Modeling, Testing, and Validation of the 2007 Chevy Silverado Finite Element Model

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This working paper summarizes recent efforts and findings derived from NCAC research. It is intended to solicit feedback on the approach, scenarios analyzed, findings, interpretations, and implications for practice reported by the research team. The statements contained herein do not necessarily reflect the views or policy of the FHWA. Please forward comments or questions to the authors noted above. These efforts will ultimately be documented and made available to advance research efforts related to this topic and guidance for practice.

ABSTRACT

The National Crash Analysis Center (NCAC) at the George Washington University (GWU) has been developing and maintaining a public domain library of finite element (FE) vehicle models for use in transportation safety research. Using the NCAC’s unique and innovative reverse-engineering process, an FE model of the 2007 Chevrolet Silverado pick-up truck was developed. This pick-up truck satisfies the requirements for a 2270P test vehicle under the soon to be adopted crashworthiness evaluation criteria specified in the Manual for Assessing Safety Hardware (MASH). Since this FE model will be extensively used to design safer roadside barriers, the representation of the suspension system and its dynamic response becomes a critical factor influencing the performance of the roadside barrier. To improve the FE model fidelity and applicability to the roadside hardware impact scenarios it is important to validate and verify the model to a multitude of component and full-scale tests. A series of highly instrumented, non-destructive, full-scale tests and destructive, component-level suspension tests were conducted to gather data for validating the suspension system of the FE model. This paper provides a description of the vehicle FE model, the various component and full-scale tests that were performed, and the current status of the model validation to these physical tests.
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INTRODUCTION

Full-scale physical crash tests are conducted by several agencies worldwide to evaluate vehicle and roadside hardware crashworthiness. It is not economically feasible to perform full-scale crash tests on the wide range of parameters that influences safety performance. With the advent of high-speed, high-memory capacity computers in the early 1990s, computer technology has reached the point where vehicle crash outcomes can be accurately predicted using the computer. Specialized FE software – appropriate for depicting the physics of motor vehicle collisions – is now used extensively to predict collision outcomes. Because of this new technology, impact simulations utilizing nonlinear FE analysis have become effective tools in designing and evaluating crashworthy vehicles and roadside safety hardware among other things. Once successfully validated, the FE models can be linked to analyze new impact scenarios (e.g., incremental changes in design parameters, a range of impact conditions).

As an update to the National Cooperative Highway Research Program’s (NCHRP) Report 350 [1], new test vehicles have been designated to be representative of the current fleet in the Manual for Assessing Safety Hardware (MASH). One of the test vehicles has been upgraded from a pick-up truck weighing 2000 kg to a pick-up truck weighing 2270 kg. Since the 2007 Chevy Silverado meets the requirements of the new pick-up truck test vehicle, an FE model representing this vehicle has been developed using the NCAC’s reverse-engineering process.

The three main requirements of a highly viable FE model are accuracy, robustness, and processing speed. The FE model should be accurate enough to yield reasonable predictions of the essential features being sought, robust enough to successfully simulate many impact scenarios, and fast enough to allow for many iterations and parameter studies. Verifying and validating the FE model to a range of component and full-scale impact tests is necessary to ensure adequate accuracy of the model.

FE MODEL DESCRIPTION

The NCAC has been developing vehicle and roadside hardware FE models for over 15 years. Over the years, the reverse-engineering methodology has evolved and a unique process for vehicle model development has been developed. The reverse-engineering process involves methodical disassembly of the physical vehicle to create a computer model of each and every part in the vehicle. The geometrical data for each part is converted to an FE mesh and carefully re-assembled with the appropriate consideration of connections and constraints between elements to create a full vehicle FE model. A series of material level characterization tests, on coupons extracted from various locations of the vehicle, are performed to gather the required input for the material models in the FE program. The fully assembled model of the 2007 Chevy Silverado pick-up truck consisting of 930,000 elements is shown in Figure 1.

Since this vehicle FE model will be extensively used in the analysis and improvement of roadside safety features, special emphasis was given to accurate representation of the suspension components and its connections. For the front suspension system (Figure 2a), the upper and lower control arms, the coil spring and damper are explicitly modeled. Their connections to the wheel spindle are modeled with appropriate joint degrees of freedom so that it would function as in the physical system. For the rear suspension system (Figure 2b), the individual leaf springs with varying thickness are explicitly modeled. The leaf springs are connected to the rear axle using the U-bolts modeled as beam elements. The ends of the leaf springs are connected to the truck frame with the appropriate joint degrees of freedom.
MODEL VALIDATION

The Silverado model was subjected to a more extensive validation than had previously been applied to NCAC vehicle models. This multi-stage validation effort included:

1. Detailed measurement of inertial properties
2. Suspension system component tests
3. Non-destructive bump and terrain tests
4. Comparisons to New Car Assessment Program (NCAP) full frontal rigid wall impact tests
5. Comparisons to recent crash tests for six common roadside barriers.

This paper describes the current validation results associated with items 1, 2, and 3 above. All of these tests [2] [3] were designed and developed to gather the required data for model validation. The majority of these tests focused on the suspension system characterization since it plays a critical role in predicting the overall response of the vehicle and performance of the roadside safety barrier [4] [5]. In addition to the impact tests, the physical vehicle’s center of gravity location and the roll, pitch, and yaw moments of inertia were measured for use in model validation.

1. Mass, Inertia, and CG Comparisons

The FE model was validated to a series of component and full-scale impact tests to ensure accurate, predictable response. However, the first step in the validation process was to ensure that the FE model has the correct mass distribution and inertia measurements compared to the physical vehicle. The Chevy Silverado’s center of gravity height and roll, pitch, and yaw moments of inertia were not available in the open literature. These measurements play a critical role in predicting the overall response of the vehicle when impacting roadside safety hardware at oblique impact angles.
Before the reverse-engineering process, the Chevy Silverado pick-up truck was sent to a specialized laboratory (SEA Limited) that could measure the center of gravity height and roll, pitch, and yaw moments of inertia. The center of gravity height and the three moments of inertia of the completed FE model was compared to the laboratory test results (Table 1). The inertia test of the physical vehicle was conducted with a full tank of gas. In order to meet the weight requirements of the MASH test vehicle, the fuel mass in the FE model was removed. The difference in measurements between the physical vehicle and the FE model was within 3%, ensuring that the total mass and mass distribution closely matched that of the physical vehicle.

<table>
<thead>
<tr>
<th></th>
<th>Physical Vehicle</th>
<th>FE Model</th>
<th>% difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight, kg</td>
<td>2337</td>
<td>2270</td>
<td>2.86</td>
</tr>
<tr>
<td>Pitch inertia, kg-m^2</td>
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<td>6028</td>
<td>2.06</td>
</tr>
<tr>
<td>Yaw inertia, kg-m^2</td>
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<td>6490</td>
<td>0.57</td>
</tr>
<tr>
<td>Roll inertia, kg-m^2</td>
<td>1051</td>
<td>1050</td>
<td>0.09</td>
</tr>
<tr>
<td>Vehicle CG height, in</td>
<td>27.96</td>
<td>28.64</td>
<td>2.32</td>
</tr>
</tbody>
</table>

Based upon the small differences between the actual vehicle and FE model for the mass, inertial properties, and center of gravity measures, it was concluded that the model is a valid representation of the actual vehicle.

2. Pendulum Impact Tests

A series of pendulum impact tests were conducted on the front and rear suspension assemblies of the 2007 Chevy Silverado after the reverse-engineering was complete. The objective of these tests was to gather dynamic suspension deflection data (bottoming out) for validating the 2007 Chevy Silverado FE model to improve its range of applicability in evaluating roadside safety features. The suspension and its connections to the frame rail were preserved during the vehicle reverse-engineering process. The frame rails were then cut in the center to separate the front and rear suspensions, so that the tests could be done independently.

The 2007 Silverado suspension assembly was tested in the following three different impact configurations
   a. Front suspension vertical loading
   b. Rear suspension vertical loading
   c. Rear suspension lateral loading

The impactor used in these tests was a 2000 kg pendulum mass suspended by four cables to a steel framework. The pendulum is released to be free falling from a predetermined height based to achieve a desired impact velocity. The tests were conducted with the pendulum located at the FHWA Federal Outdoor Impact Lab (FOIL) in McLean, Virginia.

a. Front Suspension Vertical Loading

The front suspension assembly along with the frame rails was firmly attached to a specially fabricated support structure to prevent any movement during the pendulum impact (Figure 3a). The structure was positioned such that the pendulum impacted the bottom of the front tire to induce vertical loads in the suspension assembly. The tire pressure was set to 35 psi for all tests. Three different impact speeds (3, 6 and 9 km/h) were chosen for this test series. The impact velocity was gradually increased to get full compression of the suspension assembly.
The pendulum and the front suspension assembly were instrumented with accelerometers and string pot potentiometers at several locations. Specific locations of the accelerometers and string pot potentiometers at the front suspension are shown in Figure 3b. A series of high speed digital cameras were used to capture the test event. Each of these cameras captured the test event at 500 frames per second. Time zero for data acquisition and high speed imaging was initiated when the pendulum impacted the contact switch on the tire surface.

The front suspension sub-system along with the frame rails was extracted from the full vehicle FE model of the Chevy Silverado. The frame rails were constrained as in the test set-up. The front tire was impacted with a rigid pendulum at representative impact velocities. The initial model response did not compare well with the physical test. This was expected since the model was assembled based on past experience. Component level test data were not available at that time for model verification and validation.

The suspension system model was updated as part of the validation process. Each individual part of the suspension assembly and its connections were rechecked for accuracy. The lower A-arm was originally modeled using shell elements. In order to represent the correct mass, stiffness, and inertia the lower A-arm was re-meshed using solid elements. Quasi-static uniaxial compression tests were conducted on the coil spring to investigate its stiffness. Subsequently, the coil spring had to be re-meshed from shell to solid elements to match the stiffness measured in the compression test. A revolute joint was added to represent the rotational degree of freedom provided by the pin that connects the shock damper to the lower A-arm. A nonlinear damping curve was used to represent the characteristics of the shock damper. This damping curve was optimized iteratively to match the suspension system response observed in the physical tests. The simulation set-up and the comparison of the time-history response of the front suspension dynamic deflection are shown in Figures 4a & 4b. All of the above
updates to the FE model improved the suspension system response and a good correlation was observed with respect to the physical test. Peak dynamic deflection of 80 mm was measured at a pendulum impact velocity of 9 km/h.

b. Rear Suspension Vertical Loading

Similar to the front suspension vertical tests, the rear suspension assembly along with the frame rails was firmly attached to a specially fabricated support structure to prevent any movement during the pendulum impact (Figure 5a). The structure was positioned such that the pendulum impacted the bottom of the tire to induce vertical loads in the suspension assembly. The tire pressure was set to 35 psi for all tests. A range of impact speeds from 2 to 6 km/h was chosen for this test series. The impact velocity was gradually increased to get full compression of the suspension assembly. The tests were conducted with and without the shock dampers to isolate its effect on the overall suspension response. Specific locations of the accelerometers and string pot potentiometers on the rear suspension are shown in Figure 5b.

![Rear Suspension Vertical Loading Support Structure](image)

![Schematic of Rear Suspension Instrumentation Locations](image)

Figure 5: Rear Suspension Pendulum Test Setup

The rear suspension sub-system along with the frame rails was extracted from the full vehicle FE model of the Chevy Silverado and constrained as in the physical test. The rear tire was impacted with a rigid pendulum at the impact velocities derived from the tests. The simulation set-up is shown in Figure 6. The FE model was iteratively improved to match the rear suspension system response as to the physical tests.

![Rear Suspension Vertical Loading Simulation Setup](image)

Figure 6: Rear Suspension Vertical Loading Simulation Setup
One of the critical factors influencing the response of the rear suspension system was the leaf springs. The leaf spring consists of three leaves which are held together by U-bolts. The upper and center leaves have a constant thickness while the lower leaf has a varying thickness. The lower leaf is considerably thicker at the center compared to the ends. Hence, the lower leaf spring was separated into 3 separate parts for representation in the FE model. A quasi-static compression test was conducted on the rear leaf spring assembly to measure its stiffness and validate the FE model of the leaf springs. The schematic of the quasi-static test, FE simulation of the leaf spring compression and the time-history comparison between the test and simulation is shown in Figures 7a, 7b, and 7c.

FE simulations of the rear suspension system with and without the shock dampers were performed at pendulum impact velocities derived from the physical tests. A nonlinear damping curve was used to represent the characteristics of the shock damper. This damping curve was optimized iteratively to match the suspension system response observed in the physical tests. The comparisons of the time-history response of the rear
suspension dynamic deflection with and without the shock dampers are shown in Figure 8a & 8b. Accurate representation of the leaf springs and the end stopper in the FE model improved the suspension system response and a good correlation was observed with respect to the physical tests. The rear end stop on the frame rail was modeled as a discrete spring. The force-displacement function for the discrete spring was obtained from a quasi-static test on the rear end stop. Peak dynamic deflection of 240 mm without shock dampers and 206 mm with shock dampers was measured at a pendulum impact velocity of 5.8 km/h.

c. Rear Suspension Lateral Loading

The lateral impact of the rear suspension is common in oblique impacts into roadside safety hardware. When the front of the vehicle is redirected by the safety barrier, the rear of the vehicle side slaps the barrier. In order to improve the predictive capability in these kinds of impacts, the vehicle FE model should be sufficiently validated to the lateral impact modes.

The rear suspension assembly along with the frame rails was firmly attached to a specially fabricated support structure as shown in Figure 9. The structure was positioned such that the pendulum impacted the tire side to induce lateral loads in the rear suspension assembly. The tire pressure was set to 35 psi for all tests. Two different impact velocities (3.0 km/h and 5.9 km/h) were chosen for this test series. All lateral impact tests were conducted with shock dampers in position. The pendulum and rear suspension assembly were instrumented with accelerometers and string pot potentiometers (to measure lateral deflection).

![Figure 9: Rear Suspension Lateral Loading Support Structure](image)

The FE model of the rear suspension sub-system and the frame rails was constrained as in the physical test. The rear tire was impacted with a rigid pendulum at the impact velocities derived from the tests. The simulation setup and the time-history comparison between the test and simulation are shown in Figures 10a and 10b. The FE model was iteratively improved to optimize the rear suspension system response to the physical tests. At 5.9 km/h impact velocity, the rear suspension reaches a maximum deflection of 70 mm in the test. The loading part compares well between the test and simulation. However, the unloading appears to happen faster in the simulation compared to the test, indicating that the FE model is slightly stiffer compared to the suspension system in the vehicle. The lateral deflection in this impact mode is primarily due to the compliance in the rubber parts at the pivot points of the leaf springs to the frame rail. The rubber parts are molded together with steel inserts. Additional work is required to accurately model these rubber mounts.

After the front and rear suspension vertical loading and rear suspension lateral loading tests and revisions or refinements to the model, it was concluded that the representation of these components in the FE model was sufficiently accurate.
3. Low-Speed, Non-Destructive Full-Scale Tests

The low speed non-destructive full-scale tests were conducted to measure suspension deflection when the vehicle traverses speed bumps and sloped terrains. These tests were conducted prior to vehicle tear down and reverse-engineering for FE model development. The test article used in this series of tests was a commercially available speed bump made of recycled plastic. The specific dimensions of the speed bumps are shown in Figure 11a. In pilot tests, minimal suspension deflections were observed when the test vehicle traveled over the commercially available speed bump at approximately 16 km/h. In order to induce additional deflection in the front and rear suspension, a standard 2” x 12” wood plank was added underneath the speed bump and the edges were beveled to create a smooth transition (Figure 11b). The height of the speed bump in this configuration was 3.5 inches.

The speed bumps were positioned in the following three different configurations (Figure 12).

  a. Single speed bump on passenger side only.
  b. Speed bumps on both driver and passenger sides.
  c. Two speed bumps separated by a distance of 1 meter on passenger side only.

The test vehicle was propelled using the Federal Outdoor Impact Laboratory (FOIL) propulsion system. A short cable attached to the tow dolly and both ends of the truck’s front axle, near the tires, was used to pull the truck forward. The targeted vehicle speed prior to contacting the speed bumps for all three configurations was 16 km/h ± 0.5 km/h.

The vehicle was instrumented with accelerometers and string pot potentiometers similar to the pendulum impact tests. The accelerometers were used to measure the vehicle’s acceleration and the string pot potentiometers were used to measure the vehicle’s displacement during the test. A series of high speed digital cameras were used to capture the test event. Each of these cameras captured the test event at 500 frames per second.
Full-scale FE simulations were conducted based on the above test series. Prior to running the full-scale simulations, the vehicle model was stabilized to be in equilibrium at the beginning of the simulation. This was achieved by adding discrete springs at the four suspension locations. The force-displacement response of the discrete springs was optimized so as the vehicle settles to equilibrium position under gravity loading. To check the equilibrium position, simulations were performed by dropping the vehicle onto a rigid ground under gravity loading. Figure 13 shows the force time-history measured on the ground when the vehicle is stabilizing. By 250 ms the vehicle reaches equilibrium, and the measured force is equal to the product of the mass of the vehicle and the acceleration due to gravity.

Figure 13: Equilibrium Check under Gravity Loading

The simulation set-up for speed bump test configuration 1 (single speed bump on passenger side only) and the front and rear suspension deflection time-history data is shown in Figures 14a and 14b. The maximum front right suspension deflection measured in the full-scale test was 22 mm during loading and unloading. The front suspension deflection-time response resembled a sinusoidal curve until the front right tire touched the ground after unloading. The elasticity in the tire caused bouncing which resulted in uneven unloading between 0.45 sec
and 0.65 sec. The amplitude measured in the FE simulation at the front suspension was 28 mm, slightly higher than that of the test, indicating that the front suspension is softer in the simulation compared to the test. The timing of the loading and unloading peaks matched well with the test. The rear suspension deflection-time response also resembled a sinusoidal curve. The maximum rear right suspension deflection observed was 46 mm during loading and 58 mm during unloading. The rear suspension system response in the FE simulation closely follows the test response with an amplitude of 52 mm during loading and 68 mm during unloading.

Similar observations were made for speed bump test configuration 2. The simulation set-up for speed bump test configuration 2 (speed bumps on both driver and passenger side) and the front and rear suspension deflection time-history data is shown in Figures 15a, 15b, and 15c. The maximum front right and left suspension deflection measured in the full-scale test was about 25 mm during loading and unloading. As observed in speed bump configuration 1, the front suspension in the FE simulation resulted in a softer response, primarily during unloading. The rear suspension system response in the FE simulation closely follows the test response, both in phase and amplitude.
Figure 15: Speed Bump Configuration 2 Simulation

The simulation set-up for speed bump test configuration 3 (two speed bumps separated by a distance of 1 meter on passenger side only) and the front and rear suspension deflection time-history data is shown in Figures 16a and
16b. Though the overall trends observed in the test and simulation were similar, the suspension response drifted after traversing the first speed bump.

The comparisons of the model response to the test results for the non-destructive bump tests showed good correlations. These provide additional confidence that the model can provide a valid representation of bump response. Further improvements may be possible with refinements to the model. These will be considered upon the completion of the sloped terrain test comparisons. Several of these were undertaken by traversing the test vehicle along a 6:1 sloped terrain at an approach angle of 25 degrees. The FE model is currently being verified to these tests and the suspension system will be further optimized to improve correlation with the physical tests.

**SUMMARY**

The vehicle steering and suspension system plays a critical role during oblique impacts into roadside median barriers. Simulation models used to predict crash outcomes in these impact scenarios should be accurate enough to capture the physics of the suspension deflection and failures. In order to validate the suspension systems of the new 2007 Chevrolet Silverado FE model, a series of component and full-scale tests were conducted.
The first series of tests were non-destructive low-speed full-scale tests and were conducted prior to vehicle tear down for FE model development. The vehicle was heavily instrumented and traversed over speed bumps and 6:1 sloped terrains. The suspension deflections at each of the four locations were measured in addition to vehicle linear and rotational accelerations.

The second series of tests were conducted after the FE model development was complete. In this test series, the suspension sub-system was subjected to impact loads from a swinging rigid pendulum mass of 2000 kg. The front suspension was subjected to vertical loading at gradually increasing impact velocities to gather data for the full range of suspension deflection. Similarly, the rear suspension was subjected to vertical and lateral loading. The vertical loading for the rear suspension was conducted with and without the shock dampers to characterize the effect of shock damper on rate of suspension deflection.

This series of test data has been extremely useful for validating the FE model to incrementally improve the predictive confidence in roadside barrier impacts. With the successful comparisons of the Silverado FE model to NCAP tests results and the results presented here, there is growing confidence that this is an accurate and robust model. The results of the current FE model validation efforts will be documented when complete, but all indications are that this model will provide a sound basis for many types of crash simulation applications in the future.

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REFERENCES