2010 NDIA GROUND VEHICLE SYSTEMS ENGINEERING AND TECHNOLOGY SYMPOSIUM
MODELING & SIMULATION, TESTING AND VALIDATION (MSTV) MINI-SYMPOSIUM
AUGUST 17-19 DEARBORN, MICHIGAN

INFLUENCE OF SHOCK ABSORBER MODEL FIDELITY ON THE PREDICTION OF VEHICLE HALF ROUND PERFORMANCE

Chris Masini
Oshkosh Corporation
Oshkosh, WI

Xiaobo Yang, PhD
Oshkosh Corporation
Oshkosh, WI

ABSTRACT

Model based design techniques are being used increasingly to predict vehicle performance before building prototype hardware. Tools like ADAMS and Simulink enable very detailed models of suspension components to be developed so vehicle performance can be accurately predicted. In creating models of vehicle systems, often there is a question about how much component detail or model fidelity is required to accurately model system performance. This paper addresses this question for modeling shock absorber performance by comparing a low fidelity and high fidelity shock absorber model.

A high fidelity and low fidelity mathematical model of a shock absorber was developed. The low fidelity shock absorber model was parameterized according to real shock absorber hardware dimensions. Shock absorber force vs. velocity curves were calculated in Simulink. The results from the low fidelity and high fidelity model were compared to shock absorber force vs. velocity test results.

New vehicle designs must meet requirements for maximum driver's seat acceleration during half round testing. These requirements have created a need for predicting half round performance. The low fidelity and high fidelity shock absorber models were placed in a 7 Degree-of-Freedom vehicle model. Maximum driver's seat acceleration was calculated and simulation results were compared to half round test results from a prototype vehicle for a 10mph, 8" half round test.

INTRODUCTION

Suspension systems for a new vehicle design must meet various performance requirements such as ride and handling, etc. An efficient suspension design is usually based on the ability to accurately predict suspension performance using computer simulation tools such as ADAMS or Simulink, where the fidelity of the vehicle simulation model is dependent on the fidelity of various sub-component models.

In view of the critical contribution of shock absorbers to vehicle dynamic response under half-round events, in this study, the influence of shock absorber model fidelity on prediction of half round performance of a four-wheeled vehicle is investigated. A detailed hydraulic model of a shock absorber was created in Simulink and the calculated force-velocity profile was then compared to shock absorber test results to demonstrate the high fidelity of the shock absorber model. On the other hand, a set of simplified equations were then developed to model the force-velocity curve provided by the shock absorber manufacturer. The simulation results from the simplified equations were compared to those from the Simulink model and shock absorber test results.

Finally, vehicle half round test results from a four-wheeled vehicle were compared to vehicle simulation results. Half round simulation results were calculated from a 7 degree-of-freedom (DOF) Simulink model of a four-wheeled vehicle. Simulation runs were made with the simplified shock absorber equations and the more complex hydraulic shock absorber model. A comparison of time domain traces and peak responses for half round simulation results and test results was made to show the influence of shock absorber model fidelity on prediction of vehicle half round performance.

FORCE VS. VELOCITY CURVES

Shock absorber damping is one of the primary means to tune the ride and handling of vehicle suspension systems. Vehicle designers use shock absorber force vs. velocity curves to specify the appropriate shock absorber damping characteristics. An example force vs. velocity curve is shown in figure 1. The force vs. velocity curve shows how much force a shock absorber will generate for a given amount of velocity during extension (REB) and compression (COMP). Shock absorbers are typically tested according to SAE J1574-1, "Measurement of Vehicle and Suspension Parameters for Directional Control Studies" [1]. The force vs. velocity curve is generated by applying a linear velocity or a sinusoidal input to the shock absorber.

The force vs. velocity curve is a primary communication tool between vehicle designers and shock absorber manufacturers. There are several different shock absorber designs including position sensitive, adjustable damping and variable damping shock absorbers. These designs form a damping envelope on the force vs. velocity curve. The work in this paper will focus on a shock absorber design with a single force vs. velocity curve as shown in figure 1.

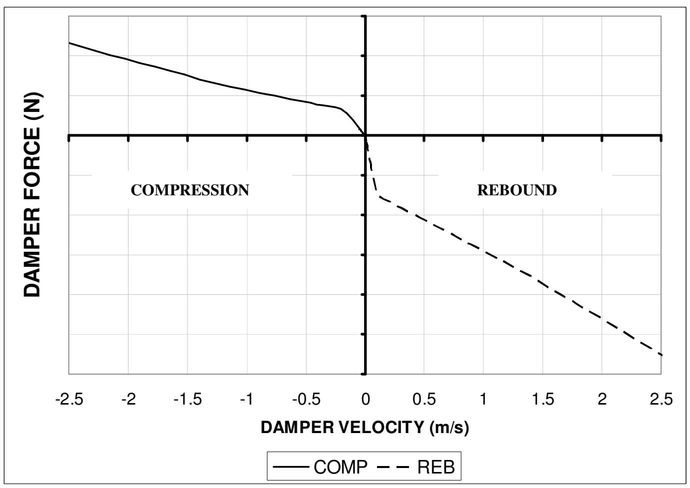

Figure 1: Shock Absorber Measured Force vs. Velocity

HIGH FIDELITY SHOCK ABSORBER MODEL

A high fidelity numerical model of the hydraulic circuit inside a shock absorber was created in Simulink. Figure 2 illustrates a schematic view of the hydraulic damper. Dynamic equations for the damper relief valve, damper check valves, damper volume and piston were developed. The governing equations for the model are developed below. The piston, relief poppet and check poppet mass dynamics are described by Newton's 2nd Law:

$$\begin{aligned}&\sum F = m \times \ddot{x}\\&\sum F = sum\ of\ \ forces\ applied\ to\ the\ mass\\&m = piston\ or\ poppet\ mass\\&\ddot{x} = piston\ or\ poppet\ acceleration\end{aligned} \tag{1}$$

The relief poppet and check poppet have a spring force which is calculated using the linear spring equation:

$$\begin{aligned}&F_{spring} = K_{spring} \times x + F_{spring\_preload}\\&K_{spring} = spring\ rate\\&x = poppet\ displacement\\&F_{spring\_preload} = spring\ preload\ on\ poppet\end{aligned} \tag{2}$$

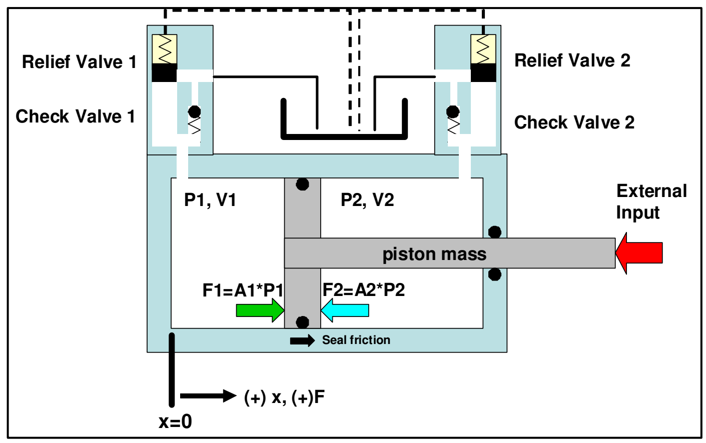

Figure 2: Schematic of Shock Absorber Hydraulic Circuit

The piston, relief poppet and check poppet have a force applied to them by the exposed pressure area:

$$\begin{aligned}&F_{area} = P_{opening\_pressure} \times A_{pressure\_area}\\&P_{opening\_pressure} = pressure\ applied\\&\qquad to\ piston\ or\ poppet\\&A_{pressure\_area} = area\ of\ \ piston\ or\ poppet\end{aligned} \tag{3}$$

For the relief and check poppet, flow area was calculated using a simple annular opening area as a function of poppet position using the equations:

$$A_{opening} = \pi \times D_{poppet\_diameter} \times x$$
$$D_{poppet\_diameter} = poppet\ diameter \qquad (4)$$
$$x = poppet\ displacement$$

A simple pi*d*x relationship was selected for the relief and check poppet because the shock absorber has a simple disc valves to relieve pressure in the damping piston. As the disc valve opens, it forms an annular area between the disc and the damping piston which is defined by equation (4).

Flow through the relief and check poppet was calculated using the orifice equation [2]:

$$Q_{poppet\_flow} = C_d \times A_{opening} \times \sqrt{\frac{2}{\rho} \times P_{opening\_pressure}}$$
$$C_d = discharge\ coefficient \qquad (5)$$
$$\rho = fluid\ density$$

A discharge coefficient of 0.62 and a fluid density of 850kg/m^3 was used for the hydraulic fluid in the shock absorber.

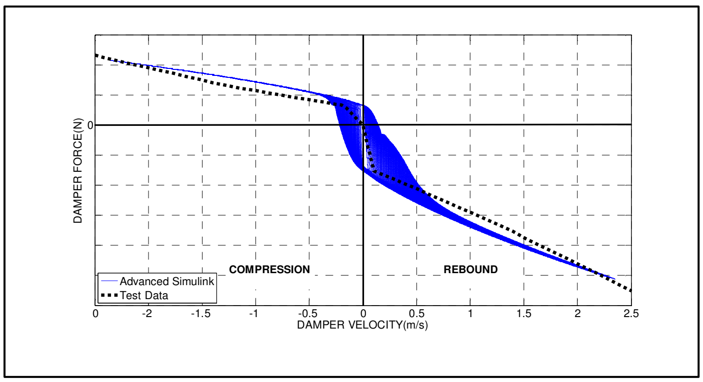

Figure 3: Shock Absorber Measured and High Fidelity Simulink Force vs. Velocity

The pressure inside the hydraulic cylinders was calculated using the dynamic pressure / flow equation [2]:

$$\dot{P} = \frac{B}{V} \times \sum (Q_{in} - Q_{out})$$
$$B = fluid\ bulk\ modulus$$
$$V = volume\ of\ shock\ absorber \qquad (6)$$
$$Q_{in} = sum\ flow\ into\ shock\ absorber$$
$$Q_{out} = sum\ of\ flow\ out\ of\ shock\ absorber$$

The bulk modulus for the selected hydraulic fluid was 1,000 MPa or 145,000psi [3].

The flow into the cylinder must take into account the motion of the piston. The time rate of change of volume in the shock absorber was calculated from piston velocity:

$$Q_{piston\_flow} = A_{piston\_area} \times \dot{x}_{piston}$$
$$A_{piston\_area} = area\ of\ shock\ absorber\ damping\ piston \qquad (7)$$
$$\dot{x}_{piston} = damping\ piston\ velocity$$

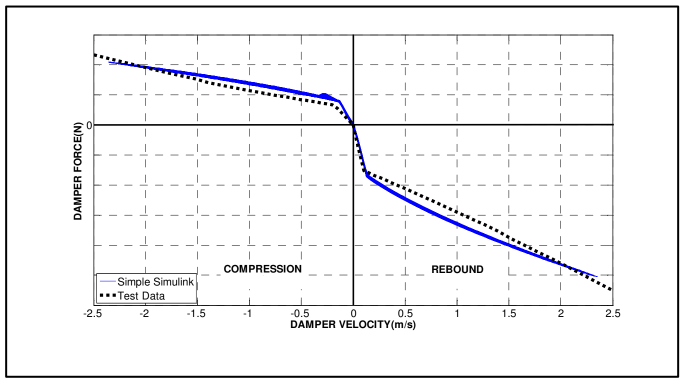

Figure 4: Shock Absorber Measured and Low Fidelity Simulink Force vs. Velocity

Damping piston velocity was calculated by integrating the acceleration of the poppet or piston mass.

To simulate the force vs. velocity curve in Simulink, the shock absorber model was constrained and simulated according to SAE J1574-1[1]. A sinusoidal input of +/-150mm was applied to the shock absorber model. The force applied to the shock absorber and the resultant velocity of the shock absorber model was calculated and plotted in figure 3. Figure 3 shows simulation results from the high fidelity shock absorber compared with test results for the shock absorber.

The high fidelity shock absorber model force vs. velocity profile showed good correlation with test results. The governing equations selected for the hydraulic components provided an accurate prediction of the force vs. velocity behavior of the shock absorber. There were some disadvantages to the high fidelity model. The high fidelity shock absorber model was tested with an ADAMS model of the vehicle. The ADAMS model would not run using co-simulation with the high fidelity Simulink shock absorber model because the high fidelity shock absorber model had high frequency components that stalled the co-simulation. A simplified representation of the shock absorber was needed to run the co-simulation with ADAMS.

LOW FIDELITY SHOCK ABSORBER MODEL

A simplified set of governing equations for the behavior of the hydraulic relief valve inside the shock absorber was developed. The simplified model was needed for use with ADAMS and to speed simulation times in Simulink. The simplified governing equations for the shock absorber were developed based on an algebraic relationship between pressure and flow through the shock absorber relief. The difference between the low fidelity model and the high fidelity model is the low fidelity model does not include the second order mass dynamics of the shock absorber piston, relief poppet, and check poppet. The mass dynamics are defined by equation (1). Also, the low fidelity model does not include the first order pressure dynamics of the shock absorber fluid volumes defined by equation (6). All other equations are similar between the low fidelity and high fidelity models.

The flow through the relief valves in the low fidelity shock absorber damping piston was calculated using external velocity inputs to the shock absorber. The source of external velocity input for the force vs. velocity simulation was a sinusoidal velocity input. The shock absorber velocity input for the half round simulation was calculated from the relative motion between the suspension and the body of the vehicle in the 7 degree-of-freedom model described below in the section titled '7 DOF Vehicle Model'. The velocity inputs to the Simulink shock absorber models were modeled after the real test inputs from the force vs. velocity test and from the half round test.

The following equations were developed for the low fidelity shock absorber model. The force vs. velocity performance of the shock absorber was parameterized using an algebraic relationship between pressure and shock absorber velocity. The model parameters represent real shock absorber component dimensions. For the low fidelity model, the orifice flow equation was used to calculate the flow through the damping piston. This is the same as equation (5) which was used for the high fidelity model [2]:

$$
\begin{aligned}
&Q_{relief} = C_d \times A_{relief} \sqrt{\frac{2}{\rho} \times \Delta P_{relief}} \\
&Q_{relief} = relief\ \ flow \\
&C_d = orifice\ \ coefficient \\
&A_{relief} = relief\ \ area \\
&\rho = fluid\ \ density \\
&\Delta P_{relief} = pressure\ \ drop\ \ across\ \ relief
\end{aligned}
\tag{8}
$$

The equation for relief valve opening area is the same as equation (4):

$$
\begin{aligned}
&A_{opening} = \pi \times D_{poppet\_diameter} \times x \\
&D_{poppet\_diameter} = poppet\ diameter \\
&x = poppet\ displacement
\end{aligned}
\tag{9}
$$

For a spring loaded relief poppet, the opening height is related to pressure by the equation:

$$
\begin{aligned}
&x_{poppet} = \frac{\Delta P_{relief} \times A_{poppet} - F_{preload\_spring}}{K_{spring}} \\
&A_{poppet} = poppet\ \ pressure\ area \\
&F_{preload\_spring} = relief\ \ spring\ \ preload \\
&K_{spring} = relief\ \ spring\ \ rate
\end{aligned}
\tag{10}
$$

Flow through the damping piston is related to the damping piston pressure by the equation:

$$
\begin{aligned}
&Q_{relief} = C_d \times \pi \times D_{relief} \times x_{poppet} \times \sqrt{\frac{2}{\rho} \times \Delta P_{relief}} \\
&where: \\
&x_{poppet} = \frac{\Delta P_{relief} * A_{poppet} - F_{preload\_spring}}{K_{spring}}
\end{aligned}
\tag{11}
$$

Finally, flow through the damping piston is related to the shock absorber piston velocity by the equation:

$$\dot{x}_{piston} = Q_{relief} \,/\, A_{piston\_area}$$
$$A_{piston\_area} = area\ of\ \ shock\ absorber\ \ damping\ \ piston \tag{12}$$

Figure 5: Test Vehicle 8” Half Round Test

Equation (12) defines the pressure vs. velocity relationship for the shock absorber. It is parameterized according to equation (11) with the actual relief valve dimensions. Equation (11) was entered into a Simulink lookup table to calculate the force generated by the shock absorber in response to velocity inputs. Calculation and test results for the force vs. velocity of the shock absorber were plotted in figure 4. Good correlation was achieved between the low fidelity shock absorber model and test results. The low fidelity Simulink shock absorber model was also tested with ADAMS and a solution could be calculated using co-simulation.

The disadvantage of the low fidelity model was the hysteresis associated with the bulk modulus of the shock absorber hydraulic fluid and the masses of the relief valve components was not calculated. The second order mass dynamics of the relief and check poppets and the first order pressure dynamics associated with the volumes in the shock absorber caused the hysteresis of the high fidelity model which is shown in figure 3. Figure 4 shows there is no hysteresis with the low fidelity model because the force vs. velocity characteristics of the low fidelity model are defined by a table created from equation (11). The low fidelity model has no first or second order system dynamics. With these observations, the low fidelity and high fidelity shock absorber models were placed inside a 7 DOF Simulink vehicle model to study the influence of shock absorber model fidelity on prediction of driver’s seat acceleration during a half round event.

7 DOF VEHICLE MODEL

One of the requirements for a new vehicle design is to meet driver’s seat acceleration limits on a half round test. During the half round test, a test vehicle is driven over half round cylinders with pre-defined diameters ranging from 2” to 12”. The half round test simulates driving over a log, obstacle or speed bump in the roadway. This test is repeated

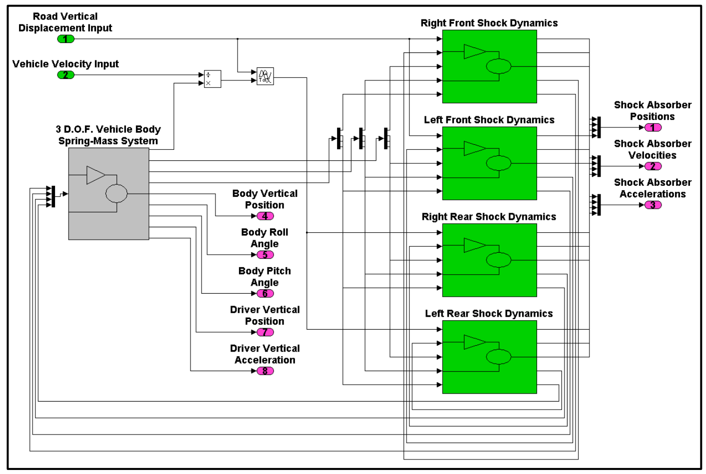

Figure 6: 7 DOF Simulink Vehicle Model

at increasing vehicle velocities until the peak acceleration of the driver’s seat exceeds 2.5 G’s. The velocity at which the vehicle exceeds 2.5 G’s is reported. Figure 5 shows a photograph of the prototype test vehicle approaching the half round.

A 7 DOF model of a test vehicle was created in Simulink to simulate the half round performance of a prototype vehicle. Figure 6 shows the schematic from the 7-DOF Simulink vehicle model. The body or sprung mass of the vehicle was defined to have 3 degrees of freedom, pitch, roll, and vertical motion. The other 4 degrees of freedom were defined as the vertical movement of the four un-sprung masses comprised of the suspension linkage, wheels, springs, shocks, and tires at each of the 4 corners of the vehicle.

Two 7 DOF models were created. One vehicle model was created with the high fidelity shock absorber model at each of the four corners of the vehicle. A second vehicle model was created with the simplified shock absorber models at each corner of the vehicle. Both vehicle models were run in parallel in the same Simulink model to eliminate errors associated with using different model set-ups.

The response of the test vehicle to half round road inputs was calculated. The road input was phased from front to rear to account for the longitudinal spacing of the axles. The lag to the rear axle was calculated by dividing the wheel base by the velocity of the vehicle. All driver's seat acceleration simulation and test results were post-processed using a 30Hz low-pass Butterworth filter to remove high frequency components from the data.

COMPARISON OF SIMULATED AND MEASURED HALF ROUND RESULTS

Half round tests were performed on a test vehicle with all simulation runs recorded on video. The test vehicle was instrumented with an accelerometer under the driver's seat and with front and rear wheel position sensors. Figure 5 shows a picture of the test vehicle approaching an 8 inch half round obstacle at 10mph.

Time domain driver's seat acceleration results are shown in figure 7. Figure 7 includes measured driver's seat acceleration data (30Hz Filt TEST DATA). Figure 7 also includes the simulated driver's seat acceleration from the 7 DOF vehicle model. Low fidelity (SIMPLE MODEL) and high fidelity (ADVANCED MODEL) shock absorbers simulation results were overlaid on top of the measured driver's seat acceleration.

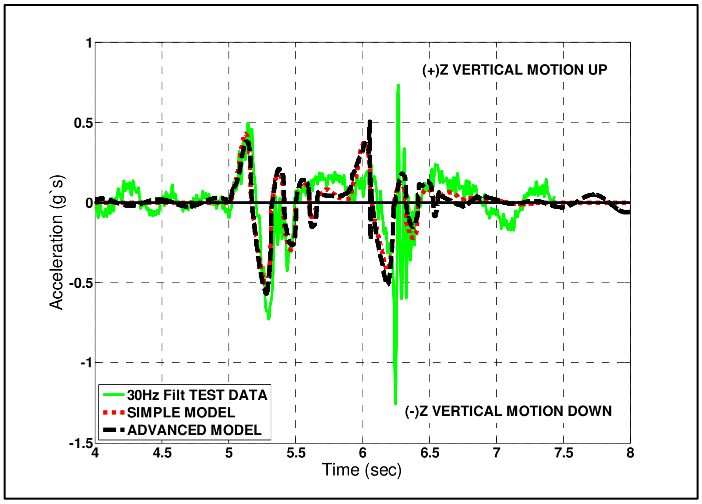

Figure 7: Measured vs. Simulated Driver's Seat Acceleration for 8" Half Round at 10mph

Figure 7 shows good correlation between the low fidelity and high fidelity shock absorber models for calculated driver's seat acceleration. Eliminating hysteresis associated with the bulk modulus of the fluid in the shock absorber had a minor influence on simulated driver's seat acceleration for the half round event. Although this was true for the half round event, additional testing would need to be completed to understand the influence of the low fidelity model on simulating fast repeating events such as an RMS course simulation or a rumble strip simulation.

When comparing the driver's seat acceleration test data to simulation results in Figure 7, the measured peak acceleration of the driver's seat was significantly higher than the simulated peak acceleration. The simulated peak acceleration response of the driver's seat was 0.6 g's. The measured response of the driver's seat was 1.3 g's. The simulated response was 54% low for prediction peak driver's seat acceleration response.

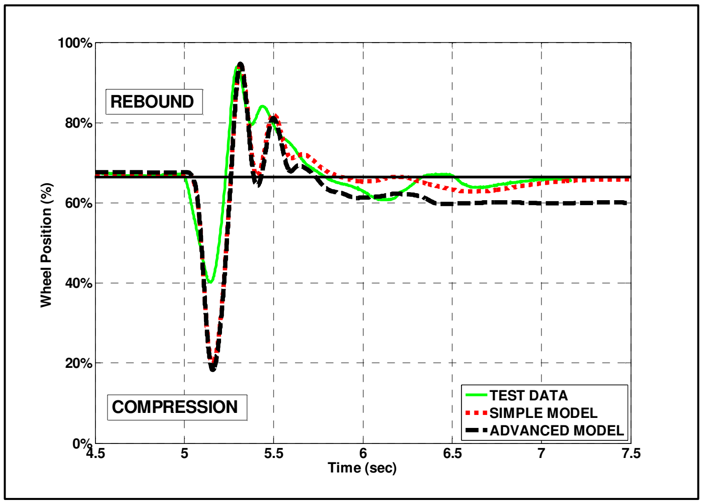

Figure 8: Measured vs. Simulated Front Wheel Vertical Motion for 8" Half Round at 10mph

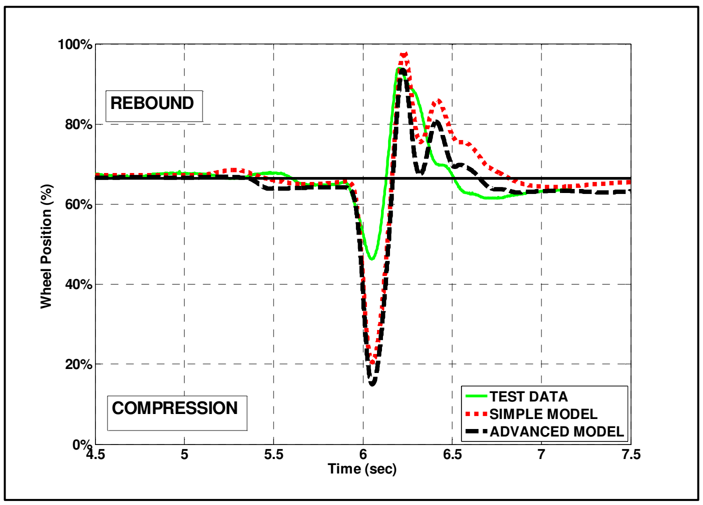

Figure 9: Measured vs. Simulated Rear Wheel Vertical Motion for 8" Half Round at 10mph

One explanation for the difference between calculated and measured drivers seat response is the measured acceleration response of the driver's seat shows a short duration(less than 100ms) peak acceleration event at 6.3 seconds. It is possible that the rear suspension on the test vehicle hit the rebound hard stop during the test, causing the acceleration spike. The response of the vehicle to impacting a hard stop was not simulated in detail in the 7 DOF Simulink model. Hard stops were used in the model, but a spike in acceleration similar to the test results could not be replicated in the simulation models.

Figure 7 also shows good time domain correlation between simulated and measured driver's seat acceleration. The timing of peaks and valleys for measured and simulated driver's seat acceleration is similar. The time domain correlation indicates that the stiffness and damping in the simulation model accurately model the prototype test vehicle.

Figures 8 and 9 show a comparison of the measured and simulated time domain response of the front and rear wheels to the 8" half round road input at 10mph. Wheel position was measured using string pots mounted to the test vehicle frame and attached near the wheel spindles. The timing of the simulation events shows good correlation to the measured time domain response of the wheels.

Figures 8 and 9 show a significant amount of error between simulation and test results during the front suspension compression event at 5.2 seconds and rear compression event at 6.1 seconds. This is the time at which the tire strikes the half round and suspension motion is initiated. Static friction of suspension components, non-linear tire spring rate behavior, and the dynamic response of the string potentiometer are all variables that could influence the suspension travel. Also, the jounce bumpers are not fully modeled in Simulink and have an influence on wheel travel. Another theory is that the simulation model is dissipating the same amount of energy as the vehicle, but the distribution of energy dissipation between the tire and the shock absorber is different for simulation and test. Additional work needs to be done to study the source for this error in wheel travel.

CONCLUSIONS

The low fidelity shock absorber model was selected to calculate the force vs. velocity behavior of a shock absorber. The force vs. velocity curve from the low fidelity shock absorber model is parameterized according to the actual dimensions of the relief valve hardware inside the shock absorber damping piston. This makes it possible to optimize the damping piston by changing actual damping valve dimensions in the simulation model. Test hardware can be fabricated based on the final optimized shock absorber parameters derived from simulation results.

For the half round event, the low fidelity shock absorber model provided nearly identical results to the high fidelity shock absorber model when comparing the acceleration response of the driver's seat as shown in figure 7. Wheel travel was also calculated with a similar response for the high fidelity and low fidelity models as shown in figures 8 and 9. The low fidelity shock absorber model could also be run with ADAMS in co-simulation of half round events.

The timing of acceleration and wheel travel events was accurately modeled in simulation when compared to test results. There was significant error in the magnitude of simulated driver's seat peak acceleration and wheel travel when compared to test results. The measured peak acceleration of the driver's seat for the 8" 10 mph half round was 1.3 g's. The calculated peak response was 0.6 g's, which is 54% error between simulation and test. Other than the one spike at 6.3 seconds, the simulation model provided a good estimate of driver's seat acceleration.

The 7 DOF model did not accurately predict the short duration high magnitude acceleration spike at 6.3 seconds, most likely caused by the suspension impacting the jounce and rebound bumpers. Additional work needs to be done to model the response of the vehicle response to suspension hard stop impacts.

REFERENCES

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