Using Scaled Vehicles to Investigate the Influence of Various Properties on Rollover Propensity

William E. Travis, Randy J. Whitehead, David M. Bevly, and George T. Flowers

Abstract— **Quantifying a vehicle's rollover propensity is a complex task and one that is of major concern to vehicle dynamicists. This research effort illustrates how a scaled vehicle can be used in determining vehicle properties that influence rollover propensity. A two-axis inertial measurement unit (IMU) and a global positioning system (GPS) unit are mounted to a scaled vehicle to measure its dynamic behavior. Vehicle maneuvers are performed on a test track and a computer vehicle simulation is used to compare the experimental results from the scaled vehicle with passenger vehicle dynamics. The simulation was able to accurately predict the dynamic behavior of the scaled vehicle, providing a link between full size vehicle roll dynamics and scale vehicle roll dynamics.**

I. Introduction

THE National Highway Traffic Safety Administration (NHTSA) reported that 3% of all light vehicle crashes in the United States involve rollover, yet are responsible for 1/3 of all passenger vehicle occupant fatalities [1]. In 2002, there were 10,626 fatalities due to single vehicle rollovers in the United States alone. Therefore, an opportunity exists to save lives by designing vehicles that are less prone to roll over [2]. However, the rollover testing of full-size vehicles is an expensive and somewhat dangerous endeavor. If results from scaled vehicles tested in a controlled environment can be related to the dynamic behavior of full-size vehicles, then such an approach can be an effective means of investigating rollover.

Scaled vehicles have proven to be reliable test beds for a variety of applications [3, 4, 5]. They also provide several advantages when testing and designing. Costs associated with a scale model vehicle [such as a radio-controlled (RC) car] are significantly lower, and modifications are easier to make than on a full-scale passenger vehicle. Since this is a scale model, the testing area is much smaller, allowing the testing environment to be more accurately controlled. Finally, pushing the vehicle to its limits to observe what happens in non-linear regions of vehicle roll is much safer with a scale vehicle than with a full-sized vehicle. [6].

There has been considerable research with regard to identifying parameters that significantly influence vehicle rollover. The most notable of these parameters are the vehicle center of gravity (CG) height and track width. In fact, they are the sole parameters that determine the Static Stability Factor (SSF). The passenger vehicle experiments to determine these parameters vary from static rollover tests [7], to dynamic tests incorporating vehicle tripping [7, 8, 9]. However, static rollover tests neglect the transient dynamics that are involved in the abrupt changes in velocity and steer angle that come before crashes, and tripping introduces high non-linearities into the dynamic system.

For the purposes of a dynamic rollover resistance-rating test, NHTSA selected the Fishhook steering maneuver as a primary candidate, which was refined in the Phase IV work [10]. The rollover propensity of a vehicle is determined from the highest speed for which it can complete the selected maneuver without achieving two-wheel lift. Since the Fishhook is conducted on-road, it is more repeatable and allows for more control over the test environment than off-road tripped tests [10]. Even though the evaluation procedure is only meant to test vehicles for on-road, untripped rollover propensity (which accounts for a small percentage of rollover crashes), it is believed that the results are still a valuable measure of overall rollover stability for relative comparison of various vehicles [11].

To study rollover propensity, this research effort uses a scaled vehicle to perform dynamic maneuvers, specifically the Fishhook 1a developed from the NHTSA study. The dynamic maneuvers are performed on the experimental test bed, as well as in a computer simulation. The computer simulation uses a transient roll and yaw dynamic model to capture the dynamics of the scaled vehicle during maneuvers.

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II. TEST BED

A. Details of the Scaled Car

A 1:10 scale radio controlled car was used as a test bed for rollover experimentation in order to validate the experiments done in the simulation. The vehicle was modified to vary CG location, spring stiffness, and roll center height. Varying the CG location provided the opportunity to view dynamics occurring at different weight splits. It also provides a means to adjust the distance between CG height and roll center height, which can be a crucial parameter when assessing steady state roll [12]. The test vehicle was configured with a rear wheel drive and front wheel steer configuration to better emulate many of today's RWD SUVs. Also, scale vehicles tend to have a relatively high cornering stiffness in comparison with full size vehicles [13]. In order to compensate for this effect, knobby tires were used to help scale the cornering stiffness as suggested by Brennan [13]. The factory steering servo proved to have an ample response time of 240 deg/s at the wheels and very little lag.

B. Details of the IMU

An inertial sensing module was constructed to meet the basic dimensional requirements of fitting on a scale car. It consists of two gyroscopes capable of 150 deg/s at a bandwidth of 40 Hz, one two-axis accelerometer capable of ±2g at 50Hz, a GPS receiver, and a Rabbit microprocessor. The gyroscopes are oriented to obtain roll rate and yaw rate on the scale car. The accelerometer is placed in the horizontal plane to obtain longitudinal (x) and lateral (y) accelerations. A GPS unit provides vehicle position, velocity, and course measurements, and when used in conjunction with accelerometers and gyroscopes can also provide measurements of vehicle sideslip [14]. The Rabbit was set up to record the inertial sensors at 100 Hz and the GPS receiver at 1 Hz.

C. Simulation Model Derivation

A MATLAB simulation was developed to explore the different parameters that dictate rollover propensity. It simulates a vehicle model using non-linear, transient dynamics of both yaw and roll. The accuracy of the computer simulation was verified by comparing simulation results to experimental data from NHTSA's Phase IV experiments on passenger vehicle rollover. It is important to note that NHTSA defines rollover as the point at which the magnitude of the normal loads on inside tires reach zero. Events happening after the occurrence of two-wheel lift contain many unknown nonlinearities that this simulation cannot accurately model. A more detailed description of the simulation model is provided in Whitehead, et al. [15].

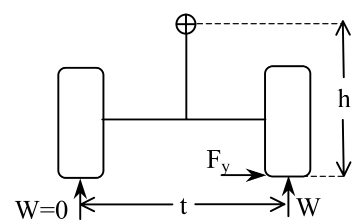
III. SIMULATION/EXPERIMENT COMPARISON

A. Maneuvers

The primary dynamic test used to determine the affects of these changing properties is a variation of the Fishhook 1a maneuver. The Fishhook 1a is a highly repeatable maneuver, as sited by NHTSA's Phase IV research. It is also an easily programmed open loop input, unlike its sibling the Fishhook 1b, which requires roll velocity data in a closed loop feedback control [10]. Variations to the NHTSA profile, including a different steering rate and the steering constant 'A', was held constant throughout all the experimental testing described herein.

B. Rollover Velocity

A simplified analysis of equations 1 and 2 was performed to develop a closed form expression for the rollover velocity (equation 5). Although they may have a significant affect on the dynamics, the inertial effects were neglected to simplify the analysis, as shown in equation 3, and provide a foundation for scaling the vehicle speed at which rollover occurs.



L=wheelbase
δ=steer angle
m=mass
V=velocity
h=CG height
W=weight
t=track
F_y=lateral force
K_{us}=understeer gradient
r=yaw rate

$$\sum F_y = ma_y = m\left(Vr * \dot{V}_y\right) \tag{1}$$

$$\sum M_{CG} = I_x \ddot{\phi} = F_y h - W\frac{t}{2} = m\left(Vr * \dot{V}_y\right)h - W\frac{t}{2} \tag{2}$$

Neglecting transient and inertial effects yields:

$$mVr = W\frac{t}{2} \tag{3}$$

$$r = \frac{V}{L + K_{US}V^2}\delta, \tag{4}$$

where K_{US} is at the limits of rollover.
Solving for V yields the rollover velocity:

$$V = \sqrt{\frac{WtL}{2mh\delta - WtK_{US}}} \tag{5}$$

C. Varying CG Location

An experimental study of the effects of CG height was performed on the scaled test bed and the results compared with those for a simulated Blazer undergoing the same test. The vehicle used for the comparison in this study is a 2001 Chevy Blazer 4x2. NHTSA's Phase IV experiment recorded data of the Blazer in a nominal configuration with a weight split of 55:45 and a SSF of 1.048. The Blazer and the scaled vehicle had the same front to rear weight distribution of 55:45 for this comparison. Figure 1 shows the effect of moving the center of gravity height for the Blazer simulation, while Figure 2 shows the same information for the scaled vehicle experimental testing [16].

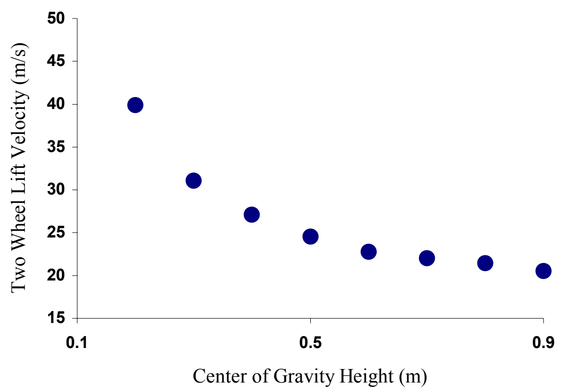


Figure 1. Simulation Data of Two-Wheel Lift Velocity versus CG Height for the Blazer.

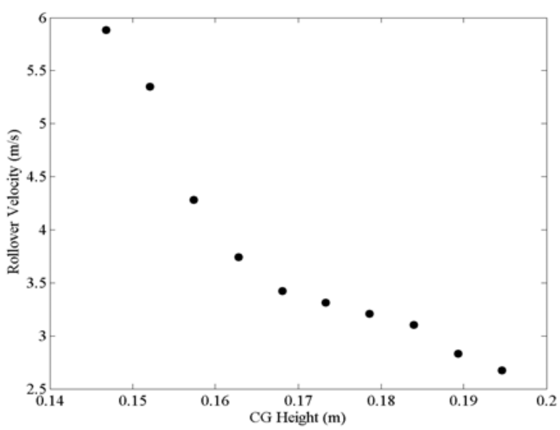


Figure 2. Scaled Experiment Data of Rollover Velocity versus CG Height

The scaled vehicle experimental data appears more linear due to the more narrow range of CG height that was used and the more limited number of data points in that window in comparison with the simulation data. In Figure 1, the SSF for the Blazer is varied from 0.1156 to 6.934, while the SSF for the scaled vehicle is varied from 0.4564 to 0.6138 in Figure 2. Equation 5 was then used to solve for the understeer gradient at rollover. As seen in Table 1, the understeer gradient grows larger with increasing velocity. This is due in part to the dynamics generating different amounts of weight transfer from run to run. Thus, the vehicle was operating on different parts of the tire curve as the normal loads varied.

TABLE 1
RC CAR UNDERSTEER GRADIENTS

V (m/s)	K_{US} (deg/g)
2.67	-3.53
2.83	-1.77
3.10	0.74
3.21	1.26
3.31	1.69
3.42	2.05
3.74	3.61
4.28	5.56
5.35	7.93
5.89	8.35

Rollover velocities with their corresponding understeer gradients.

D. Varying CG Longitudinal Location

The front to rear weight distribution for the simulated vehicle and for the scaled vehicle were each varied from 10:90 to 90:10, while holding the SSF constant. The results for are shown in Figures 3 and 4.

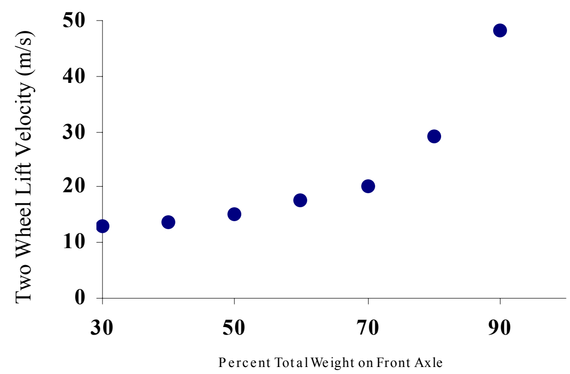


Figure 3. Simulation Data of Two-Wheel Lift Velocity versus Weight Distribution of a Blazer for a Fishhook 1a maneuver.

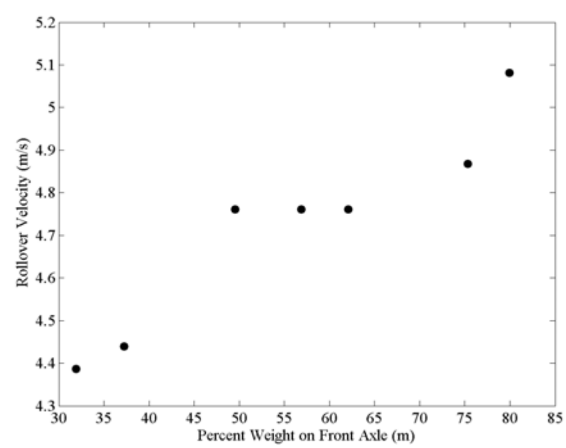


Figure 4. Scaled Experiment Data of Rollover Velocity versus Weight Distribution for a Fishhook 1a maneuver.

For the purposes of this study, the scaled car and Blazer configuration was modified to make the suspension symmetrical from front to rear by having identical front and rear roll centers, suspension stiffness, and damping. The parameters varied in this set of tests are the lengths ‘a’ and ‘b’ (where a represents the distance from the front axle to the CG, and ‘b’ represents the distance from the rear axle to the CG), while their sum (the wheelbase) is held constant. The effects on the understeer curve due to variation in the weight distribution are isolated with this approach. Examination of Figure 4 along with the understeer calculations of Table 1 reveals a correlation between understeer and rollover propensity. As the scaled vehicle’s weight split is shifted to the rear, it begins to oversteer as well as roll at a lower velocity than when the weight distribution is more towards the vehicle front.

E. Comparison of Experimental Scaled Vehicle Dynamics with Simulations of the Scaled Vehicle

In order to assess the accuracy of the simulation with regard to capturing the dynamics of the scaled vehicle, a ‘virtual scaled vehicle car’ was developed. Figure 5 shows a comparison of the roll dynamics from an experiment with simulation results for a Fishhook 1a maneuver, with ‘A’ equaling 7.3 degrees.

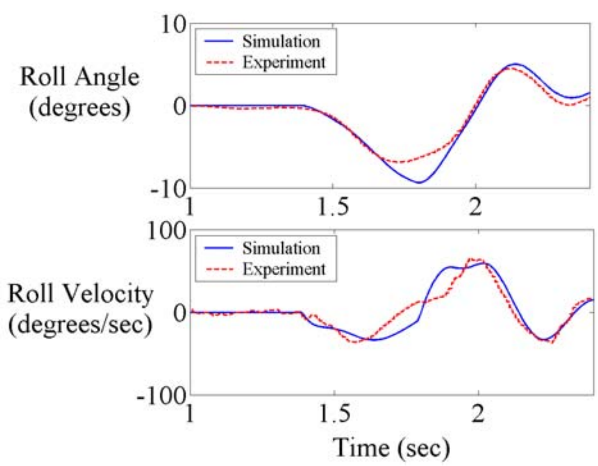


Figure 5. Simulation versus Experimental Dynamics in Roll

Figure 6 shows the normal forces on each tire during the same maneuver as Figure 5. During this experiment, minor two-wheel lift was observed. The simulation captures the two-wheel lift as can be seen when the normal force of both tires on either the inside or the outside simultaneously become zero. It should be noted that the two-wheel lift in this maneuver is minimal, since the maximum time that there was two-wheel lift was less than 0.3 seconds.

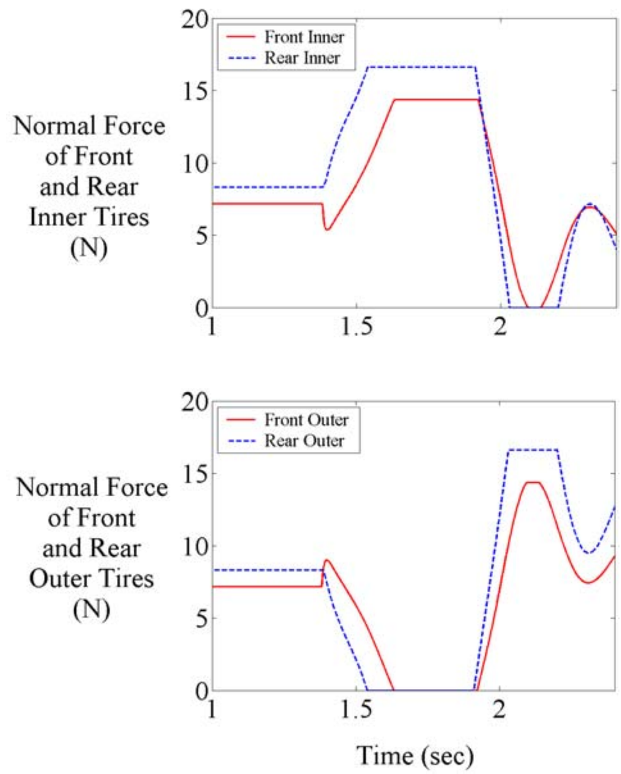


Figure 6. Simulation Data of Normal Forces in a Fishhook 1a Maneuver

Based upon the above results, it can be concluded that the simulation model is able to accurately predict the dynamic behavior of the scaled vehicle. So, by extension, it is also believed that a scaled vehicle test bed can provide insights that are relevant to passenger vehicles, as the same basic behaviors that were obtained experimentally can also be produced by using a passenger vehicle simulation.

IV. RC Car Dynamics

A. Tests

The scaled vehicle was subjected to four tests designed to measure roll. Each test was conducted by holding all other parameters as constant while varying either velocity, steer angle, CG height, or longitudinal CG location. Values common to all tests include the steering profile (a variation of a Fishhook 1a) and suspension configuration, which includes spring stiffnesses, dampers, roll center heights, and tires.

B. Velocity Variation

A variation of a Fishhook 1a with a steer angle, ‘A’, of 7.3 degrees was used for the steering profile. The vehicle velocity was increased, starting from 3.4 m/s and going up to 4.0 m/s. Example results are shown in figure 7. The vehicle remained stable at lower velocities because those do not produce sufficient lateral force to induce rollover. This is evident at a velocity of 3.4 m/s. The vehicle was approaching its limits at 3.9 m/s but was still stable. Two-wheel lift occurred and a high roll angle was recorded, but the car did not roll over. A slight increase in velocity (to 4.0 m/s) provided a sufficiently high enough lateral acceleration to quickly induce rollover.

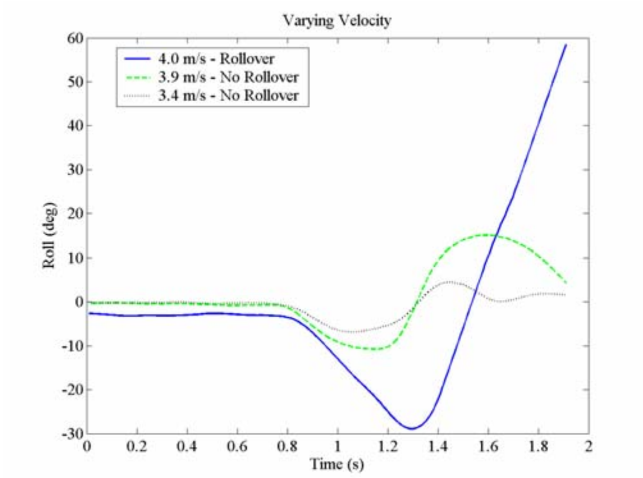


Figure 7. Experimental data showing roll angle at different velocities.

C. Steer Angle Variation

The vehicle speed was set at 3.0 m/s and a variation of a Fishhook 1a was used as the steering profile. The steer angles were started at 1.7 degrees and were gradually increased to a value of 9.3 degrees, at which point rollover occurred, as shown in figure 8. As expected, small amounts of roll occurred with small steer angles, and larger amounts occurred with larger angles.

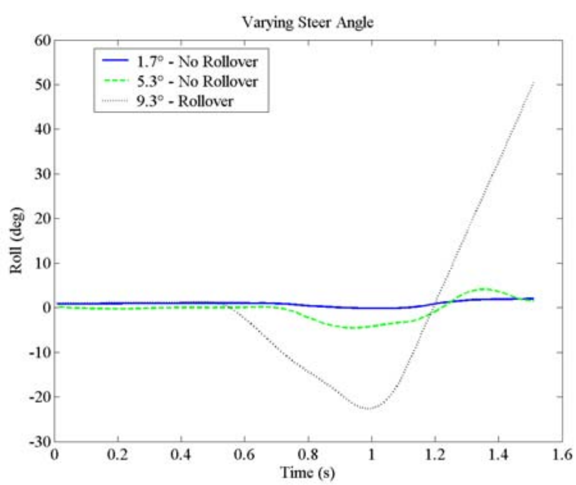


Figure 8. Experimental data showing roll angles for different steer inputs.

D. CG Height Variation

CG height was raised from an initial value of 0.152m (Pos. 1) to a final value of 0.163m (Pos. 2). The vehicle velocity was set at 3.4 m/s and a variation of a Fishhook 1a maneuver was used as the steering profile, with a steer angle amplitude of 7.3 degrees. Example results are shown in figure 9. The vehicle became unstable and experienced rollover when the CG height was at Pos. 2. However, for the CG height in Pos. 1, the selected velocity did not produce unstable behavior.

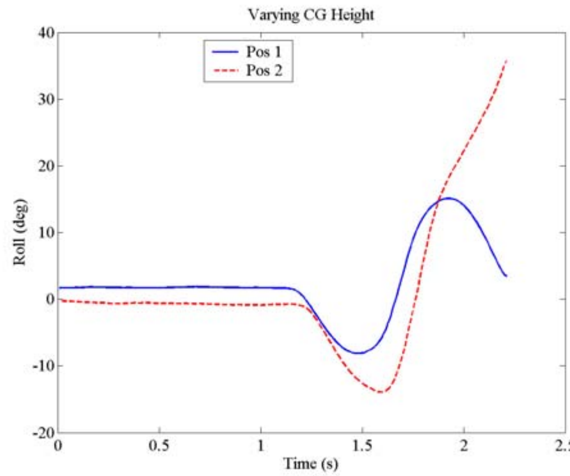


Figure 9. Experimental data showing roll angle at two different CG heights.

E. CG Longitudinal Location Variation

Testing was performed using a constant velocity of 3.9 m/s and a variation of a Fishhook 1a maneuver with a steer angle, 'A', of 5.3 degrees. Figure 10 shows the results for three different weight distributions on the vehicle (68:32, 50:50, and 32:68, front to rear). Shifting the CG forward produced a roll profile that mimicked the 50:50 split. Moving the CG to the rear of the vehicle has a more dramatic effect on the roll angles as the rollover propensity was raised. It is important to note that the SSF in these tests is held constant at 0.6138, but the rollover propensity of the vehicle is changing. This shows that the SSF cannot be used alone in determining the rollover propensity of any vehicle.

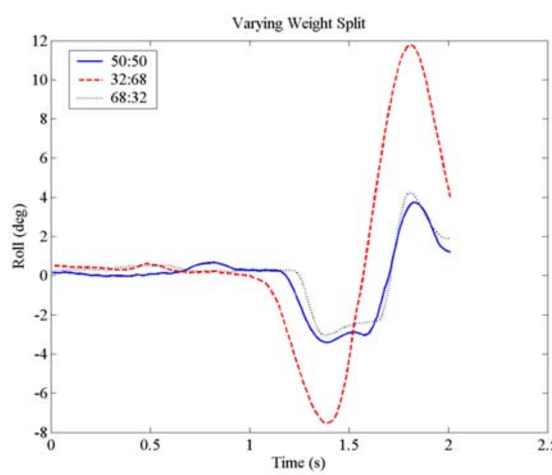


Figure 10. Experimental data showing roll angles at different CG locations.

V. RESULTS

An assessment of the dynamic behavior of a scaled vehicle provides insight with regard to the influence different parameters. The first and second tests show basic dynamic effects that occur at different steer angles and velocities. For both cases, larger inputs stimulate the roll dynamics to eventually become unstable.

As is predicted by the SSF for full size cars, rollover propensity increases as the CG height increases for the scaled vehicle. The SSF does not account for CG longitudinal location, but the stability of the scaled vehicle appear to be heavily dependent on it. A third and fourth set of tests evaluated the effects of CG location and show that it is a major parameter influencing rollover propensity for the scaled vehicle. It experiences significantly higher amounts of roll as the CG is moved towards the rear of the vehicle.

Finally, the strong correlation between the experimentally observed dynamics of the scaled vehicle with simulation results for full-sized vehicles serves to verify that scaled vehicles can be effectively used for some forms of rollover research.

VI. APPENDIX

A. Scaled Car Simulation Parameters

Mass of entire vehicle: 3.16 kg

Mass of sprung mass: 2.844 kg

Mass of un-sprung mass: 0.316 kg

Yaw moment of inertia (Iz): 0.23 N-m-sec2

Roll moment of inertia (Ix): 0.05625 N-m-sec2

Distance from CG to front axle tire patch: 0.137 m

Distance from CG to rear axle tire patch: 0.1182 m

CG height of sprung mass: 0.1468 m

CG height of un-sprung mass: 0.02 m

Front roll center (tuned): 0.02 m

Rear roll center (tuned): 0.02 m

Front track width: 0.168 m

Rear track width: 0.175 m

Distance between front spring and damper attachment points on the sprung mass: 0.0492 m

Distance between front spring and damper attachment points on the un-sprung mass: 0.1016 m

Distance between rear spring and damper attachment points on the sprung mass: 0.0762 m

Distance between rear spring and damper attachment points on the un-sprung mass: 0.1206 m

No anti-roll bar

Front springs (tuned): 3050 N/m per spring

Rear springs (tuned): 3050 N/m per spring

Front dampers (tuned): 90 N*s/m per damper

Rear dampers (tuned): 90 N*s/m per damper

B. Basic Blazer Parameters Used in Simulation

For a list of Blazer parameters, see reference [15].

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