International Journal of Automotive Technology, Vol. 7, No. 6, pp. 703−714 (2006)

1229−9138/2006/030−09

SENSITIVITY ANALYSIS OF SUV PARAMETERS ON ROLLOVER PROPENSITY

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(Received 23 January 2006; Revised 21 May 2006)

ABSTRACT−The growing concern surrounding rollover incidences and consequences of Sports Utility Vehicles (SUV) have prompted to investigate the sensitivity of critical vehicle parameters on rollover. In this paper, dynamic rollover simulation of Sports Utility Vehicles is carried out using a validated nonlinear vehicle model in Matlab/Simulink. A standard model is considered and critical vehicle parameters like CG height, track width and wheel base are varied within chosen specified limits to study its influence on roll behavior during a Fishhook steering maneuver. A roll stability criterion based on Two Wheel Lift Off (TWLO) phenomenon is adopted for rollover propensity prediction. Further dynamic rollover characteristics of the vehicle are correlated with Static Stability Factor (SSF), Roll Stability Factor (RSF) and Two Wheel Lift Off Velocity (TWLV). These findings will be of immense help to SUV chassis designers to determine safety limits of critical vehicle parameters and minimize rollover incidences.

KEY WORDS : Rollover, Parameter sensitivity, Roll stability factor, Two wheel lift off (TWLO)

NOMENCLATURE

m : total mass of the vehicle
m_s : total sprung mass of the vehicle
h : height of the vehicle CG from ground
T : track width of the vehicle
L : wheel base of the vehicle
a : longitudinal position of CG from front axle
b : longitudinal position of CG from rear axle
r : yaw rate
I_{xxs} : roll inertia
I_z : yaw inertia
I_{xz} : inertia product
e : distance between CG and roll axis
s : distance between springs/dampers
K : spring stiffness
B : damping coefficient
$K_{f\phi}$: front roll stiffness
$K_{r\phi}$: rear roll stiffness
K_ϕ : total roll stiffness
F_x : longitudinal tire force
F_y : lateral tire force
F_z : vertical tire force
A_x : longitudinal acceleration
A_y : lateral acceleration
ϕ : roll angle
δ : road wheel steer angle
g : acceleration due to gravity

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1. INTRODUCTION

The wide popularity of light trucks including Sports Utility Vehicles (SUVs), pick-ups and mini-vans with dangerous combination of high CG height and narrow track width have raised important road safety concerns. The improvements in highway systems and vehicle performances have enabled these vehicles to travel at higher speeds aggravating rollover injuries and fatality chances. During rollover crashes the unrestrained occupants are subjected to variety of loads and impacts mainly due to chances of occupant ejection and roof crush (Deutermann, 2002). Figure 1 shows that the fatality chances in SUV rollover are nearly three times than that of small passenger cars (Brewer, 2001).

Rollovers can be broadly classified as untripped and tripped rollovers. An untripped rollover is characterized by one-sided vertical movement of the vehicle due to dynamic weight transfer from inner wheels to the outer wheels. Once the entire vertical tire force on the inner wheels diminish, the vehicle CG exceeds the stable limit leading to a rollover event. A tripped rollover is initiated by lateral hit against an obstacle or curb or lateral slip on a surface with varying friction characteristics (Kim *et al.*, 2006).

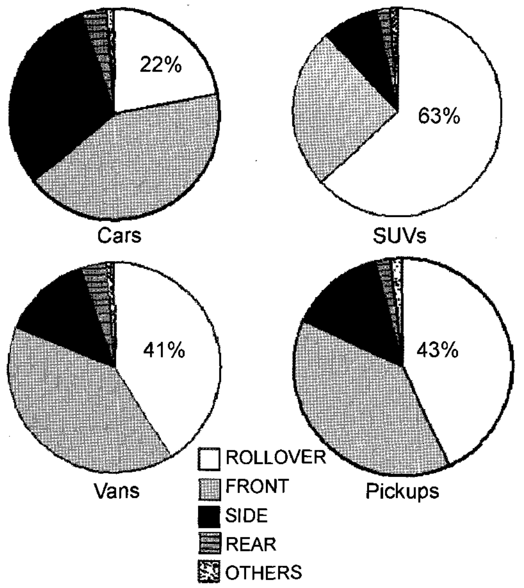

Figure 1. Occupant fatality distribution by crash mode 1999 FARS (Brewer, 2001).

Figure 2 shows the increasing trend of fatal rollover crashes experienced by SUVs among the light truck vehicles from 1991 to 2000. According to National Highway Traffic Safety Administration (NHTSA) in 2002 alone, more than 10,000 people died in rollover crashes. As part of their New Car Assessment Program (NCAP), the Star Rating Scheme aims at providing rollover safety awareness among the customers. Thus it is very essential for chassis designers to establish standards and safety limits of critical vehicle parameters to reduce the high rollover propensity and make SUVs safer and less dangerous to others on the road. The conventional methods of field tests prove to be time consuming, expensive, less flexible, difficult and dangerous to perform. In view of these limitations, an accurate and reliable simulation model can play a significant role in studying the dynamic rollover behavior of these vehicles. In this paper, SUV parameter sensitivity on untripped rollover phenomena using non-linear vehicle modeled in Matlab/Simulink is investigated.

2. VEHICLE PARAMETERS INFLUENCING ROLLOVER

The higher rollover likelihood of a SUV can be attributed to many factors. The rollover phenomenon begins with vehicle tending to oversteer and yaw so that it begins to slide laterally. A vehicle's physical dimensions, total weight and its distribution on each axles, tires, tire pressure and suspension characteristics can all contribute to this tendency. It is well known fact that the CG height above ground level and track width are the primary factors determining the rollover propensity. Assuming the vehicle to be a rigid body the Static Stability Factor (SSF) (Gillespie, 2001) can be computed as

$$SSF=\frac{T}{2h}=\frac{Track\ Width}{2\times(C.G.\ Height)} \tag{1}$$

Therefore with an increase in SSF, the vehicle attains stability due to higher rollover threshold values. An increase in CG height or decrease in track width would increase rollover tendency. But SSF valu alone is a poor representation of dynamic roll propensity of the vehicle during severe steering maneuvers. Figure 3 shows the Static Stability Factor of different light trucks as a function of vehicle mass with driver only configuration (Heydinger, 1999). It is evident that the SUVs are highly prone to rollover as they have the lowest Static Stability Factor compared to other vehicle types.

Figure 4 gives an overview of the percentage change in time taken to rollover with 10% variation in corresponding vehicle parameters (Chan, 2000). In this study, the total roll time is defined as the time required for the vehicle to

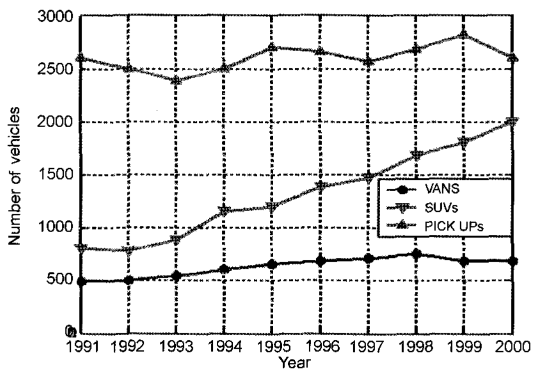

Figure 2. Light trucks involved in fatal rollover crashes by body type [source: NCSA, NHTSA, FARS 1991–2000] (Garrott, 2001).

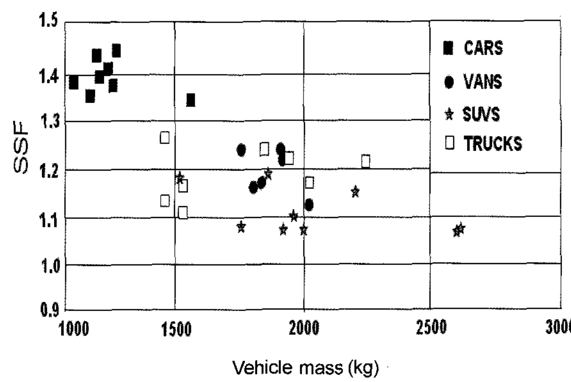

Figure 3. SSF vs. vehicle mass with driver only configuration (Heydinger, 1999).

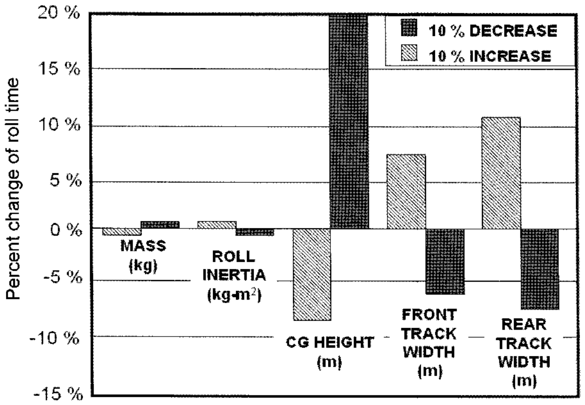

Figure 4. Parametric comparison of a basic 4 wheeled truck (Chan, 2000).

roll 90 degrees onto its side. A decrease in roll time indicates the higher rollover propensity of the vehicle considered and vice versa. With 10% increase in mass and CG height the roll time is expected to decrease, whereas with 10% increase in roll inertia and track width the roll time increases.

3. NON-LINEAR VEHICLE MODEL

A nonlinear vehicle model with 8 DOF along with System Technology Inc. (STI) nonlinear tire model is developed and used for the entire analysis (Segel, 1956 & Alluom, 1997). The vehicle is represented by two masses–sprung mass and unsprung mass. The pitching motion causing longitudinal weight transfer is not considered as longitudinal acceleration is kept negligible. Moreover, the aerodynamic coefficients are neglected and a plane road without irregularities is considered for all simulations. The vehicle body is assumed to be rigid. The vehicle motions considered for modeling include longitudinal velocity, lateral velocity, yaw rate, four wheel steer angles as shown in Figure 5 and sprung mass roll motion as shown in Figure 6. The different equations used for representing the nonlinear vehicle model is summarized in Appendix I.

4. VEHICLE CONFIGURATION

The nonlinear vehicle is validated against available test results for Ford Taurus passenger car and Chevy Blazer-Reduced Rollover Resistance (RRR) SUV configuration (Forkenbrock, 2002). The Ford Taurus has a SSF of 1.408. The Chevy Blazer RRR includes roof mount ballast positioned such that longitudinal CG location of the vehicle is not affected, but SSF reduces by 0.05 from the nominal configuration. The unknown parameters like distance between springs, spring stiffness, coefficient of damping and distance between roll axis and center of gravity are tuned during validation. The Chevy Blazer Nominal configuration is used for all parameter sensitivity studies. Appendix II summarizes the important vehicle parameters used for different vehicles in this study.

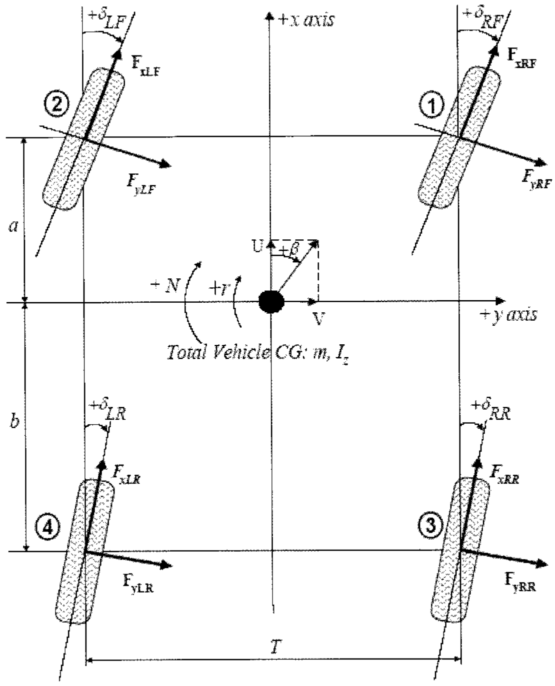

Figure 5. Vehicle variables on the ground plane.

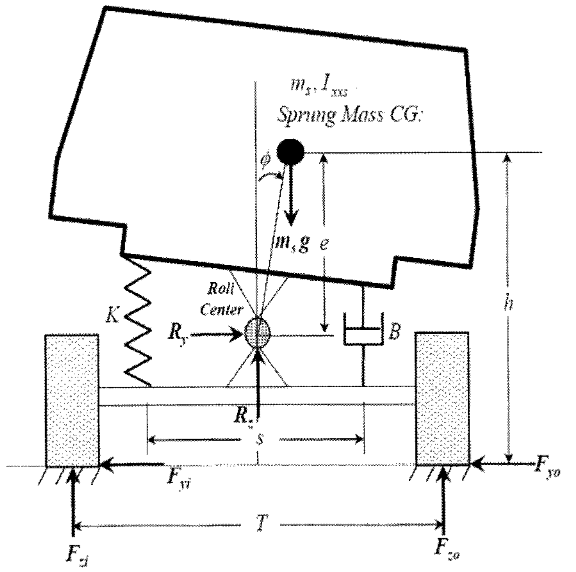

Figure 6. Vehicle roll model.

5. EXPERIMENTAL VALIDATION

This section describes the validation of the nonlinear vehicle model developed in Matlab/Simulink against available experimental results NHTSA phase IV reports (Forkenbrock, 2002). The unknown vehicle parameters are tuned by trial and error method. Tuning is achieved by fixing these unknown parameters with a base value initially and then adjusting their values for the best matching results. Thereafter these parameters are held fixed for rest of the simulations.

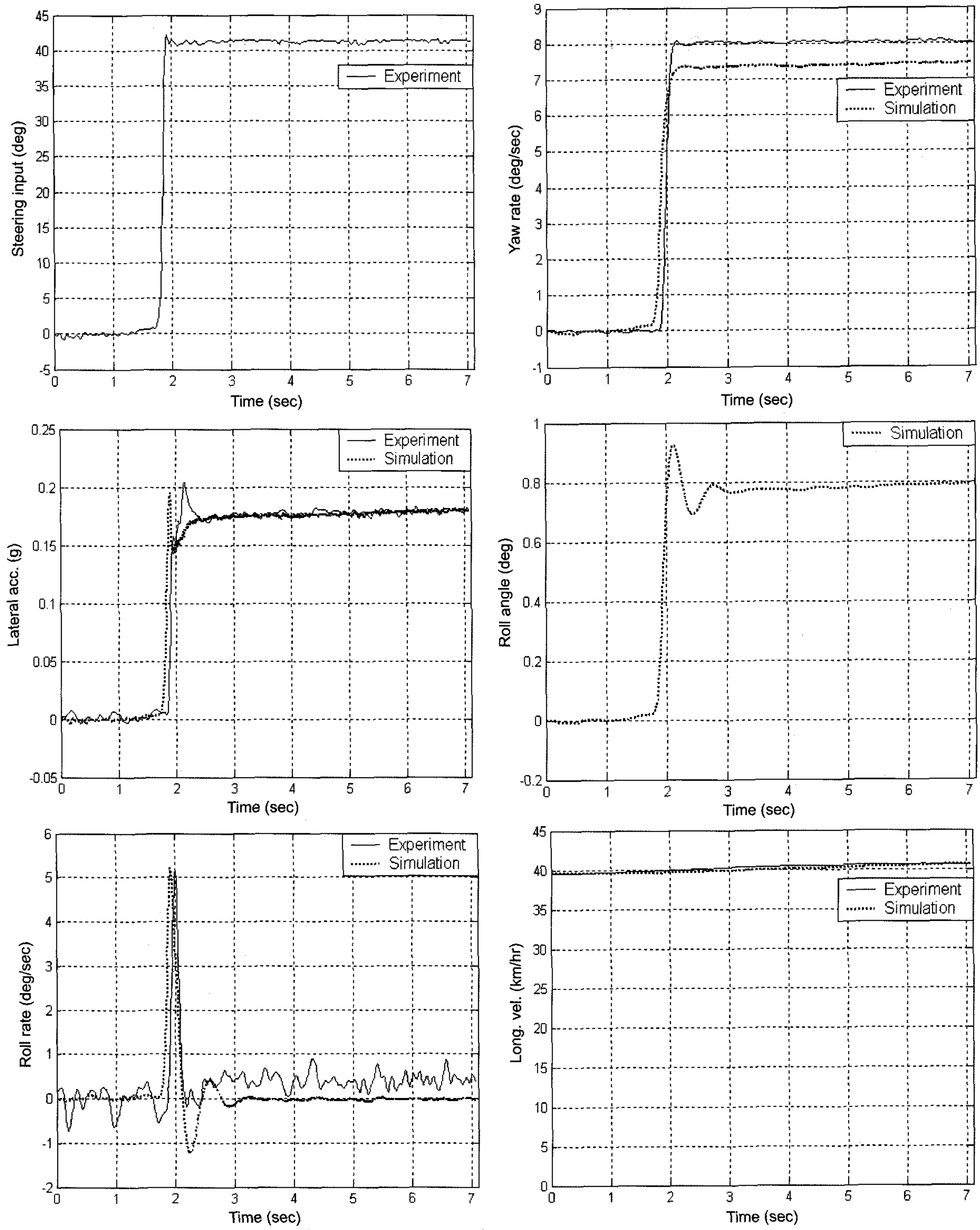

Figure 7. Validation results for 0.2 g J-turn maneuver (Ford Taurus GL, 1994).

5.1. Validation against J-Turn Maneuver

Figure 7 shows the comparison between experimental and simulation results for 1994 Ford Taurus GL passenger car. The steer input used was J-turn at a constant velocity of 39.6 km/hr capable of generating a maximum lateral acceleration of 0.2 g. The vehicle dynamic characteristics like lateral acceleration, roll rate and longitudinal velocity are in good agreement with the experimental results. The yaw rate prediction of the vehicle is an important characteristic for model validation. The yaw rate predicted by the simulation is slightly lower which may be due to measurement errors during experiments. Due to unavailable experimental data roll angle could not be compared.

5.2. Validation against Fishhook Maneuver

The non-linear vehicle model is validated against available

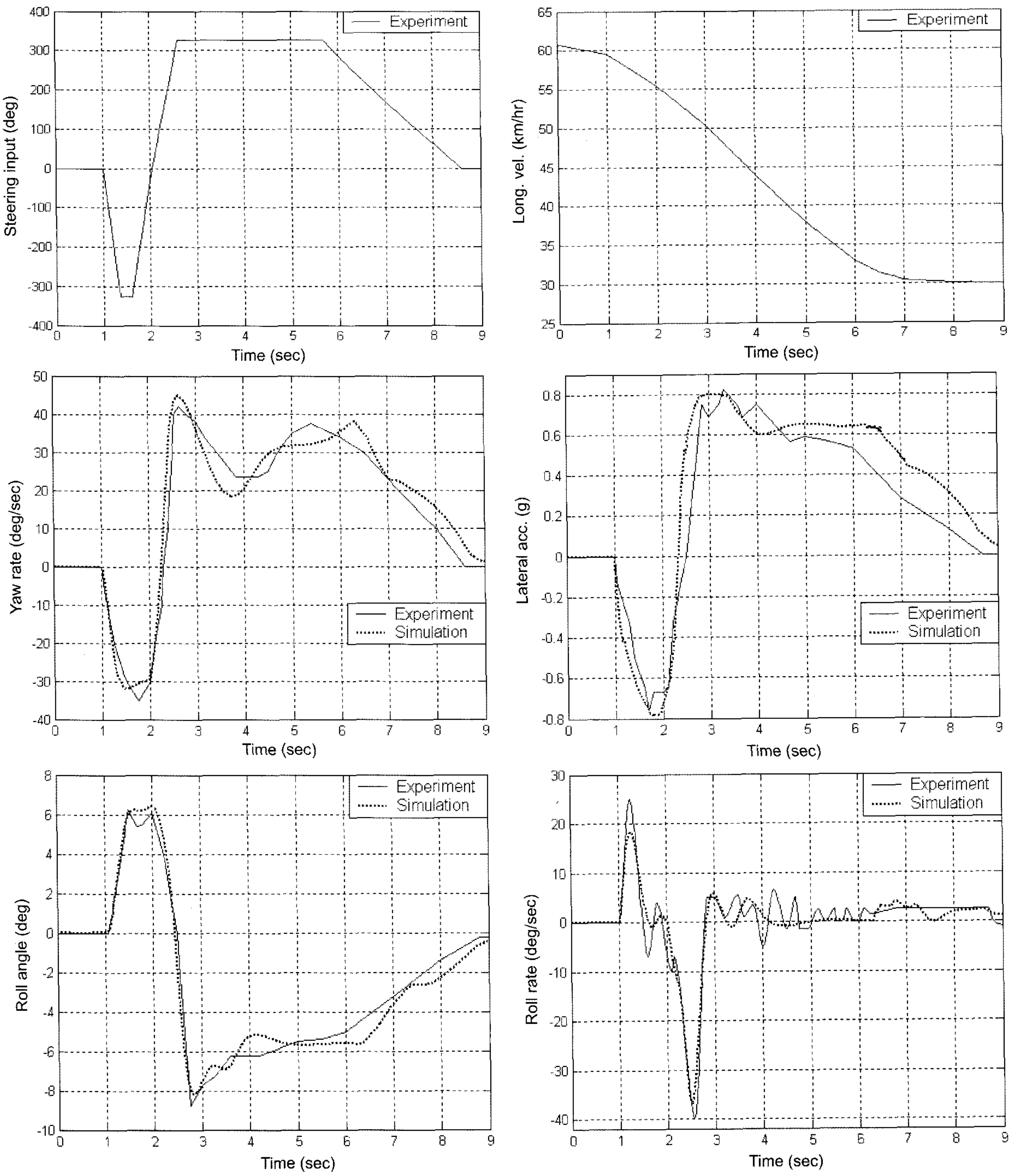

Figure 8. Validation results for fishhook 1a maneuver (Chevy Blazer RRR, 2001).

experimental results of Chevy Blazer Reduced Rollover Resistance (RRR) Configuration for Fishhook 1a steering input. The maximum steering angle used is around ±326 degrees capable of generating 0.8 g at a speed of 60.83 km/hr. The velocity profile as available from NHTSA phase IV report is provided as input. The entrance velocity is around 60.83 km/hr. The suspension parameters and distance between roll axis and the Center of Gravity are again tuned for this SUV configuration till the results are in good agreement. Figure 8 shows that the yaw rate and roll angle are in good agreement with the NHTSA test results. The roll rate characteristic is quite similar to that available from NHTSA phase IV results with a maximum of 20 deg/sec and a minimum of –40 deg/sec. There

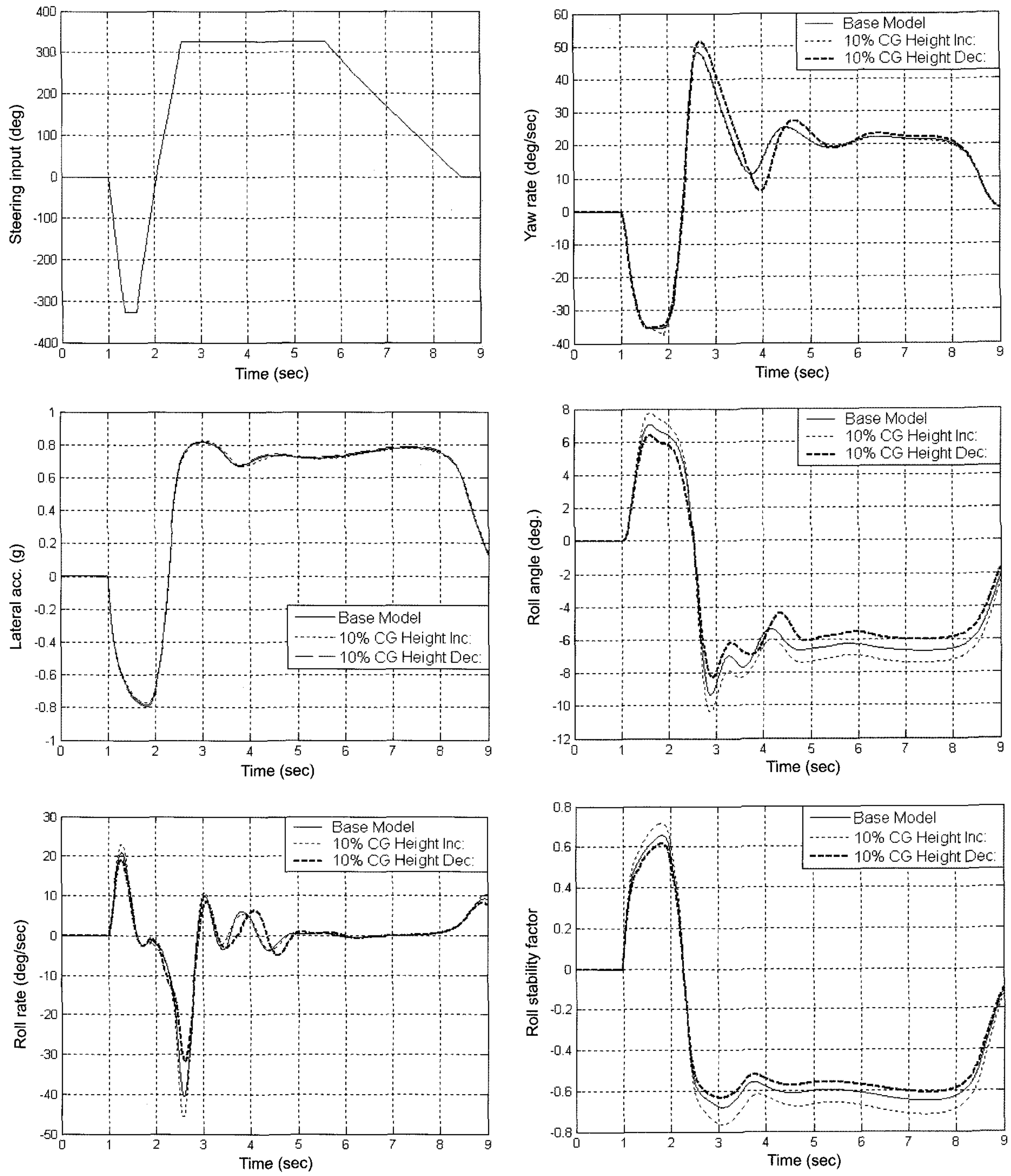

Figure 9. Vehicle characteristics due to CG height variation.

exists small discrepancy in lateral acceleration characteristics after 5 sec. Thus validation of the nonlinear vehicle model is performed.

6. SIMULATION METHODOLOGY

The Chevy Blazer Nominal configuration is selected as the base SUV configuration for all parameter sensitivity studies. The vehicle parameters are varied one at a time within chosen limits and simulations are carried out for Fishhook 1a steering maneuver. The simulation is done for no braking and no wind condition at a speed of 60.83 km/hr. The combined plot of yaw rate, lateral acceleration, roll angle and roll rate are analyzed. To evaluate dynamic rollover propensity the following stability criteria are also used.

6.1. Roll Stability Factor (RSF)

The Roll Stability Factor (Kim, 2005) is defined as the ratio of the difference between the sum of right side wheel loads and the sum of the left side wheel loads to the sum of all the wheel loads.

$$RSF=\frac{\sum_{i=1}^{2}\{(F_{ZR})_i-(F_{ZL})_i\}}{\sum_{i=1}^{2}\{(F_{ZR})_i+(F_{ZL})_i\}} \tag{2}$$

When the vertical load is balanced the RSF attains zero value denoting stable balanced condition. During rollover phenomena the vehicle experiences a lateral weight shift from inner wheels to the outer wheels. At Two Wheel Lift Off (TWLO) condition the load on inner wheels diminishes to zero and RSF attains unity. RSF can take any value between +1 and −1. Thus RSF indicates the measure of instability and the direction of vehicle roll.

6.2. Two Wheel Lift Off Velocity

The actual rollover phenomenon starts with TWLO accompanied by a lateral shift in CG. Once TWLO occurs, it becomes very difficult for the vehicle to attain a stable condition. The minimum longitudinal vehicle velocity at which the TWLO phenomena can be observed is defined as Two Wheel Lift Off Velocity (TWLV) (Whitehead, 2004). Even though NHTSA does not consider minor TWLO less than 2 inches as potential enough to cause rollover events, this paper considers minor TWLO for evaluating dynamic rollover tendency. The TWLO event is determined by observing the normal tire force diagram on each side of the vehicle and Roll Stability Factor plot.

7. RESULT AND ANALYSIS

7.1. Effect of CG Height

To study the influence of CG height on rollover propensity, CG height and distance between CG and roll axis are varied simultaneously in equal amounts keeping all the other vehicle parameters same as that of the base model. The vehicle roll sensitivity due to 10% variation in CG height is analyzed for Fishhook 1a steering maneuver. Figure 9 shows that the maximum roll angle and roll rate is experienced by 10% increase in CG height, followed by base model and then by 10% decreased model. The lateral acceleration for 10% CG increase is found to be slightly higher indicating instability. The lateral acceleration and yaw rate characteristics of the base model and 10% CG height increased model are found to be overlapping. Thus 10% variation in CG height does not show an appreciable difference in yaw rate and lateral acceleration characteristics. The maximum value for RSF is observed for 10% increase in CG height followed by the base model. 10% decrease in CG height show lower RSF value indicating higher roll stability.

A series of simulations are performed to correlate TWLV and CG height. Figure 10 shows that decrease in

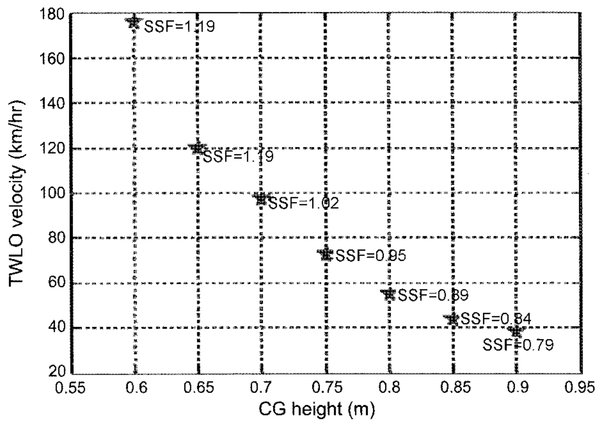

Figure 10. Two wheel lift off velocity vs. CG height.

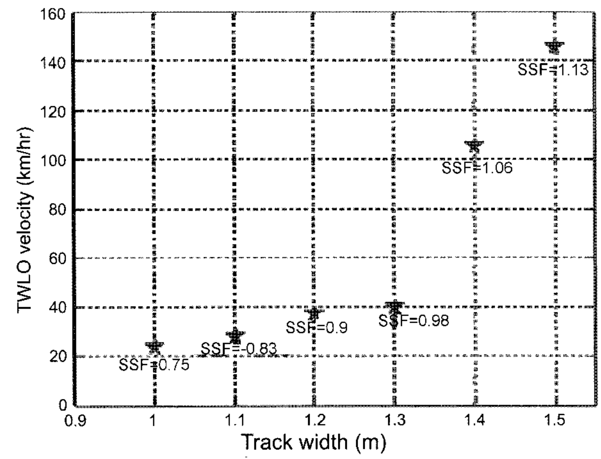

Figure 11. Two wheel lift off velocity vs. track width.

CG height results in higher TWLV and enhanced roll stability. As Static Stability Factor is indirectly proportional to CG height, the vehicle stability decreases with increase in CG height and vice versa. Figure 10 confirms that the base SUV configuration chosen with a CG height of 0.6629 m is quite stable. The vehicles with CG height in the range of 0.55 m to 0.65 m offer high roll stability.

7.2. Effect of Track Width

The influence of track width on rollover propensity is

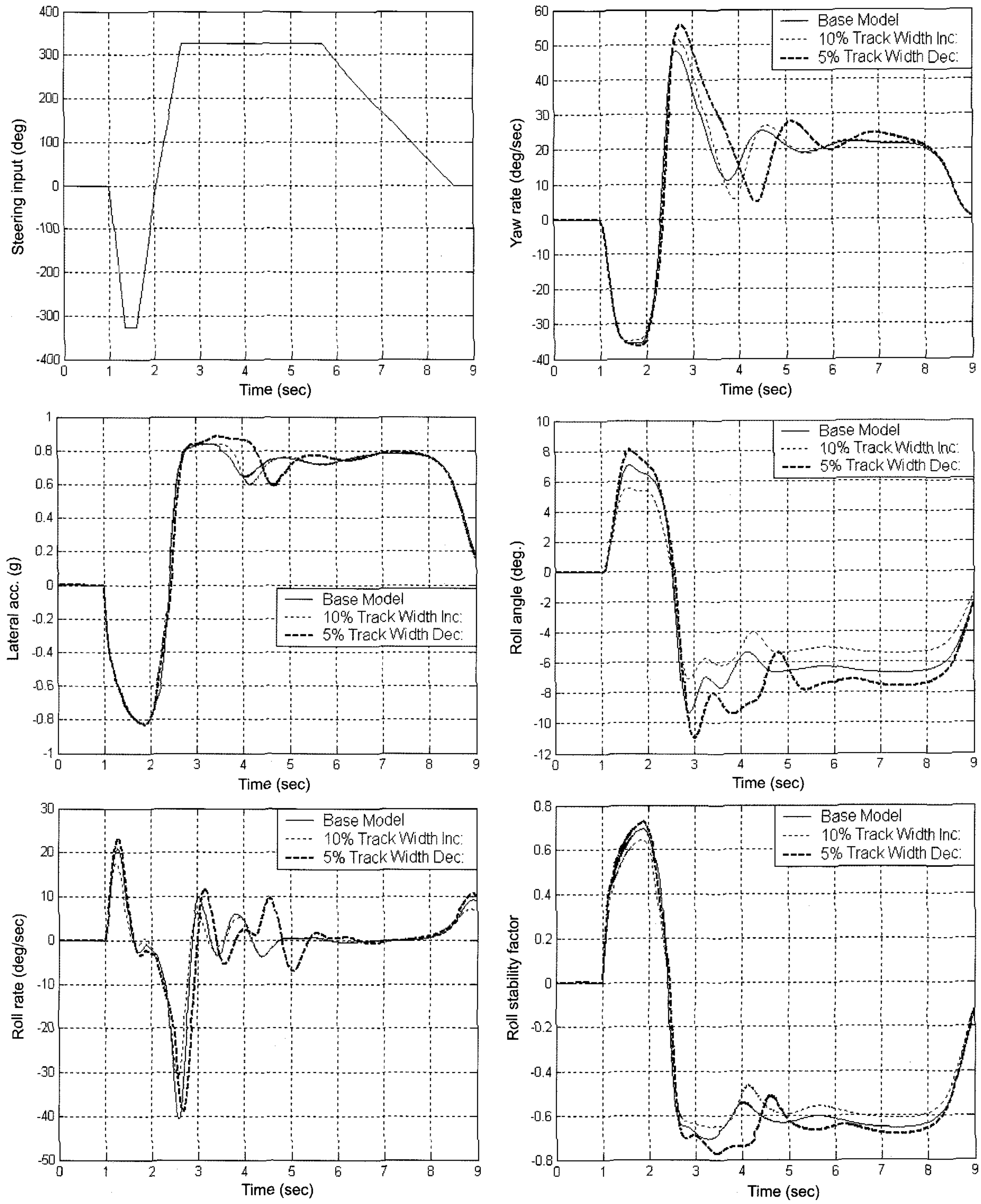

Figure 12. Vehicle characteristics due to track width variation.

analyzed by changing track width and distance between suspensions simultaneously in equal amounts. TWLV for different track width varied models are shown in Figure 11. It is evident that wider track widths offer higher rollover threshold and more stability to the vehicle. The vehicle with track width less than 1.4 m shows higher tendency to rollover. As SSF is directly proportional to track width, wider track width ensures better roll stability. The simulations are done for Fishhook 1a steering input. Figure 12 shows that 5% decrease in track width shows higher yaw rate, lateral acceleration, roll angle and roll rate values compared to the base model and that of 10%

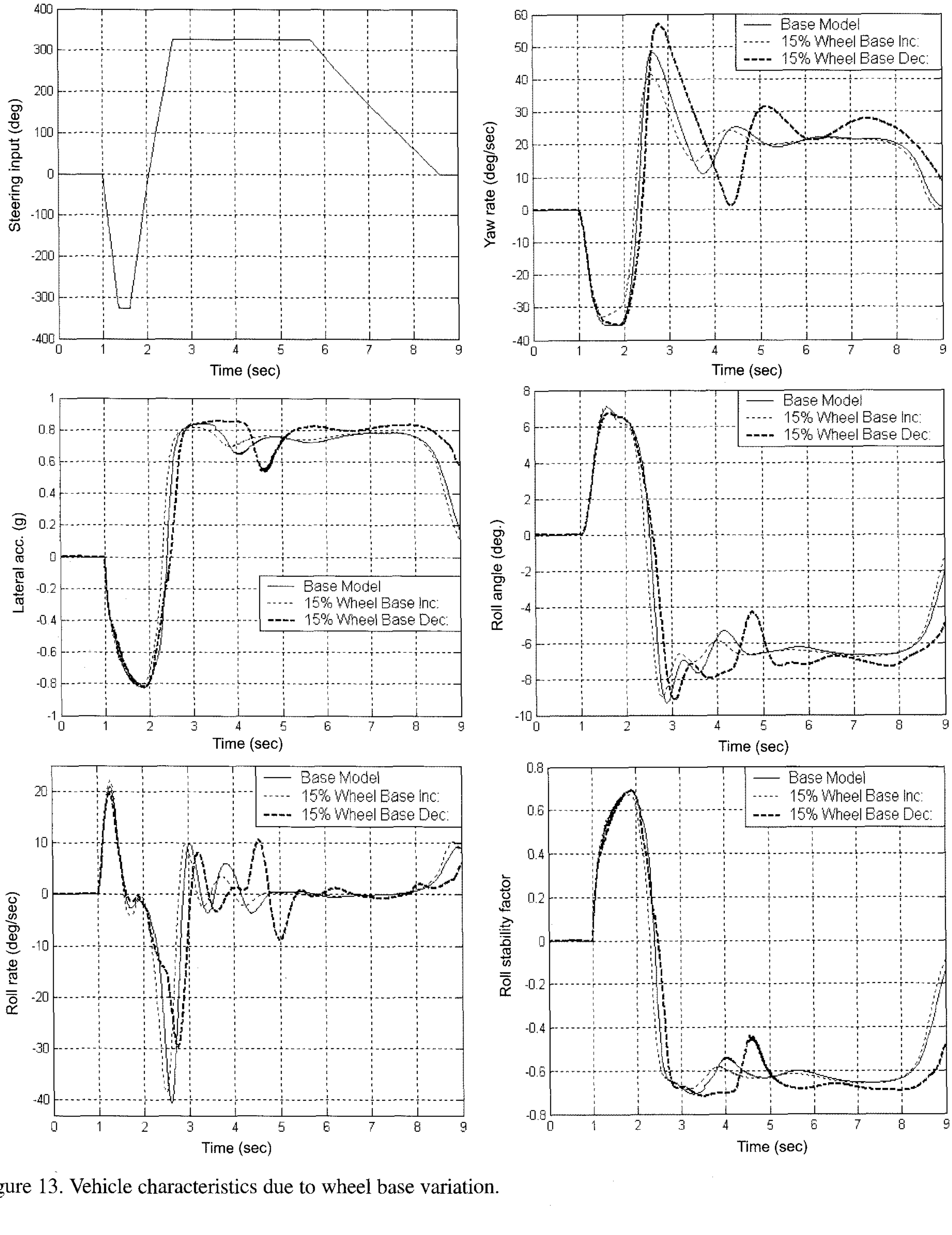

Figure 13. Vehicle characteristics due to wheel base variation.

track width increased model. With decrease in track width the vehicle tends to yaw, oversteer and finally becomes unstable. The RSF plot shows similar trend of increasing rollover propensity with narrow track widths.

7.3. Effect of Wheel Base

The effect of wheel base on rollover propensity is studied by varying the wheel base alone. With the variation in wheel base simultaneous variation in horizontal distance between CG and front axle is provided.

Figure 13 indicates the vehicle characteristics for 15% variation of wheel base. The maximum yaw rate and lateral acceleration are experienced for 15% decrease in wheel base. The roll angle plot does not show considerable change for 15% variation in wheel base. The RSF plot indicates that the 15% decrease in Wheel base has higher value compared to the other two models indicating instability with decrease in wheel base. The Two Wheel Lift Off phenomena is not observed for 15% variation in wheel base. It will be interesting to study the influence of wheel base for higher variations and severe steering maneuvers at higher speeds. These plots indicate that the vehicles having same SSF may have different dynamic rollover propensities depending on other critical vehicle parameter.

8. CONCLUSION

A nonlinear vehicle model is modeled in Matlab/Simulink and validated against available test results of Ford Taurus GL passenger car and Chevy Blazer Sports Utility Vehicle. The vehicle model shows good agreement with experimental results and is also capable of capturing important vehicle dynamic characteristics during rollover phenomena. The developed nonlinear vehicle model is used to study the vehicle parameter sensitivity of Sports Utility Vehicles on dynamic rollover. This simulation method proves to be a cost effective, time saving and reliable rollover predictor. This also offers flexibility in changing various vehicle parameters that may not be practically realizable in case of actual field tests.

This paper mainly investigates the sensitivity of vehicle roll behavior due to variation of three vehicle parameters – CG height, track width and wheel base. The result proves that track width and vehicle CG height are primary factors determining the rollover propensity of a vehicle. The relationship of Two Wheel Lift Off velocity with CG height and Track width is also obtained. The study on wheel base within the specified variation shows that longer wheel base ensures better roll stability. The paper also tries to correlate the dynamic rollover propensity with Roll Stability Factor and TWLO velocity. RSF is helpful in predicting TWLO occurrence and the direction of vehicle roll. RSF measurement can play a vital role in developing air bag deployment schemes, rollover warning systems, rollover prevention and other stability enhancement algorithms. The TWLO velocity plot can aid in finding safety limits of vehicle parameters under investigation during SUV chassis design.

From this analysis it is understood that there are many other critical vehicle parameters like vehicle mass, weight distribution on each axle, tire model and specification, tire pressure, inertia, suspension and damping parameters which play a significant role in triggering actual rollover event during severe steering maneuvers. The future research will be aimed at developing an ADAMS vehicle model which will be compared with the non linear vehicle model and the available experimental results to study the influence of different vehicle parameters on rollover propensity.

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APPENDIX-1

Longitudinal Motion

$$\sum F_x:\ m(\dot{U}-rV)=\sum_{i=1}^{4} X_i \tag{3}$$

Lateral Motion

$$\sum F_y:\ m(\dot{V}-rU)=\sum_{i=1}^{4} Y_i \tag{4}$$

where X_i and Y_i are defined as

$$X_i=F_{xi}\cdot\cos(\delta_i)-F_{yi}\cdot\sin(\delta_i) \tag{5}$$

$$Y_i=F_{xi}\cdot\sin(\delta_i)+F_{yi}\cdot\cos(\delta_i) \tag{6}$$

U and V account for the longitudinal and lateral velocities in body-fixed coordinates respectively and r is the yaw velocity of the vehicle. The following coordinate changes are required to convert the body fixed velocities into inertial velocities.

Longitudinal:

$$U_{inertial}=U\cos(\psi)-V\sin(\psi) \tag{7}$$

Lateral:

$$V_{inertial}=U\sin(\psi)+V\cos(\psi) \tag{8}$$

The vehicle accelerations are defined as

Longitudinal:

$$A_x=\dot{U}-rV \tag{9}$$

Lateral:

$$A_y=\dot{V}-rU \tag{10}$$

Yaw Motion

$$\sum M_z{:}I_z\dot{r}=a(Y_1+Y_2)-b(Y_3+Y_4)$$
$$+\frac{T}{2}(X_2+X_4)-\frac{T}{2}(X_1+X_3) \tag{11}$$

Roll Motion

The sprung mass roll angle can be obtained by force analysis as shown in Figure 6 (Whitehead, 2004). The vehicle frame is assumed to be rigid and lateral weight transfer is through the suspension. The spring and damping characteristics are assumed to be linear.

$$I_{xxs}\ddot{\phi}=R_z\cdot e\cdot\sin\phi+R_y\cdot e\cdot\cos\phi$$
$$-\frac{s}{2}(F_{Bo}+F_{Ko})-\frac{s}{2}(F_{Bi}+F_{Ki}) \tag{12}$$

where $(F_{Bo}+F_{Ko})$ and $(F_{Bi}+F_{Ki})$ represent outer and inner wheel suspension forces respectively. The spring forces are defined as:

$$F_{Ki}=F_{Ko}=K\cdot z_{KB} \tag{13}$$

$$\text{where, } Z_{KB}=\frac{1}{2}[d_1-d_1\cos\phi]+\frac{s}{2}\sin\phi \tag{14}$$

The damping forces are defined as:

$$F_{Bi}=F_{Bo}=B\cdot\dot{z}_{KB} \tag{15}$$

$$\text{where, } \dot{Z}_{KB}=\frac{1}{2}\dot{\phi}[d_1\sin\phi-s\cos\phi] \tag{16}$$

The reaction force from sprung mass is calculated as

$$R_z=m_s(\ddot{Z}+g) \tag{17}$$

$$\ddot{Z}=e\cdot\ddot{\phi}\cdot\sin\phi+e\cdot\dot{\phi}^2\cdot\cos\phi \tag{18}$$

The reaction force from unsprung mass is calculated as

$$R_y=m\cdot a_y+F_{yi}+F_{yo} \tag{19}$$

The normal load on each wheel is a function of vehicle's static weight, pitch weight transfer and the roll weight transfer associated with lateral acceleration and roll angle as shown below:

$$F_{zRF}=mg\frac{b}{2L}-mA_x\frac{h}{2L}-\frac{K_{f\phi}}{K_\phi}\left(\frac{mA_yh+mge\phi}{T}\right) \tag{20}$$

$$F_{zLF}=mg\frac{b}{2L}-mA_x\frac{h}{2L}+\frac{K_{f\phi}}{K_\phi}\left(\frac{mA_yh+mge\phi}{T}\right) \tag{21}$$

$$F_{zRR}=mg\frac{b}{2L}+mA_x\frac{h}{2L}-\frac{K_{r\phi}}{K_\phi}\left(\frac{mA_yh+mge\phi}{T}\right) \tag{22}$$

$$F_{zLR}=mg\frac{b}{2L}+mA_x\frac{h}{2L}+\frac{K_{r\phi}}{K_\phi}\left(\frac{mA_yh+mge\phi}{T}\right) \tag{23}$$

APPENDIX-2

Table 1. Vehicle data used for this analysis (Forkenbrock *et al.*, 2002; Whitehead *et al.*, 2004).

VEHICLE PARAMETERS	CHEVY BLAZER		FORD TAURUS
	NOMINAL	RRR	
Total vehicle mass (kg)	1884.254	1965.902	1542
Total sprung mass (kg)	1695.829	1769.312	1356
Total unsprung mass (kg)	188.425	196.59	186
Roll inertia (kg-m^2)	743.252	761.834	670
Yaw inertia (kg-m^2)	3775.939	3779.715	2786
Inertia product (kg-m^2)	222	222	166
Wheel base (m)	2.718	2.718	2.69
Track width (m)	1.4225	1.4225	1.5494
CG height (m)	0.6629	0.70104	0.55
Distance of CG from front axle (m)	1.216	1.216	0.92
Distance of CG from rear axle (m)	1.502	1.502	1.77
Steering ratio	18.5	18.5	16
Wheel rolling radius (m)	0.329	0.329	0.329
TUNED PARAMETERS			
Distance between roll axis and CG (m)	0.454	0.454	0.454
Distance between springs or dampers (m)	1.2	1.2	1.4
Spring stiffness (N/m)	80000	80000	72500
Damping coefficient (Ns/m)	5500	5500	4500