is nonlinear transient response analysis (Ref. 4).

Figure 2: Nonlinear Element Formulation in MSC/NASTRAN

APPLICATION

The nonlinear shock absorber model is implemented in a full vehicle NVH simulation model of a full-size pick-up truck. To achieve reduced turnaround time, condensed models are used to represent cab, frame, box and control arms; in other words, DMIG models are used instead of detailed finite element models (Ref. 5). Overall, the full vehicle NASTRAN model has 35000 grids, 33000 elements and 24000 DMIG entries. For simplicity, the same shock absorber is used in both front and rear suspensions. This means all four shock absorbers in the vehicle are modeled using the same parameters as identified earlier.

The full vehicle model is exercised to evaluate shake and impact harshness performance of the truck. To simulate shake performance, the excitations are input only on the passenger side with time delay between front and rear tires as shown in Fig. 3. The time delay is calculated based on the vehicle speed and the wheelbase. In this case, the vehicle traverses on a recessed manhole cover of 18” diameter and one-inch depth at 32mph. To simulate impact harshness, excitations are applied in-phase at both front and rear wheels with time delay between the front and rear wheels. The input excitation corresponds to the condition in which vehicle traverses over a concrete slab of two-inch width and one-inch height. Figure 4 shows the input condition for simulating impact harshness performance.

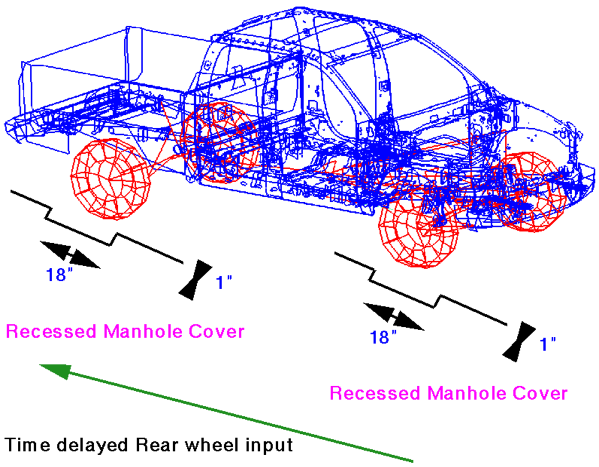

Figure 3: Shake Excitation Input Model

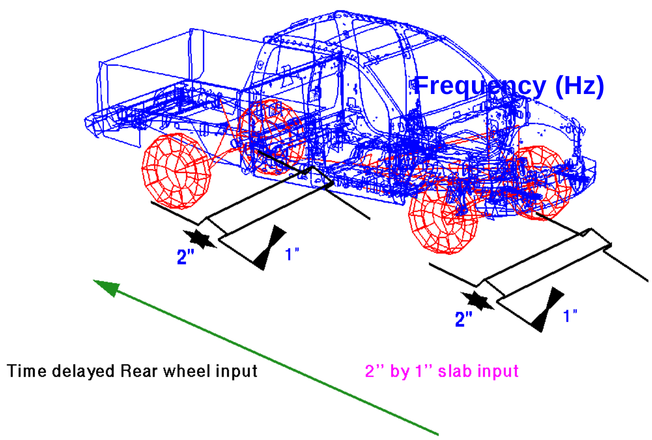

Figure 4: Impact Harshness Excitation Input Model

NUMERICAL RESULTS

In this section, we present numerical results on shake and impact harshness performance of the truck. These results are presented in two segments. While the first segment addresses shake results (Figs. 5-10), the second segment focuses on impact harshness (Figs. 11 and 12). We begin with Fig. 5, which demonstrates whether the nonlinear shock absorber model works or not; that is, to demonstrate that the model can distinguish between compression and rebound motion as well as limit force build-up with respect to velocity. In particular Fig. 5a compares test data with simulation results for control force variation. As shown, the simulation results obtained by using the nonlinear shock absorber model follow the test data, which means the element formulation is capable of mimicking the test curves. Similarly, Fig. 5b provides a comparison between two sets of simulation results. In other words, it shows the difference between using the linear and nonlinear shock absorber model. This comparison shows that linear shock absorber model puts in high shock forces particularly during compression motions.

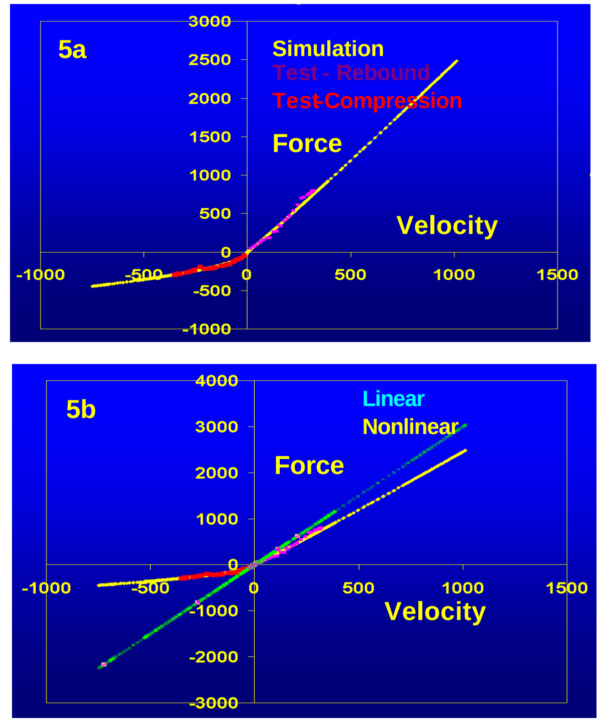

Figure 5: Comparison for Force vs. Velocity

Figure 6 compares response at driver's seat by using linear shock absorber model. In the left side plots, velocity at the seat is presented as a function of time, and in the right side plots, it is presented as a function of frequency. The results are from two cases. In the first case, the vehicle traverses over a pothole and in the second case, the vehicle traverses over a bump. In the first case, the shock goes through extension to begin with, and in the second case, the shock is in compression. Since same damping is used in both compression and rebound, we see same response magnitude in both time and frequency domains for both cases. But in the time domain, the two responses are out-of-phase. Such a response with same magnitude for both compression and rebound is in total contrast to that of on-road events, which demonstrates the inadequacy of the linear model for shock absorbers.

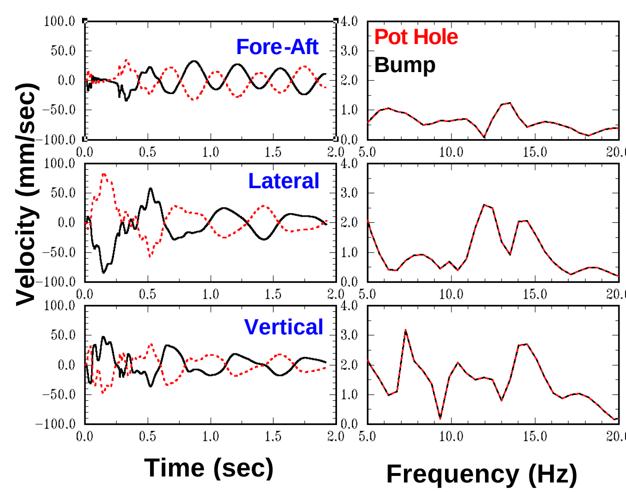

Figure 6: Seat Response Comparison with Linear Shock Absorber Model

The seat response comparison for the same cases as considered in Fig. 6 but with the nonlinear shock absorber model is given in Fig. 7. As expected, the results are different in two respects. First, the response magnitude is not the same; in fact, there is about 37% difference in response magnitude at vehicle beaming frequency (7.25Hz) between the two cases. Second, the response characteristics are altered in the sense that they are no longer symmetric. Such a variation in response can be captured only by using nonlinear shock absorber model since it accounts for different damping for compression and rebound motions.

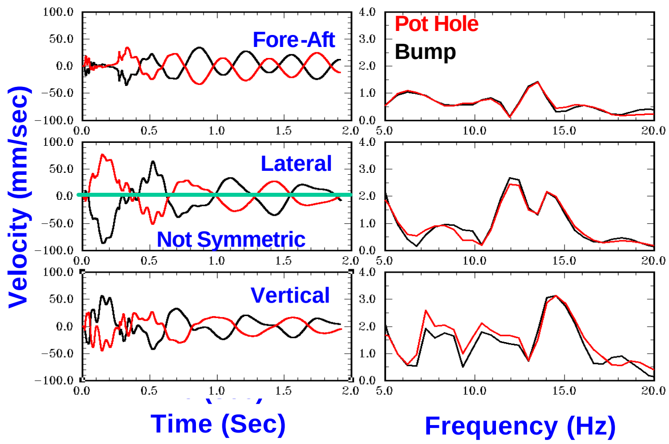

Figure 7: Seat Response Comparison with Nonlinear Shock Absorber Model

In the next figure (Fig. 8), we discuss the effect of reduced damping on seat response. The intent is to investigate the impact of using the nonlinear shock absorber model in capturing frequency dependency. For simplicity, we present only the vertical component of the seat response. The top plot of Fig. 8 shows response in time domain and the bottom plots shows the response in frequency domain. The results are for four cases: (i) linear shock absorber model; (ii) nonlinear shock absorber model; (iii) linear shock absorber model with reduced damping; and (iv) nonlinear shock absorber model with reduced damping. Two points are noteworthy. The ensuing oscillations after the front wheel gets the hit control the peak response at 14.5 Hz in the shake frequency range. Similarly, with a delay, the rear wheel gets the hit and the subsequent oscillations control response at 7.25Hz in the beaming frequency range. Compared to nonlinear model prediction, the linear model over predicts at 7.25Hz and under predicts at 14.5Hz. This is because the linear model overestimates the shock damping throughout the frequency range. By contrast, the nonlinear model includes slightly lower damping than the linear model does in the low frequency and includes much lower damping at 14.5Hz. When the damping is reduced, the linear model shows an increase of 20% in response magnitude at 14.5Hz and the nonlinear model predicts an increase of only 8%. Although the magnitude of damping is reduced, the nonlinear shock absorber model still follows a nonlinear curve, which limits the magnitude of input force into the suspension.

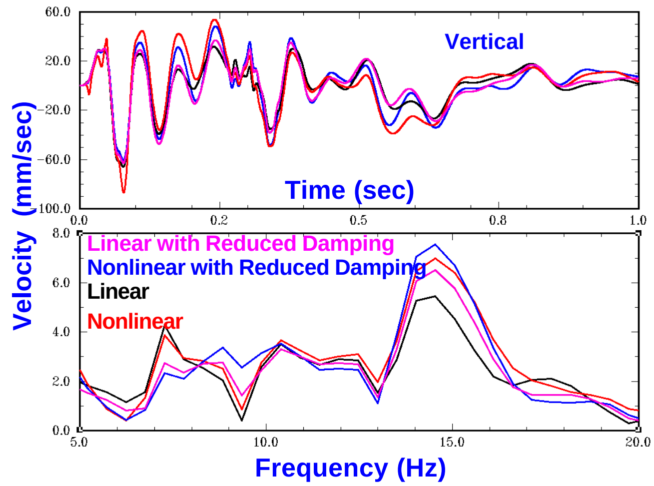

Figure 8: Effect of Reducing Damping on Seat Response

In Figs. 9 and 10, we discuss the effect of increased input force magnitude. For this, we vary the depth of the manhole cover. The results are for two cases. While the depth is one inch in the first case, it is two inches in the second case. As seen from Fig. 9, the linear model reacts to the increase in input force linearly and accordingly, the model predicts increased response. Whereas, the nonlinear model limits the input forces and thereby controls the rate of increase in response magnitude at 7.25Hz. This can also seen in Fig 10, which shows that the linear model puts in more forces into the suspension system than the nonlinear model does.

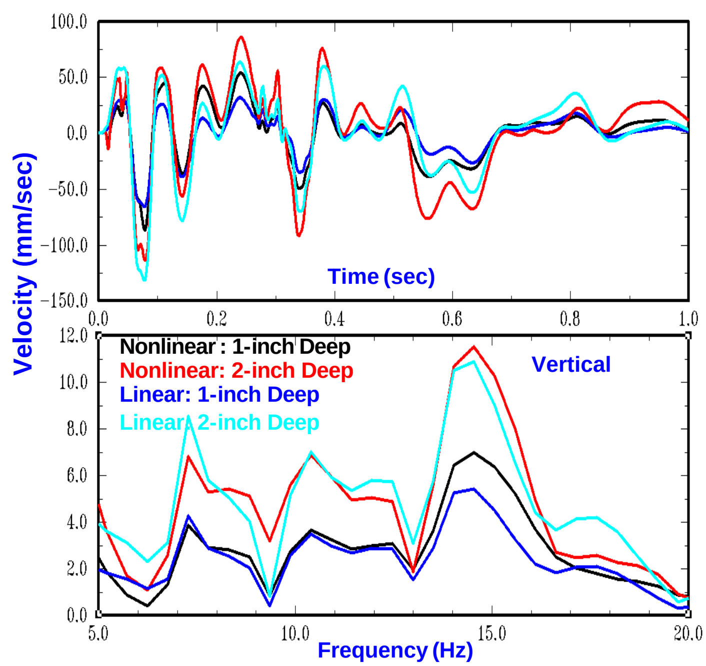

Figure 9: Effect of Input Force on Seat Response

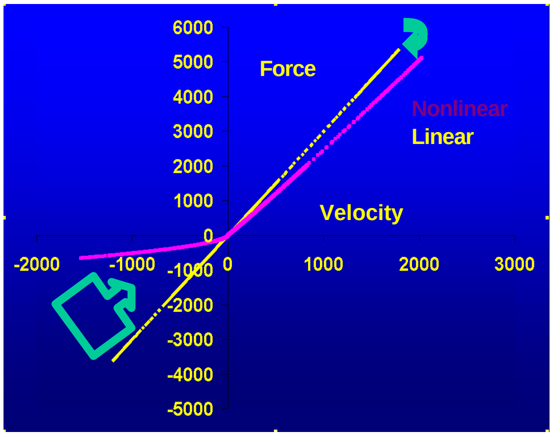

Figure 10: Control Force Comparison between Linear and Nonlinear Shock Absorber Models for 2inch Depth Manhole Cover

Finally, we discuss the second segment of results (Figs. 11 and 12), which focuses on impact harshness simulation. The results correspond to an on-road event where the vehicle traverses over a concrete slab of 2”x1” section at 20 mph (see Fig. 4). The front wheels get in-phase inputs first and then the rear wheels get in-phase inputs with a time delay. Here, the focus is on the effect of damping introduced in the suspension system by the linear and nonlinear shock absorber models. Specifically, Fig. 11 shows velocity comparison at driver’s seat. As shown, there is a significant difference in response magnitude, in particular, in the vertical component. As seen from Fig. 12, with the linear model, the damping force is overestimated when compared to the nonlinear model and consequently, the response is higher at 7.25Hz and lower at 14.5Hz. These effects can also be seen in time history signals. As the rear wheels get the hit, the shock absorbers go through compression. Under such motion, the linear shock absorber model puts in more force in to the suspension when compared to the nonlinear model and thereby predicts higher response. When the damping is reduced, the nonlinear shock absorber model shows reduced response at 7.25Hz and increased response at 14.5Hz. This also shows that the shock absorbers in the front suspension need to have more damping than that of the rear suspension so that the ensuing oscillations after the front wheels get the hit will be absorbed quickly.

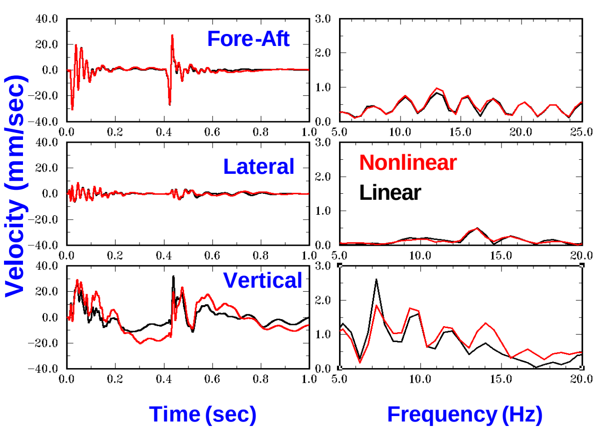

Figure 11: Seat Response Comparison for Impact Harshness Simulation

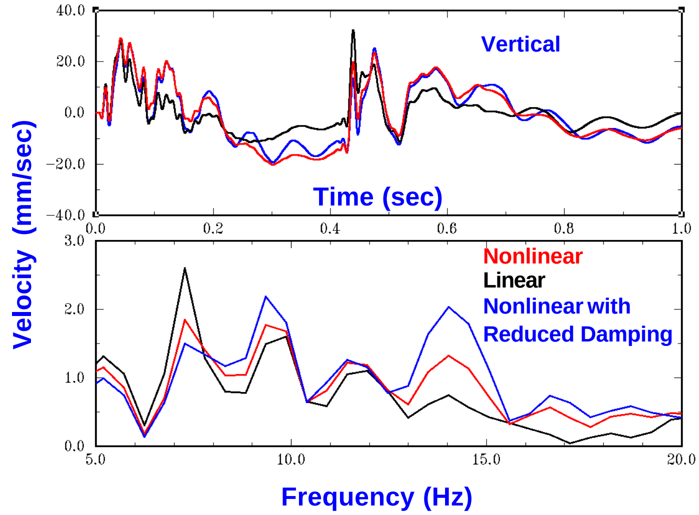

Figure 12: Effect of Damping on Impact Harshness Response

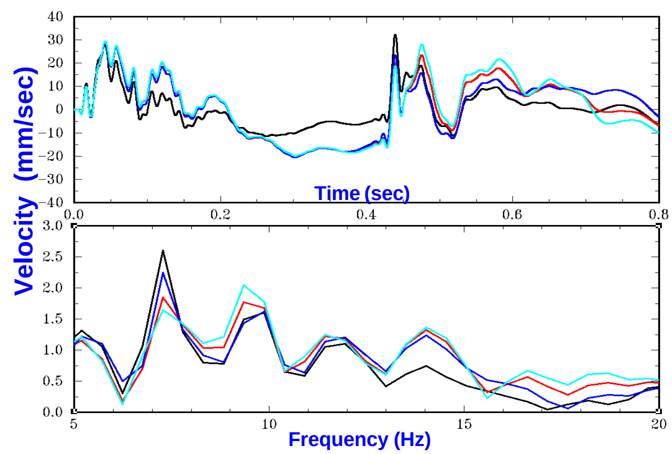

Figure 13: Effect of Reducing Compression Damping on Impact Harshness Response (Legend: Linear, Nonlinear, Nonlinear with Reduced Compression Damping, Nonlinear with Increased Compression Damping)

In Fig. 12, the damping for both compression and rebound motions are reduced. However, it is widely recognized that reducing damping for compression motion in relation to rebound motion is a prudent approach while attempting to improve vehicle response (Ref. 1). Accordingly, an analytical study is conducted in understanding the influence of damping variation for compression motion only. The results of this study are given in Fig. 13, which show that the magnitude of response at seat is improved by nearly 11% at 7.25 Hz when the damping for compression motion is reduced. Based on these findings, a shock absorber was built with low-compression damping and the vehicle beaming performance (at frequency 7.25Hz) was evaluated subjectively. The overall conclusion based on this ride evaluation was a half-point improvement in beaming performance (Ref. 6). Further work is in progress in tuning shock absorber force-velocity curves for both rebound and compression motions to improve vehicle response at frequencies 7.25 Hz and 14.5 Hz.

CONCLUSIONS

A nonlinear shock absorber model is developed and implemented in full-vehicle simulation of shake and impact harshness performances of a full-size pick-up truck. The developed model is simple and accurate in capturing the dynamics of shock absorbers for road events wherein the suspension goes through large displacements. Specifically, the nonlinear shock absorber model distinguishes between compression and rebound motions of the suspension and captures nonlinear and frequency dependent characteristics; this offers considerable promise in analytically tuning shock absorbers for improving shake and impact harshness performance of body-on-frame vehicles.

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