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DEVELOPMENT & VALIDATION OF A GENERIC 3-D VEHICLE MODEL & GUI FOR STUDYING THE HANDLING PERFORMANCE

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ABSTRACT

The evolving capability of modern simulation packages continues to facilitate the use of mathematical models to virtually prototype vehicles. A generic 3-D vehicle model with a simple representation of suspension kinematics that can represent the behavior of most independent suspensions and solid axle suspensions is presented. This paper provides the technical background for the model and the graphical user interface built within the ADAMS/View software environment. Applications of the model and GUI may include vehicle design, virtual prototyping and crash simulation.

The performance of the 3D model is compared with the results obtained from CARSIM Educational Version [3]. Also, comparison results between the model and data from National Advanced Driving Simulator (NADS) [6,7] for a 1998 Chevrolet Malibu are presented. The results show that mathematical model captures the essentials of the real world performance of a vehicle and that MSC.ADAMS is a powerful simulation package to virtually prototype vehicles.

Keywords: ADAMS/View, Vehicle Dynamics, Simulation, Multi-body Dynamics, Virtual Prototyping, Vehicle Design, and Vehicle Handling Performance

1. INTRODUCTION

As product complexity increases and competitive pressures reduce the time available for the product development, hardware prototyping becomes increasingly important but less feasibly to accomplish. As an alternative, virtual prototyping, using simulations of mathematical models of the proposed product for testing, has become

increasingly attractive. These techniques not only reduce the costs and time associated with hardware prototype, they also help the designer experiment with innovative design variations earlier in the product design cycle when there is still time to incorporate major changes.

One of the primary tools for virtual prototyping of products, in which dynamics are important, is the numerical simulation of dynamic system models representing the design. Many commercial software packages such as MSC.ADAMS [4], AutoSim [5], etc. are used as they offer features to aid in model description, model development, equation formulation, numerical integration. In recent years, large size models are becoming the common approach for studying the system behavior as the software packages are getting more and more sophisticated. These types of models can provide the required predictions and information about the system performance. However, large size models are described by hundreds of parameters and finding a relation between the model parameters and system behavior is a very difficult process. Besides, the large number of parameters increases the model development and run times significantly. The objective of this paper is to present a 3D vehicle dynamics model including a simple suspension system (as a representative of the suspension kinematics) to study the handling performance of vehicles. The development of a graphical user interface to facilitate the change of the design parameters is also described. The model response is compared to that of CARSIM Educational Version and also some data taken from publications in the literature [6,7].

The paper is organized as follows: first, the 3 D vehicle model and graphical user interface are presented in Section 2. Next, the performance of the model is presented in two case studies in Section 3. Finally, conclusions are given in Section 4.

2. MODEL DESCRIPTION

The model is based on a rigid body that represents the main body of the vehicle and has six DOF. The model includes a sprung mass with six degrees of freedom (x, y, z, roll, pitch, yaw). An additional four bodies are added, each with a single translational DOF, to account for the vertical movements allowed by the suspensions for independent suspension (for solid axle suspensions, the degrees of freedom are axle roll and jounce). The wheel bodies are positioned such that the origins of their local coordinate systems are

nominally at the locations of the centers of tire contact (Figure 1). Longitudinally, the origins of the front and rear wheels are separated by the vehicle wheelbase. Laterally, they are separated by the vehicle front and rear track widths.

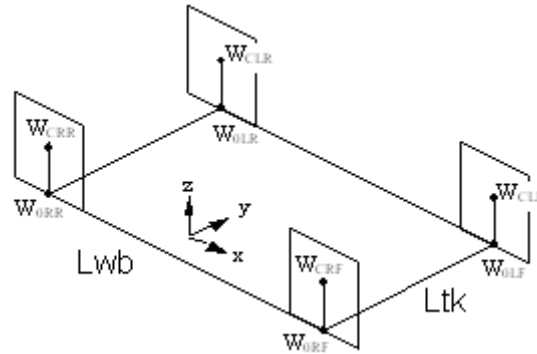


Figure 1: Descriptions of the rigid bodies in the model

2.1. Suspension Model

User has the ability to choose from independent and solid axle suspension [1,2,10] for the front/rear axle. An independent suspension is one in which vertical movement of one wheel does not cause noticeable movement of the other wheel if the anti-roll bar is disconnected. A schematic model for independent suspension is shown in Figure 2 (left). Solid axle suspension has an actual axle or linkage system that causes both wheels to roll together. A schematic model for a solid axle is presented in Figure 2 (right). The axle is modeled as a rigid body that has two degrees of freedom relative to the sprung mass of the vehicle unit. One is jounce and the other is roll. In the model, the axle rolls about a point called the roll center, and then translates vertically relative to that hinge.

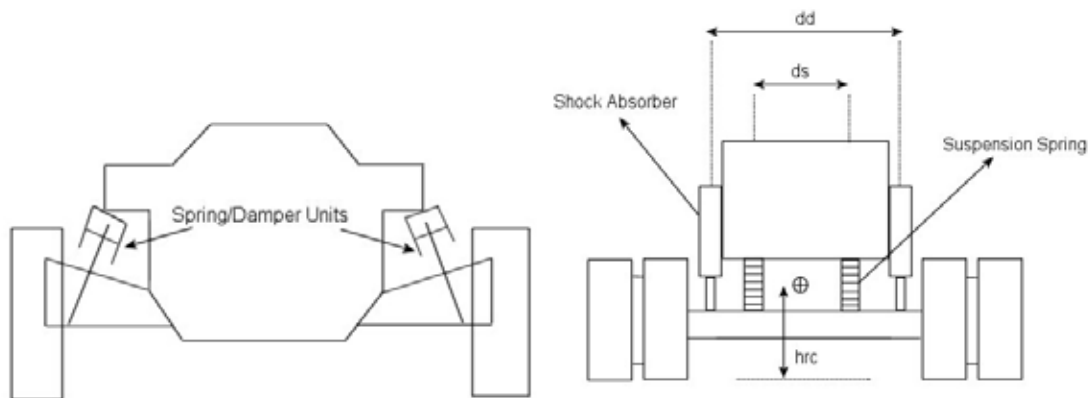


Figure 2: Schematic model for independent (left) and solid axle suspension (right)

The model uses two tables of force deflection data to describe the behavior of the spring when being compressed and when being relaxed. Bump stops are incorporated in the spring model by an increase in spring rate at the deflection at which the bump stops are encountered. A sample plot for suspension spring force versus suspension deflection is shown in Figure 3 (left).

Shock absorbers are dampers that produce a force resisting motion in the suspension. The force is dependent on velocity, but may vary nonlinearly with velocity because of the orifices and blow-off valves that are used in shock absorbers. A sample plot for shock force versus shock compression rate is shown in Figure 3 (right).

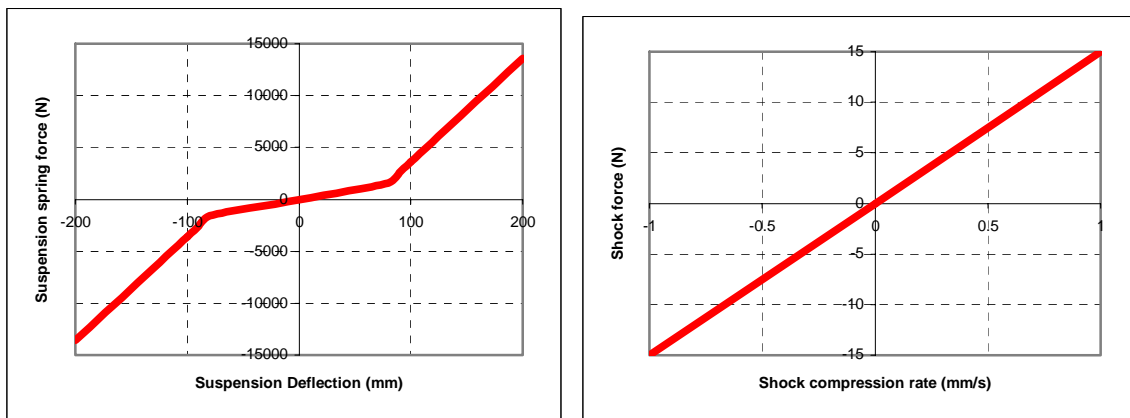


Figure 3: Suspension spring force versus suspension deflection

Axle roll is resisted in the model by the vertical suspension springs and also by a torsional spring that accounts for the difference between roll moment as predicted by the spring effects alone and the roll moment that is measured with a laboratory test rig. Additional roll stiffness can be produced in the suspension by mechanisms such as stabilizer bars. Specifications of auxiliary roll stiffness allow you to replicate the influences of these other mechanisms. The auxiliary roll moment is a tabular function of both the roll of the axle relative to the sprung mass and the static load (at the ground) supported by the wheels. A sample plot for auxiliary roll moment versus suspension roll is shown in Figure 4.

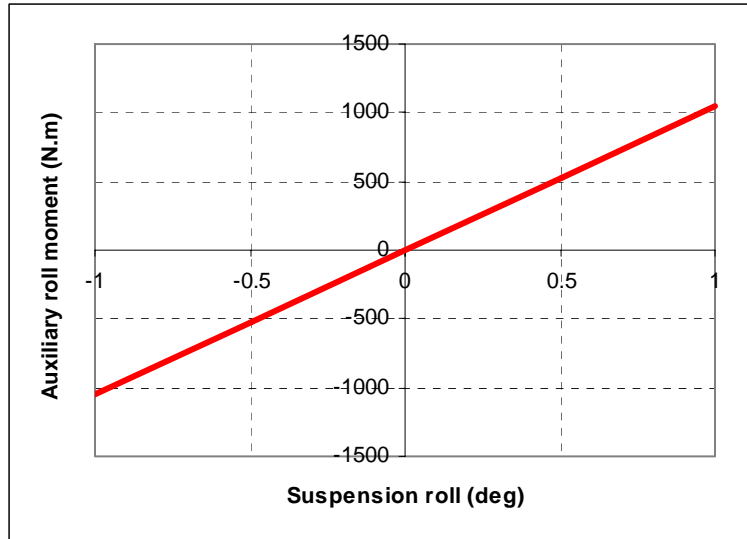


Figure 4: Auxiliary roll moment versus suspension roll

Wheel camber is due in part to a nonlinear relationship to suspension travel. In the model, this relationship is represented with a table to define how wheel camber changes with suspension movement in an independent suspension. Camber angle is positive when the wheels lean out. A sample camber angle versus suspension jounce plot is shown in Figure 5.

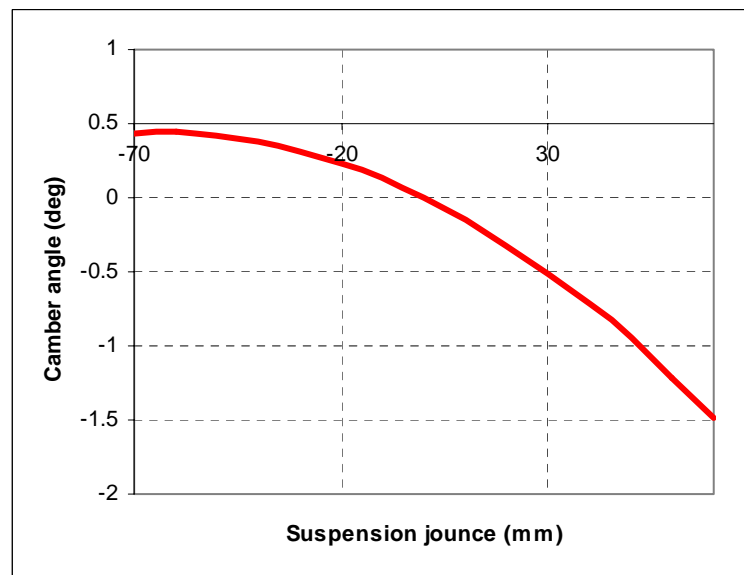


Figure 5: Camber angle versus suspension jounce

2.2. Tire Model

When a tire generates lateral force by slipping sideways with respect to the direction it is pointed, the angle between the wheel heading and the tire velocity vector is known as slip angle. The lateral force is strongly dependent on the slip angle and vertical tire load. The tire model assumes that the tire behavior is symmetric about the origin with respect to lateral slip. A sample lateral force as a function lateral slip and vertical tire load is shown in Figure 6 (left).

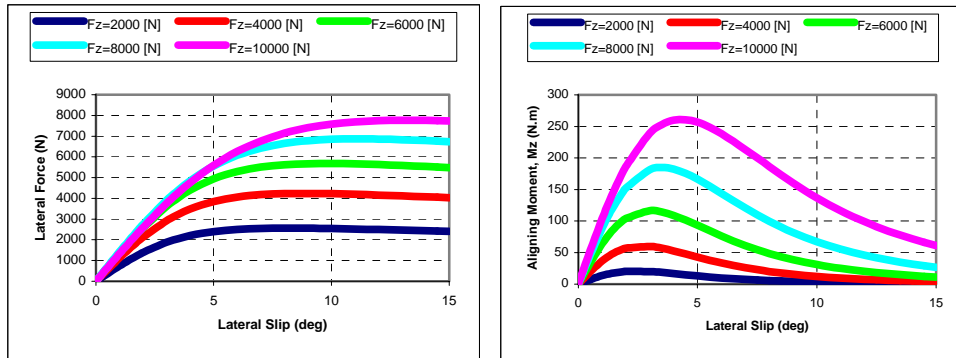


Figure 6: Tire force (left) & aligning moment (right) versus slip angle and vertical load

When a tire assumes a slip angle and develops a lateral force, the resultant force is generated toward the rear of the contact patch. It is standard practice to measure tire forces and moments at the center of the contact patch. When the lateral tire force is resolved to this point, a moment, known as the aligning moment, results. The aligning moment is nominally related to the lateral slip angle and the vertical load. A sample aligning moment as a function lateral slip and vertical tire load is shown in Figure 6 (right).

The GUI developed in ADAMS/View is shown in Figure 7. Vehicle parameters toolbox allow the user to enter the parameter for a specific vehicle, while vehicle model toolbox allows the user to select the suspension type (independent or solid axle) on each axle.

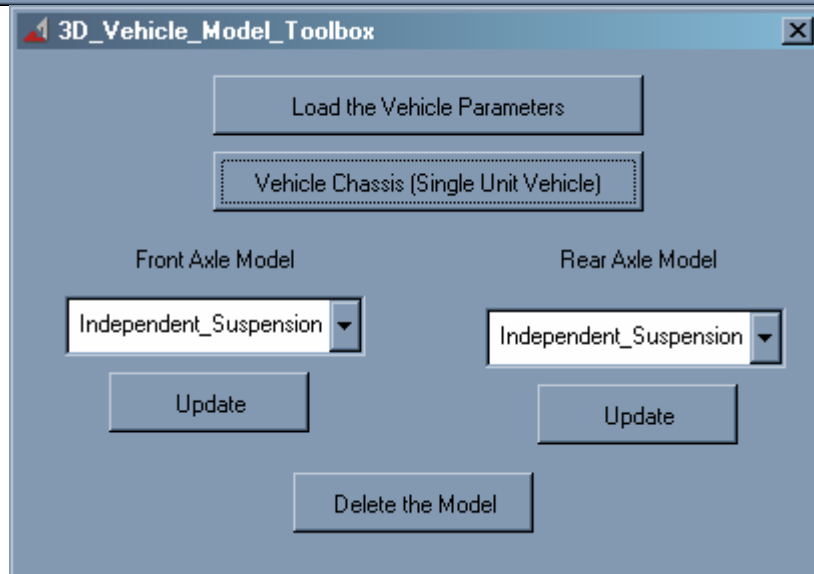
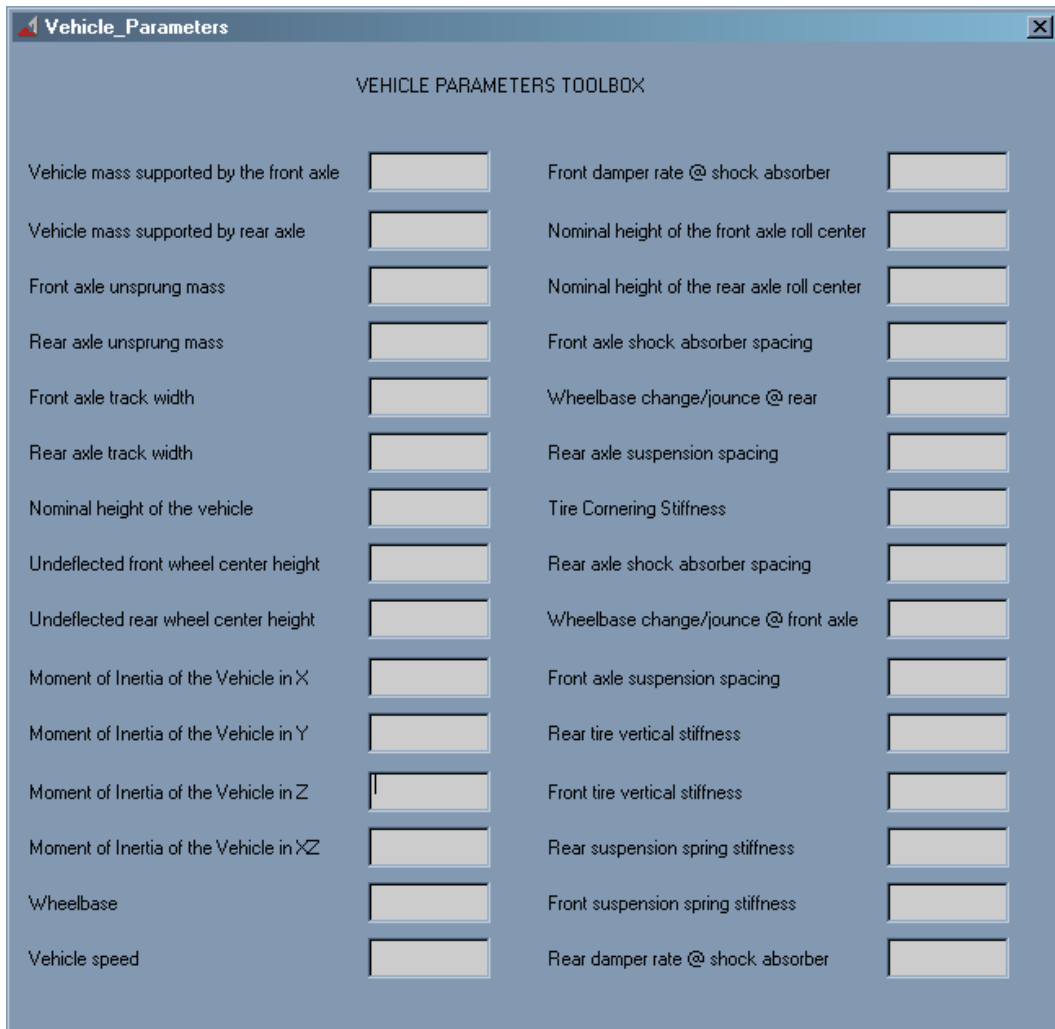


Figure 7: GUI - Vehicle Parameters and Model Toolboxes in ADAMS/View

3. CASE STUDY

3.1. Comparison: MSC.ADAMS model and CarSim Educational

The model developed in MSC.ADAMS is tested against CARSIM Educational under 30-degree step steer and vehicle speed of 100 kph. The comparison results are shown in Figure 8 to Figure 12.

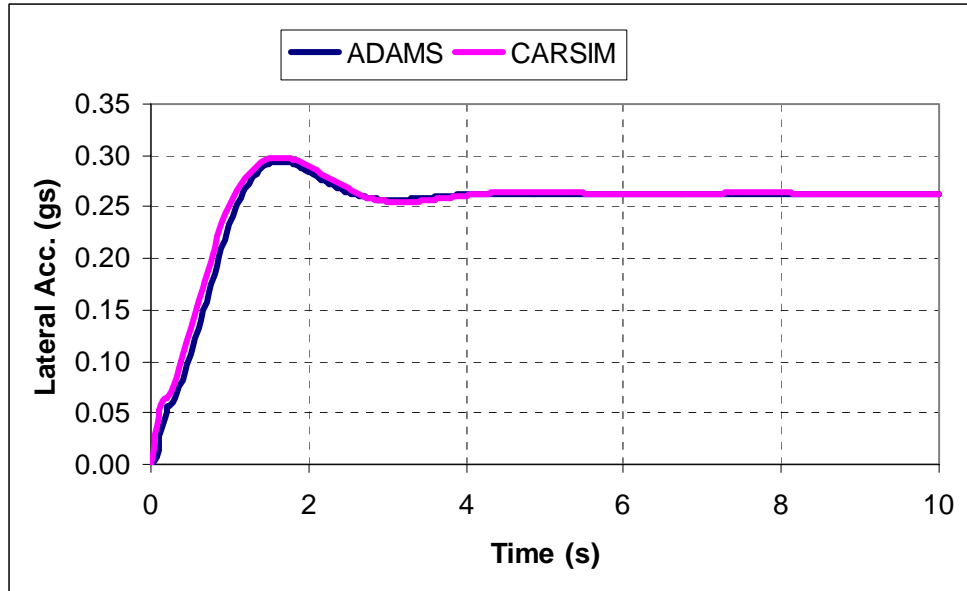


Figure 8: MSC.ADAMS and CarSim Educational results on Lateral Acceleration

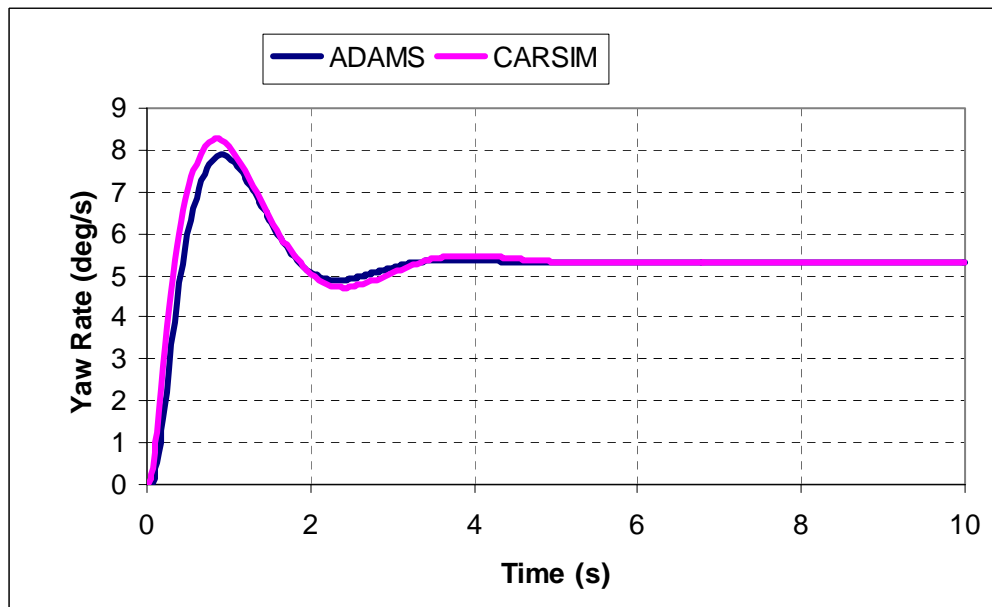


Figure 9: MSC.ADAMS and CarSim Educational results on Yaw Rate

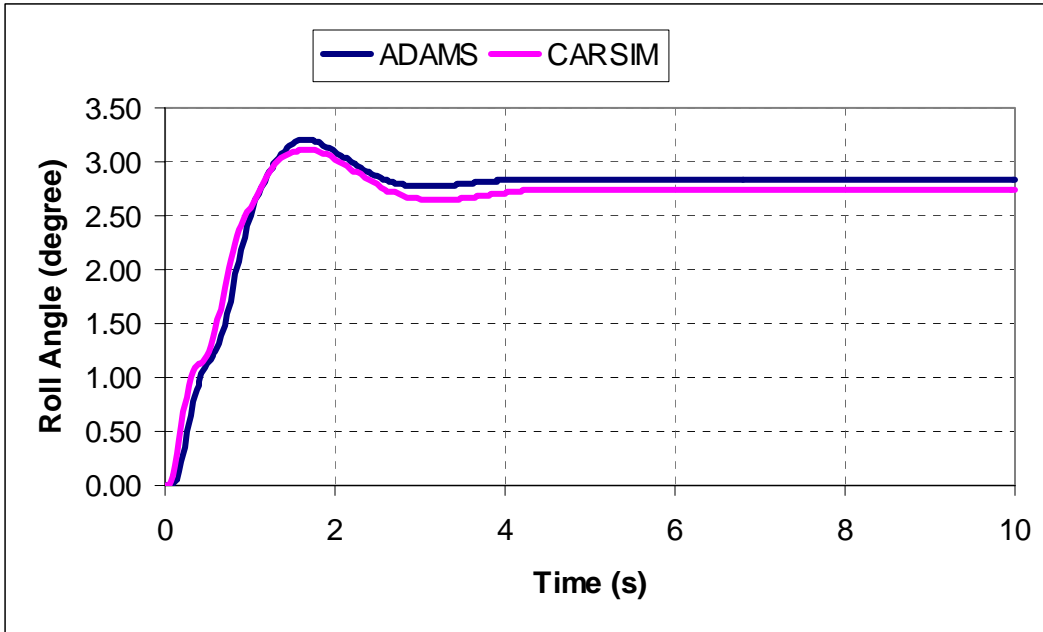


Figure 10: MSC.ADAMS and CarSim Educational results on Roll Angle

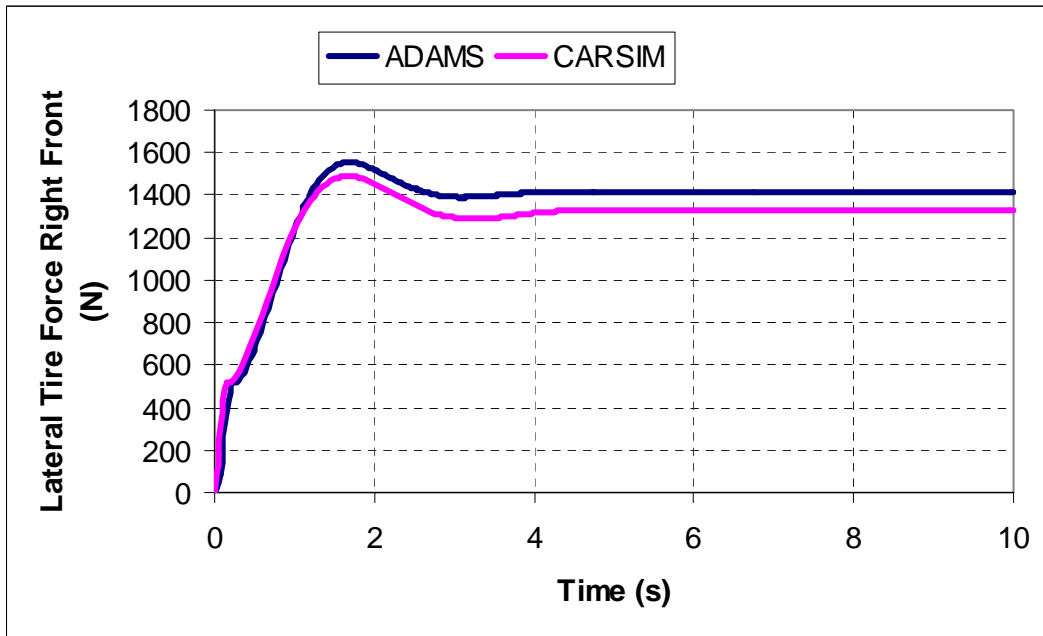


Figure 11: MSC.ADAMS and CarSim Educational results on Lateral Tire Force

Overall, the results on lateral acceleration, vehicle roll, yaw rate and lateral tire force correlate very well. Steady state values for the lateral acceleration and yaw rate are the same, and there is % 6 error between the models on the steady state value on the tire forces. The differences between the results can be attributed to the lateral tire model, and can be improved by adding the “tire lag”.

3.2. Comparison: MSC.ADAMS model versus NADS Results

The simulation is compared against maneuvers designed to simulate real world driving conditions, like lane-change maneuvers. Case studies and measurements from Salaani [6,7] are used for that purpose. MSC.ADAMS model results are compared with that of the data from National Advanced Driving Simulator (NADS) for a 1998 Chevrolet Malibu. The steering input profile is shown in Figure 12. The vehicle speed corresponding to this maneuver is 26 mph.

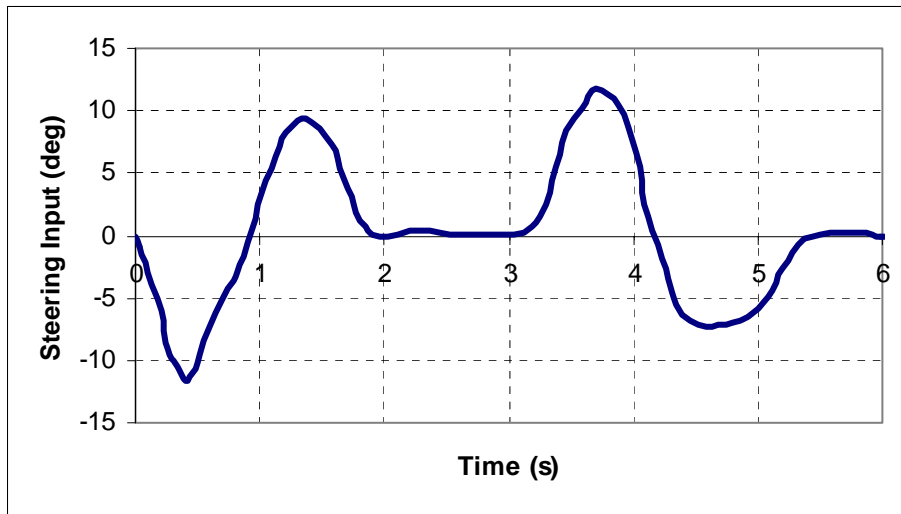


Figure 12: Steering Wheel versus Time

The performance of the model is compared to the measurement data from NADS. More specifically, lateral acceleration and roll angle results are compared in Figure 13 and Figure 14, respectively.

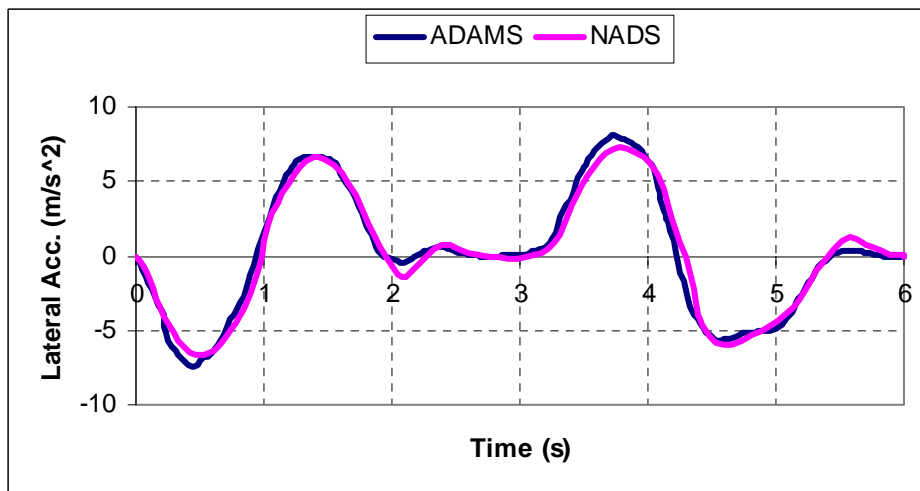


Figure 13: Lateral Acceleration versus Time

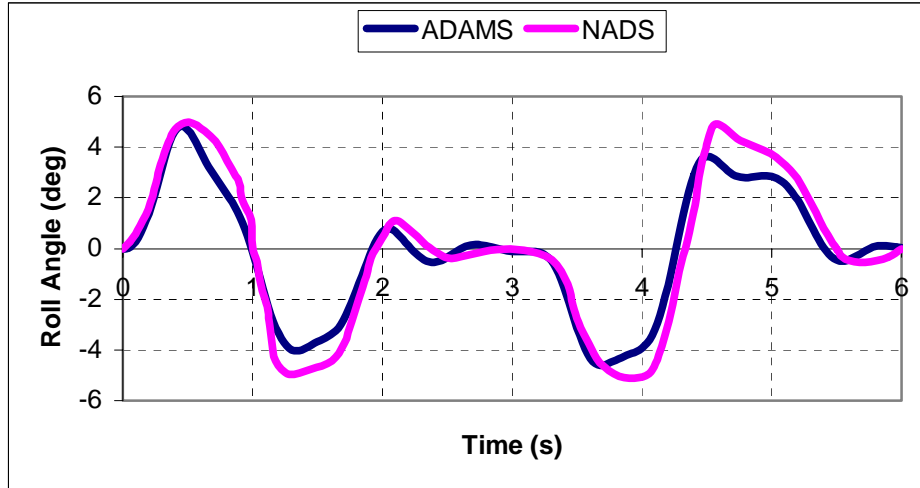


Figure 14: Roll Angle versus Time

Relative error between the model and data are calculated for the peak values and time to reach those peak values on lateral acceleration and roll angles. The results of that analysis are tabulated in Table 1.

Table 1: Relative Error on lateral acceleration and roll peak and time measures

Error Analysis	Acc. Time (%)	Acc. Peak (%)	Roll Time (%)	Roll Peak (%)
1 st Peak	6	7	6	4
2 nd Peak	1	1	5	19
3 rd Peak	2	11	3	9
4 th Peak	1	5	2	26

4. CONCLUSIONS AND SUMMARY

A generic 3-D vehicle model to study the handling performance is presented in this paper. The variables known to affect the performance of the vehicle have been parameterized in GUIs that enable study of their effects. Comparison results against CARSIM Educational and National Advanced Driving Simulator show that simulation outputs show a good correlation.

5. REFERENCES

1. Day, T. D., Roberts, S. G., York, A. R., 2001. "SIMON: A New Vehicle Simulation Model for Vehicle Design and Safety Research." SAE 2001-01-0503.
2. Gillespie, T. D., 1992. "Fundamentals of Vehicle Dynamics." SAE, Warrendale, PA.
3. Mechanical Simulation Corporation, 2000. "CARSIM Educational Version 4.5 User Reference Manual."
4. Orlandea, N. and Chace, M. A., 1977. "Simulation of a Vehicle Suspension with ADAMS Computer Program." SAE, Detroit, SAE preprint, 770053.

5. Riley, S. M., 2000. "Model Reduction of Multibody Systems by the Removal of Generalized Forces of Inertia." Ph.D. Thesis, The University of Michigan, Ann Arbor, MI.
6. Salaani, M. K., Guenther, D. A., and Heydinger, G. J., 2001. "Parameter Determination and Vehicle Dynamics Modeling for the NADS of the 1998 Chevrolet Malibu." SAE 2001-01-0140, SAE 2001 World Congress Detroit, Michigan March 5-8, 2001.
7. Salaani, M. K., Guenther, D. A., and Heydinger, G. J., 2001. "Model Validation of the 1998 Chevrolet Malibu for the National Advanced Driving Simulator." SAE 2001-01-0141, SAE 2001 World Congress Detroit, Michigan March 5-8, 2001.
8. Sayers, M. W. and S.M. Riley, 1995. "A Generic Multi-body Vehicle Model for Simulating Handling and Braking." IAVSD Symposium 1995.
9. Sayers, M. W. and S.M. Riley, 1996. "Modeling Assumptions for Realistic Multi-body Simulation of the Yaw and Roll Behavior of Heavy Trucks." SAE Paper No 960173.
10. Wong, J. Y., 1993. "Theory of Ground Vehicles." 2nd edition, John Wiley & Sons, Inc.