2004-01-2094

A Study of the Effect of Various Vehicle Properties on Rollover Propensity

Randy Whitehead, William Travis, David M. Bevly, and George Flowers
Department of Mechanical Engineering, Auburn University

ABSTRACT

This paper investigates the effect of various vehicle parameters on rollover propensity using computer simulation. The computer simulation's accuracy is verified by comparing it to experimental data from NHTSA's Phase IV testing on rollover of passenger vehicles. The vehicle model used in the simulation study considers the non-linear, transient dynamics of both yaw and roll motion. The vehicle model is subjected to a specific steering input defined by NHTSA, the Fishhook 1a. A correlation between the vehicle parameter of center of gravity location and rollover propensity is found using the validated vehicle simulation.

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]. Due to the high fatality rate of rollover crashes, Congress passed the "Transportation Recall, Enhancement, Accountability and Documentation Act" (TREAD) in November of 2000. TREAD charged NHTSA to conduct dynamic rollover resistance rating tests, which NHTSA in turn made part of its New Car Assessment Program (NCAP). 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. 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 vehicle testing is conducted on-road, the results are more repeatable and give more control over the test environment than do off-road tripped tests [2]. 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 [3].

It should also be noted that NHTSA's rollover propensity testing procedure is not the only means that have been developed for evaluating un-tripped rollover propensity. Various static rollover tests have been developed and verified [4]. Other tests have also been developed that incorporate vehicle tripping [5, 6]. However, the static rollover tests neglect the transient dynamics associated with the abrupt changes in velocity and steer angle that come before crashes. The dynamics associated with tripped rollover also tend to be substantially different from untripped rollover, which is the focus of this paper.

It is well known that center of gravity height and track width, the two parameters that make up the Static Stability Factor (SSF), are the major vehicle parameters that contribute to rollover propensity. In this paper, the effects of other vehicle properties, such as suspension configuration and weight distribution, are evaluated to determine their influence on rollover propensity, while holding the SSF constant. A detailed description of the basic simulation model, including validation testing, and a discussion of the results for several parametric variation studies are provided in the following sections.

SIMULATION MODEL

Dynamic simulation has proven to be an efficient and accurate method for analyzing vehicles and evaluating their dynamic behavior [7,8]. As a part of the present effort, a simplified model that captures the essential vehicle dynamics associated with un-tripped rollover was developed and validated.

MODEL DEVELOPMENT – The primary dynamics that are of concern are those associated with the yaw and roll motions. Pitch dynamics (which cause longitudinal weight transfer) are neglected since longitudinal accelerations were kept small. The simulation model has 3 degrees of freedom (DOF) and was developed by deriving the equations of motion (EOM) using the free body diagrams (FBD) in Figures 1, 2, and 3.

Yaw Equations - The Yaw equations are derived from the "bicycle model" as seen in Figure 1 [16]. The 2-wheel bicycle model is the most commonly known version. However, for the purposes of this study, a 4-wheel bicycle model is used so that lateral weight transfer can be included in the yaw dynamics. It is assumed that the slip angles are symmetric along the x-axis of the vehicle, which is valid at high speeds and zero Ackerman Effect [9,10].

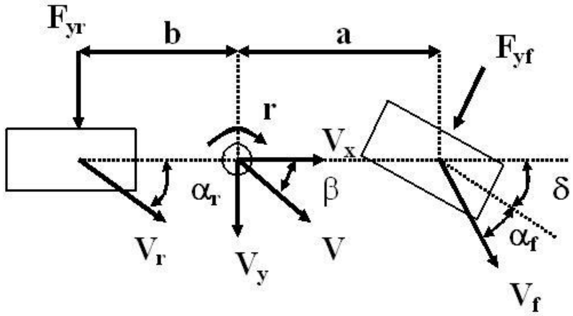

Figure 1. Bicycle Model, Yaw FBD

Summation of moments about the center of gravity yields:

$$I_z \cdot \dot{r} = -F_{yr} \cdot b + F_{yf} \cdot a \cdot \cos(\delta)$$

Summation of the forces in the y-axis direction yields:

$$\dot{V}_y = \left(F_{yr} + F_{yf}\right) \cdot \frac{\cos(\delta)}{(M+m)} - V \cdot r \cdot \cos(\beta)$$

The tire slip angles can be calculated from:

$$\alpha_r = a\tan\left[\frac{V_y - rb}{V_x}\right] \quad \alpha_f = a\tan\left[\frac{V_y + ra}{V_x}\right]$$

The vehicle's longitudinal and lateral velocities are defined by:

$$V_x = V\cos(\beta) \qquad V_y = V\sin(\beta)$$

The non-linear tire model uses the Pacejka Tire Model to calculate lateral force as a function of tire normal force and tire slip angle. Since the model of the tires used in the NHTSA experiment data is unknown for this work, the general parameters for the tire "Magic Formula" are from those tabulated in reference [11]. Additionally, camber thrust and longitudinal effects were also unknown and therefore neglected.

Roll Equations - The Roll Equations are derived by separating the sprung and un-sprung masses, as shown in Figure 3 and Figure 4, and applying Newton's second law (as formulated for rigid bodies). For the simulation testing, rollover is calculated by measuring the velocity of a specific maneuver at the instant that the normal forces of the tires (F_z) go to zero on either both left side or both right side tires. For simplicity, incidents of single wheel lift are not considered as a rollover event. Also, the un-sprung mass is considered to be in steady state for roll.

Summing the moments about the sprung mass yields:

$$I_{xM} \cdot \ddot{\phi}_M = R_z d_1 \sin(\phi_M) + R_y d_1 \cos(\phi_M)$$
$$-\frac{S}{2}(F_{bo} + F_{ko}) - \frac{S}{2}(F_{bi} + F_{ki})$$

The reaction force from the sprung mass is calculated as:

$$R_z = M(\ddot{z} + g)$$
$$\ddot{z} = d_1 \ddot{\phi}_M \sin\left(\dot{\phi}_M^2\right) + d_1 \dot{\phi}_M^2 \cos\phi_M$$

The reaction force from the un-sprung mass is:

$$R_y = m \cdot a_y + F_{yi} + F_{yo}$$
$$a_y = a_{cen} + \dot{V}_y$$

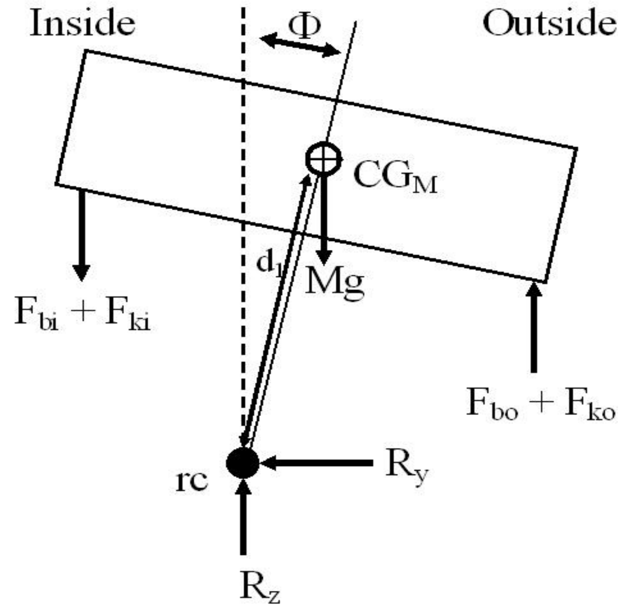

Figure 2. Roll FBD Sprung Mass

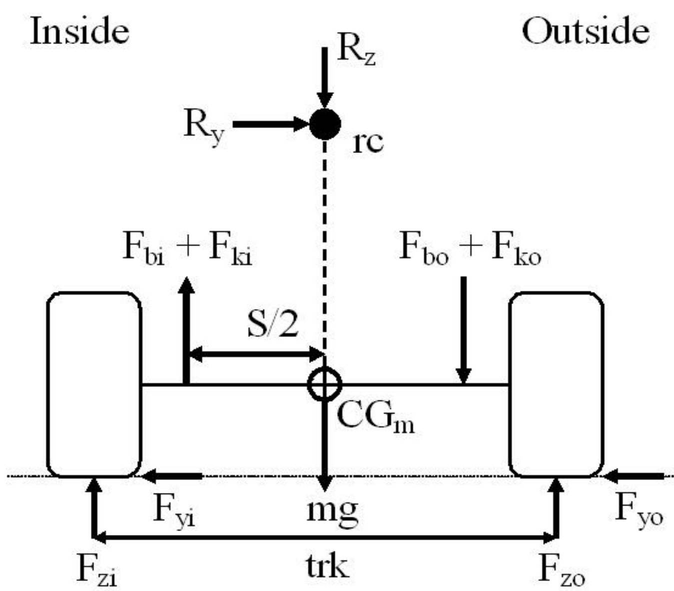

Figure 3. Roll FBD Un-Sprung Mass

The lateral weight transfer can be expressed as:

$$\Delta F_z = \frac{2h_{cgm}}{trk}(F_{yi} + F_{yo}) + \frac{2R_y}{trk}(h_{rc} - h_{cgm}) + \frac{S}{trk}(2F_b) + \frac{S}{trk}(2F_k)$$

The springs and dampers were assumed to be linear. Therefore the spring and damper forces can be calculated as:

$$z_{kb} = \frac{1}{2}\left[d_1 - d_1\cos(\phi_M)\right] + \frac{S}{2}\sin(\phi_M)$$

$$\dot{z}_{kb} = \frac{1}{2}\dot{\phi}\left[d_1\sin(\phi) + S\cos(\phi)\right]$$

$$F_{ki} = F_{ko} = k \cdot z_{kb} \qquad F_{bi} = F_{bo} = b \cdot \dot{z}_{kb}$$

The frame was assumed to be perfectly rigid such that all of the lateral weight transfer (ΔF_z) is through the suspension [10]. Additionally, suspension kinematics are considered stationary. The suspension kinematics influence roll centers, track width, and introduce jacking forces during the translation and rotation of suspension members [12]. However in this paper, the effects of suspension kinematics were neglected.

MODEL VALIDATION – In order to validate the vehicle model described above, simulation results were compared to NHTSA Phase IV experimental data for the Fishhook 1a maneuver (also known as the Fixed Timing Fishhook). The Fishhook 1a maneuver uses a steering input consisting of an initial steer followed by a counter steer at a set entrance velocity. The velocity profile of this maneuver is characterized by the vehicle reaching a desired steady state speed, known as the entrance speed, and coasting through the rest of the maneuver once the initial steer is begun. The profile consists of an initial zero steer angle followed by going to a steer angle 'A' at a rate of 720 degrees per second at the handwheel. 'A' is specific to each vehicle configuration, and is defined by multiplying 6.5 by the steer angle of the handwheel at which the vehicle experiences 0.3 g of lateral acceleration in a Slowly Increasing Steer (SIS) maneuver. The SIS maneuver is performed at a constant velocity of 50 mph with a continually increasing steer input of 13.5 degrees per second at the handwheel.

The steer angle 'A' is held constant for 0.250 seconds then a countersteer to '-A' at the same rate occurs. The steer angle '-A' is held constant for 3 seconds, after which it returns to zero, completing the maneuver. A graph of this profile is shown in Figure 4.

The vehicle used for comparison in this study is a 2001 Chevy Blazer 4x2. NHTSA's Phase IV experiments recorded data for the Blazer in three different configurations: Nominal, Reduced Rollover Resistance (RRR), and Rear Mounted Ballast (RMB). The Nominal configuration is the Blazer vehicle equipped with a driver, data acquisition, and outriggers on board. It has a weight distribution of 55:45 front to rear and a SSF of 1.048. The RRR configuration is the vehicle in the Nominal configuration with added weight on the top of the vehicle. This serves to raise the center of gravity vertically by 5% from the Nominal, resulting in a SSF of 0.989, and retains the same longitudinal weight split as the Nominal. The RMB configuration is the vehicle in the Nominal configuration with weight added to the rear of the vehicle, effectively moving the center of gravity longitudinally by 10.5 inches toward the rear, and retaining the same SSF. The RMB has a front to rear weight distribution of 44:56. For the Nominal and RRR configurations, there is available experimental data for a Fishhook 1a maneuver. For the RMB configuration, the only experimental data available to this study is for a Fishhook 1b maneuver, which is similar to the Fishhook 1a, but not identical to it [2,13,14,15].

The two main unknown properties of the Blazer needed for the simulation were the suspension roll damping and roll stiffness. In order to address this problem, a baseline for these parameters was estimated at first then tuned. Tuning was accomplished by adjusting the roll stiffness, damping, and roll center location until the simulation response closely matched with the experiment data for a J-Turn maneuver in the Nominal configuration. The values for the roll stiffness and damping, and roll centers determined in this fashion were held constant throughout all the various maneuvers and Blazer configurations used in the simulations.

The simulation model contains a 'virtual garage' that consists of 'virtual vehicles,' each with a configuration to match those of the Blazer used in the NHTSA experiment, i.e. Nominal, RRR, and RMB. Parameters in the virtual garage, such as wheelbase, track width, roll stiffness, roll damping, and roll center location, are held constant. The virtual vehicles varied from one another by having different center of gravity locations, total weight, and mass moments of inertia about both the yaw and roll axes.

Figures 4, 5, and 6 graphically compare the NHTSA experiment data with corresponding results from the simulation model. The inputs to the simulation are the steer angle and velocity data from the NHTSA experiments. The states compared are lateral acceleration, yaw rate, and roll angle.

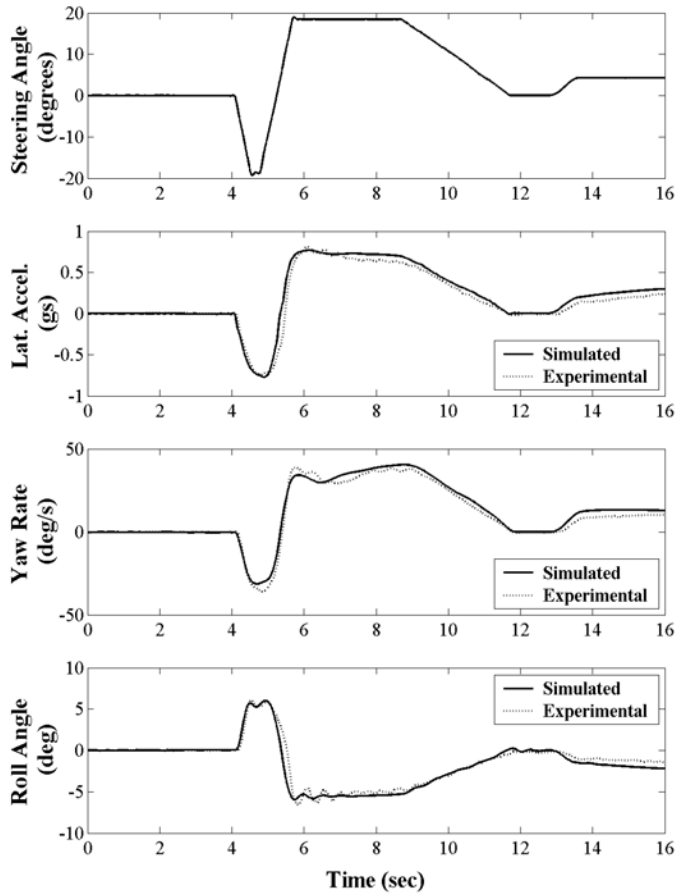

Figure 4. Fishhook 1a Nominal Configuration.
Steer Angle, Lateral Acceleration, Yaw Rate, and Roll Angle Graphs

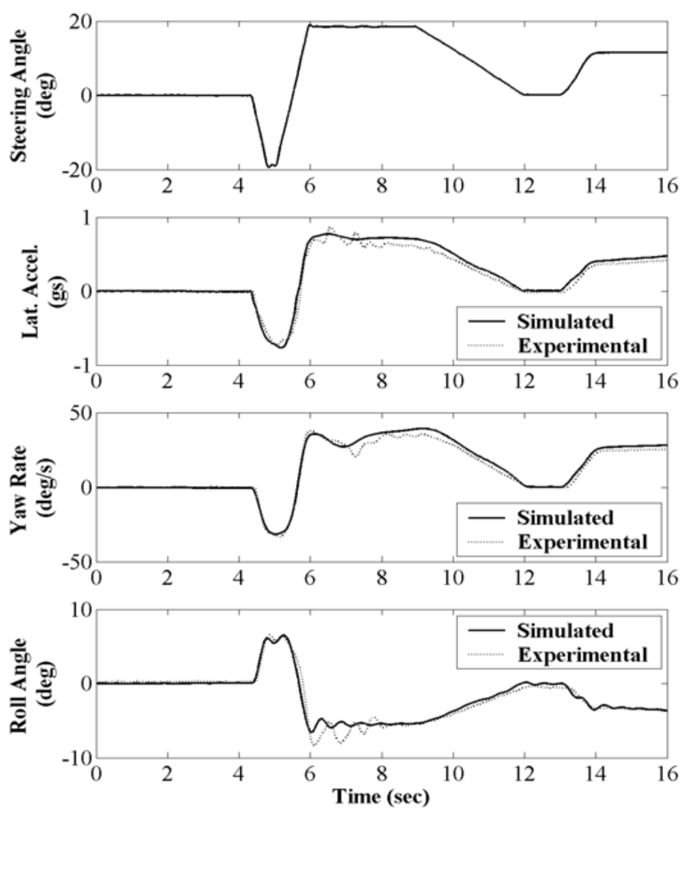

Figure 5. Fishhook 1a RRR Configuration
Steer Angle, Lateral Acceleration, Yaw Rate, and Roll Angle Graphs

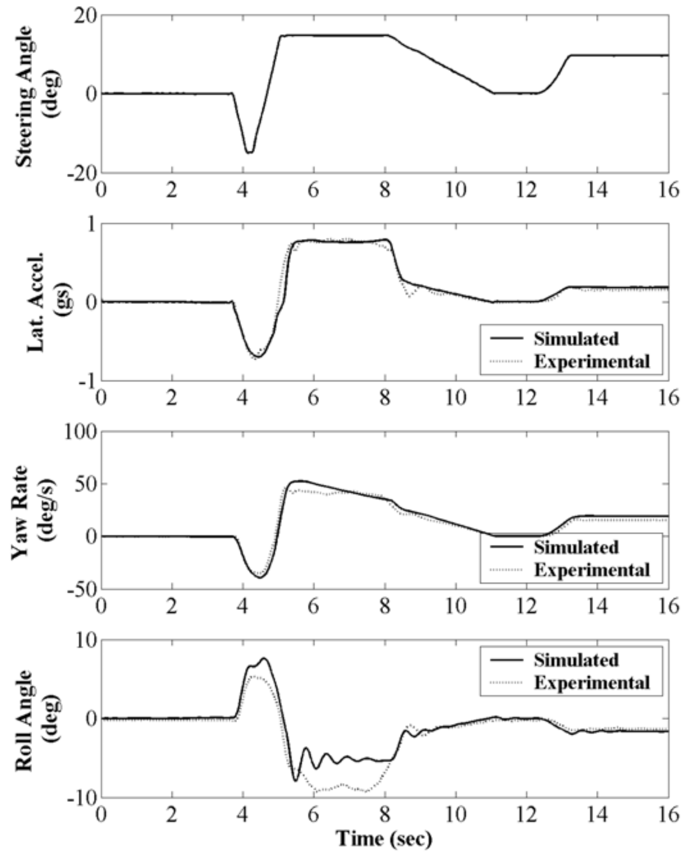

Figure 6. Fishhook 1b RMB Configuration
Steer Angle, Lateral Acceleration, Yaw Rate, and Roll Angle Graphs

In Figure 6, the largest discrepancy between the simulation results and the experiment data is for the roll angle. The experiment data reveals that two-wheel lift occurs, while the simulation result predicts only one-wheel lift that approaches two-wheel lift, as can be seen from the normal forces for each tire, as shown in Figure 7. This difference is most likely due to limitations of the simulation model. Some of the modeling assumptions may not be valid. For example, the roll stiffness may not be linear. Also, the assumption of un-sprung mass being in steady state may no longer be accurate after one wheel lift occurs for the roll dynamics (yet it appears to remain valid for the yaw dynamics). However, this discrepancy is small and it occurs only in the Fishhook 1b RMB data.

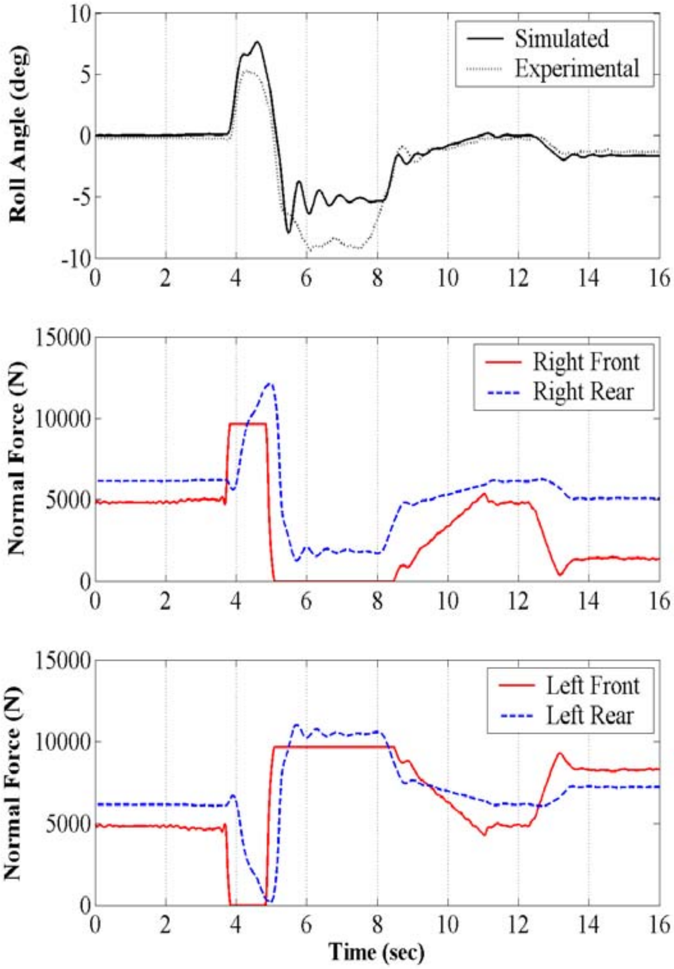

Figure 7. Fishhook 1b RMB Configuration Roll Angle and Normal Force (Fz)

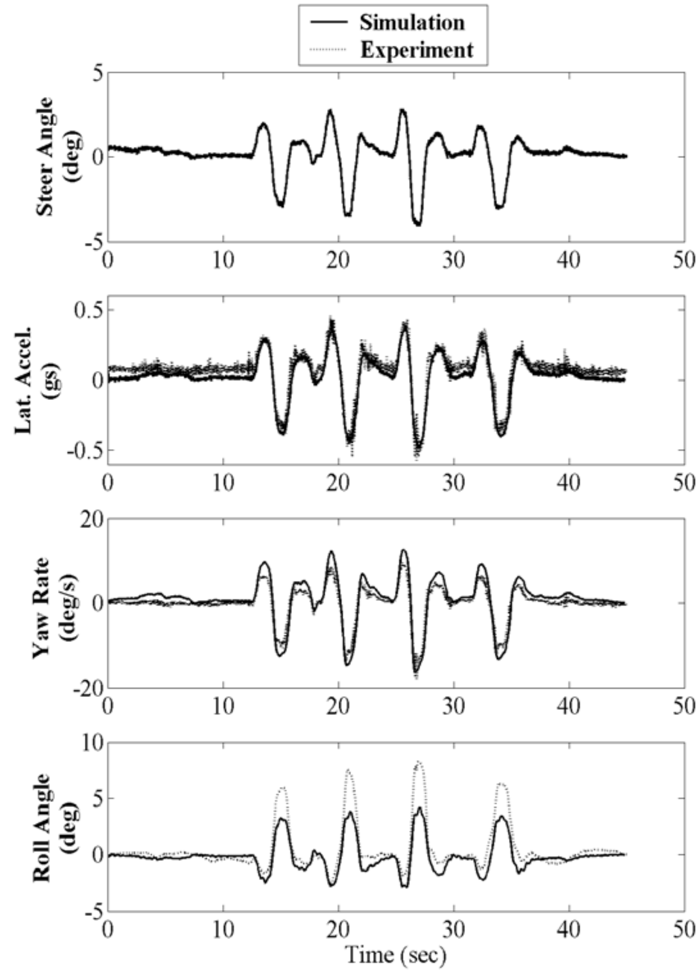

Figure 8. Lane Change Maneuver with the Nominal Blazer

The GPS and Vehicle Dynamics Lab at Auburn University Department of Mechanical Engineering, like NHTSA, has also done maneuvers with an instrumented Blazer to measure the vehicle's dynamics. These tests were done at Auburn's NCAT facility. This data was compared with simulation model results using the Nominal configuration of the Blazer, and using the velocity and steer angle data from the experiment data. Figure 8 shows a comparison of the Auburn experimental data with results from the simulation model. Differences between the simulation and experiment roll angles can be clearly seen in Figure 8. This is caused by a change in the bank angle of 4 degrees during the lane change maneuver.

The virtual vehicle used in the simulations was the Nominal Blazer. This is the same virtual vehicle that was initially tuned with NHTSA experiment data for a J-Turn maneuver. The similarity of the dynamic behavior as predicted by the simulation and that from the experimental data for maneuvers different from the J-Turn provides considerable assurance of the validity of the tuned parameters for this simulation.

SIMULATION STUDIES

Once the simulation model was validated, vehicle parameters were varied in order to assess their effect on rollover propensity. The dynamic test used to determine the effects of these properties is the Fishhook 1a maneuver. The Fishhook 1a is a highly repeatable maneuver as cited by NHTSA's Phase IV research and is an easily programmed open loop input (unlike its sibling the Fishhook 1b which requires roll velocity data in a closed loop feedback control) [2]. For each parametric variation, a new SIS constant had to be determined. This was accomplished by using the simulation to find the steer angle at 0.3g lateral acceleration during the SIS maneuver. Rollover velocity is calculated by determining the vehicle speed at which a steering maneuver causes "two wheel lift." "Two wheel lift" is defined to occur when the normal forces on both inside tires go to zero. By comparing the rollover velocities with the various vehicle properties, a correlation is identified between rollover propensity and those properties.

Varying CG Vertical Location – A series of simulation experiments that varied the CG height was performed on the virtual Blazer to assess the effect of this parameter on rollover propensity. The Blazer configuration used was the Nominal case, and the only parameter that varied was the CG height. The SIS constant was held constant at 1.268 degrees measured at the tire. Figure 9 shows the corresponding Blazer simulation results.

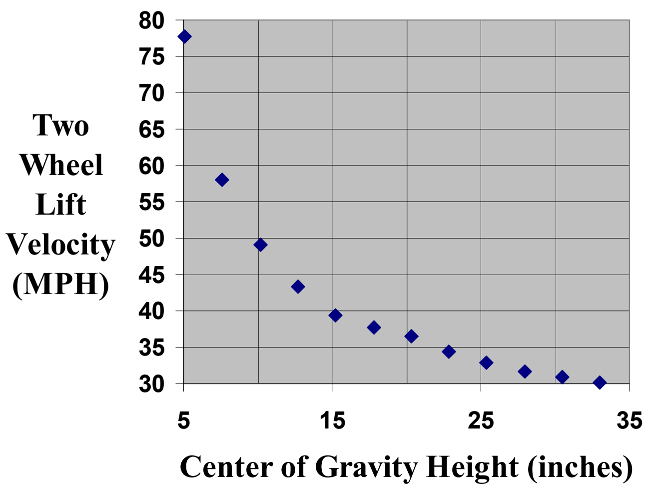

Figure 9. Simulation Data of Two-Wheel Lift Velocity versus CG Height for the Blazer

Since the track width is held constant, the SSF varies with CG height from 3.467 to 0.2774. The Nominal configuration has a SSF of 1.048, which corresponds to a CG height of 16.76 inches. The two wheel lift velocity for the Nominal configuration as determined by the simulation is 38.8 mph. NHTSA documented two wheel lift for the same configuration and maneuver as being 40.2 mph [13]. The observed trend of the data is not surprising since it is well known that a vehicle will rollover at a lower velocity if the CG is raised or the SSF is decreased.

Varying CG Longitudinal Location – In this set of simulation experiments, the front to rear weight split was varied from 30:70 to 70:30, while holding the SSF constant. The Nominal configuration of the Blazer was also modified so that the roll center, suspension stiffness and damping were the same front to rear. With these changes to the suspension, the effects on the understeer curve of changing the weight split are isolated from other factors. The parameters varied in this simulation test are the lengths 'a' and 'b', while their sum, the wheelbase, is held constant. The change in weight distribution caused the SIS constant to also change for each different weight split. The SIS constant varied from 0.675 to 1.827 degrees at the tire for the splits of 30:70 to 70:30 respectively as shown in Figure 11.

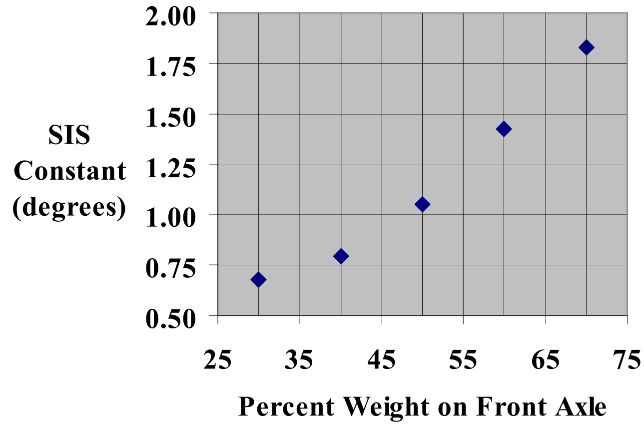

Figure 11. Simulation data of SIS Constant versus Weight Split

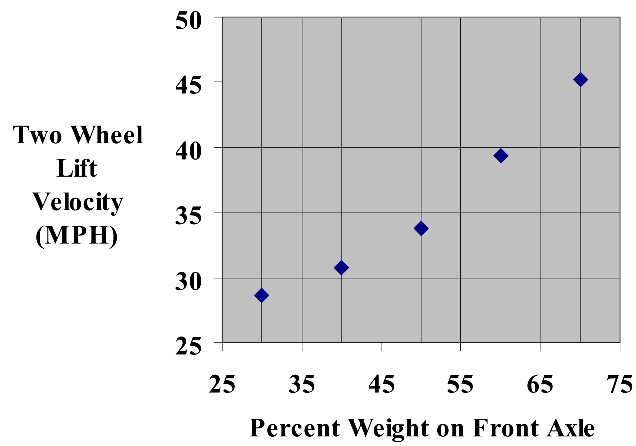

Figure 12. Simulation Data of Two-Wheel Lift Velocity versus Weight Split of a Blazer in a Fishhook 1a Maneuver

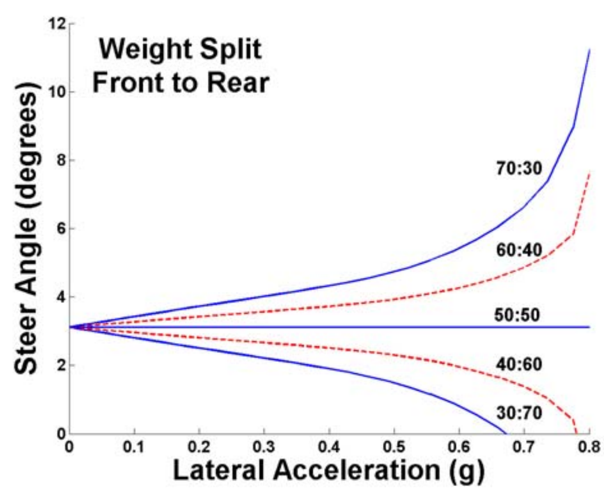

Figure 13. Simulation Data showing the Understeer Curve for the various Weight Splits

Figure 12 shows the two-wheel lift velocity as a function of the percent of total weight on the front axle. Figure 13 shows the corresponding understeer curve. Inspection of these two figures reveals a correlation between understeer and rollover propensity. As the Blazer's weight is shifted towards the rear, the vehicle begins to oversteer as well as roll at a lower velocity than when the weight is shifted towards the front axle. Also note, as the weight is shifted towards the front axle, the corresponding SIS increases causing the Fishhook 1a to because a more severe maneuver, however the two wheel lift velocity continues to increase as the weight is moved to the front axle. NHTSA's Phase IV experiments reveal this same correlation between weight split and rollover propensity. The Nominal and RMB have approximately the same SSF, and weight splits of 55:45 and 44:56 respectively. NHTSA performed a Fishhook 1b maneuver on both of these configurations. With the initial steer to the left, and countersteer to the right, the Nominal configuration shows two wheel lift at 40.2 mph, while the RMB shows two wheel lift at 34.9 mph [13].

Another interesting observation from the simulation data was seen for two vehicle configurations with a SSF equal to 1.048. One vehicle configuration experiences two-wheel lift at 28.6 mph and the other experiences it at 45.2 mph. These speeds were recorded for weight distributions of 30:70 to 70:30 front to rear, respectively. So, even though the Static Stability Factor is considered the most important measurement of a vehicle's rollover propensity, two vehicles with the same SSF may experience a difference in rollover velocity for the same maneuver. This clearly demonstrates that there are other vehicle parameters that play an important role with regard to rollover propensity besides center of gravity height and track width.

CONCLUSION

This paper has shown that a simple vehicle model developed to study the transient dynamics of passenger vehicles can capture the dynamics seen in events leading to rollover. This was demonstrated with a correlation between simulation results and experimental data; therefore giving validation to using the simulation to study rollover propensity.

This paper has considered variations in two vehicle properties, the center of gravity vertical and longitudinal locations. The influence of these parameters on vehicle rollover propensity has been evaluated with both model simulation studies and with experimental data. It was shown that the SSF is not the sole determinate of a vehicle's rollover propensity, and that there are vehicle properties other than SSF, such as weight distribution, that can significantly influence rollover propensity.

The simulation studies have also revealed what appears to be a correlation between the understeer curve and rollover propensity. The vehicle understeer increases as the two-wheel lift velocity increases. Future research will investigate the effect of other vehicle properties, such as suspension setup, tire models, etc., on rollover propensity using simulation and scaled experiments [17].

ACKNOWLEDGMENTS

This work was supported by the U.S. Department of Transportation under contract FHWA-TCSP-TCSP6.

REFERENCES

[1] Ponticel, P. 2003. "Dynamic Testing Rollover on the Way," *Automotive Engineering International* (November 2003): 26-28.

[2] Forkenbrock, G. J., Garrott, W.R., Heitz M. and O'Hara, B.C. 2003. An Experimental Examination of J-Turn and Fishhook Maneuvers that May Induce On-Road, Untripped, Light Vehicle Rollover. SAE Paper No. 2003-01-1008.

[3] Viano, D. C. and Chantal, P. 2003. Case Study of Vehicle Maneuvers Leading To Rollovers: Need for a Vehicle Test Simulating Off-Road Excursions, Recovery and Handling. SAE Paper No. 2003-01-0169.

[4] Christos, J. P. and Guenther, D. A. 1992. The Measurement of Static Rollover Metrics. SAE Paper No. 920582.

[5] D'Entremont, K. L., Zhengyu, L. and Nalecz, A. G. 1993. An Investigation into Dynamic Measures of Vehicle Rollover Propensity. SAE Paper No. 930831.

[6] Cooperrider, N. K., Hammound, S. A. and Thomas, T.M. 1990. Testing and Analysis of Vehicle Rollover Behavior. SAE Paper No. 900366.

[7] Garrott, W. R., and Heydinger, G. J. 1992. An Investigation, Via Simulation, of Vehicle Characteristics that Contribute to Steering Maneuver Induced Rollover. SAE Paper No. 920585.

[8] Allen, R. W., Szostak,H.T, Rosenthal, T. J. and Klyde,D. H. 1990. Field Testing and Computer Simulation Analysis of Ground Vehicle Dynamic Stability. SAE Paper No. 900127.

[9] Gillespie, T. D. 1992. *Fundamentals of Vehicle Dynamics*. Society of Automotive Engineers. Warrendale, PA. ISBN: 1-56091-199-9.

[10] Milliken, D. L. and Milliken, W. F. 1995. *Race Car Vehicle Dynamics*. Society of Automotive Engineers. Warrendale, PA. ISBN: 1-56091-526-9.

[11] Bakker, E., Nyborg, L. and Pacejka, H. B. 1987. Tyre Modelling for Use in Vehicle Dynamics Studies. SAE Paper No. 870421.

[12] Hac, A. 2002. Rollover Stability Index Including Effects of Suspension Design. SAE Paper No. 2002-01-0965.

[13] Forkenbrock, G.J., Garrott, W.R., Heitz, M. and O'Hara, B.C. NHTSA. 2002. A Comprehensive Experimental Examination of Test Maneuvers that May Induce On-Road, Untripped, Light Vehicle Rollover – Phase IV of NHTSA's Light Vehicle Rollover Research Program (October). Report No. DOT HS 809 513.

[14] Bixel, R. A., Garrott, W.R, Guenther, D. A., Gary J. Heydinger, Howe, J. G. and Payne, M. 1999. Measured Vehicle Inertial Parameters – NHTSA's Data Through November 1998. SAE Paper No. 1999-01-1336.

[15] NHTSA. Phase IV Rollover Testing. 2002. George Washington Transportation Research Institue.

[16] Bevly, D. M., Ryu, J., Sheridan, R., and Gerdes, J. C, Integrating INS Sensors with GPS Velocity Measurements for Continuous Estimation of Vehicle Side-Slip and Tire Cornering Stiffness. Submitted to *IEEE Transactions on Intelligent Transportation Systems.*

[17] Travis, W., Whitehead, R., Bevly, D. M., and Flowers G. T. Using Scaled Vehicles to Investigate the Influence of Various Properties on Rollover Propensity. Proceedings of the 2004 American Controls Conference, June 2004.

CONTACT

Randy Whitehead is a M.S. candidate in the Department of Mechanical Engineering at Auburn University. He received his B.S. in Mechanical Engineering from Auburn University in August 2003. Randy was the frame designer of the 2003 Auburn Formula SAE team, and has been apart of Auburn FSAE since 2001. His email address as of this publication is whiterj@auburn.edu. "Trust in the Lord with all your heart... and he will direct your paths" - Proverbs 3:5-6.

APPENDIX

This is a list of the variables used on the Nominal Blazer configuration for the simulation.

Mass of entire vehicle: MT = 1907.1 kg
Mass of sprung mass: M = 1525.7 kg
Mass of un-sprung mass: m = 381.4 kg
Mass moment of inertia about the z-axis: Iz = 3833.31 N-m-sec2
Mass moment of inertia about the x-axis: Ix = 550.54 N-m-sec2
For more details on mass moments of inertia see [13].
Distance from CG to front axle tire patch: a = 1.216 m
Distance from CG to rear axle tire patch: b = 1.502 m
CG height of sprung mass = 0.6629 m
CG height of un-sprung mass =0.4 m
Front roll center (tuned) = 0.4 m
Rear roll center (tuned) = 0.25 m
Front track width = 1.445 m
Rear track width = 1.405 m
Distance between front springs: Skf = 0.7747 m
Distance between rear springs: Skr = 0.9906 m
Distance between front dampers: Sbf = 0.7747 m
Distance between rear dampers: Sbr = 0.762 m
Distance between front anti-roll bar (ARB) points where force is applied to the suspension: Sarbf = 1.0287 m
Distance between rear ARB points where force is applied to the suspension: Sarbr = 0.6731 m
Force applied by front ARB: 700 N per degree roll
Force applied by rear ARB: 400 N per degree roll
Stiffness of a front springs (tuned) = 75000 N/m per spring
Stiffness of a rear springs (tuned) = 70000 N/m per spring
Damping of a front dampers (tuned) = 5000 N*s/m per damper
Damping of a rear dampers (tuned) = 4000 N*s/m per damper
Steering ratio from handwheel angle to tire steer angle = 1/18