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Simulation Study of Oscillatory Vehicle Roll Behavior During Fishhook Maneuvers

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ABSTRACT

Sustained roll oscillations were observed while performing the National Highway Traffic Safety Administration (NHTSA) fishhook (FH) test on several vehicles. This phenomenon has also been observed on several manufacturers' high center of gravity (CG) vehicles with both solid and independent rear suspensions. Roll oscillation can be accompanied by non-convergent yaw, heave, and pitch. A study was initiated to quantify the influence of vehicle inertia, suspension, powertrain, and tire characteristics on the FH vehicle response. Design of Experiments (DOE) methods were used to quantify the main effects and the significant interactions between vehicle design variables. The CarSim program was used to simulate the vehicle dynamics and the iSIGHT program was used to automate the DOE analysis.

Summary findings show that tire lateral force high slip behavior influences yaw instability. This alone is not sufficient for developing diverging roll oscillations. Additionally, results show that suspension jounce travel and bumper rate, in conjunction with tire overturning moment and non-positive tire cornering stiffness, influence yaw, roll, and pitch stability. Simulation results suggest that the primary cause of roll oscillations is the transfer of some energy from the longitudinal mode into the roll and heave modes. This effect can also be influenced by other factors like the distance between the CG and the roll axis, yaw-roll cross product of inertia, for example. Also described is optimization of FH performance (minimization of wheel lift and roll oscillations) with respect to some suspension characteristics.

INTRODUCTION

NHTSA, in conjunction with the Alliance of Automobile Manufacturers, developed the Dynamic Maneuvering Rollover Test or Fishhook Maneuver to provide a dynamic assessment of rollover resistance. NHTSA's final rule was published October 14, 2003. Performance in this test along with Static Stability Factor (SSF) determines the Rollover Resistance Rating that contributes to NHTSA's New Car Assessment Program (NCAP) Vehicle Ratings. SSF is the ratio of vehicle average half-track to the CG height with driver only.

The Fishhook Maneuver consists of a large-amplitude, high-rate, steer input in one direction, held until the vehicle roll rate transitions through zero, followed by a large-amplitude, high-rate, counter-steer (i.e., steer in opposite direction) and held for 3 seconds, and finally a return to zero steering-wheel angle. The amplitude of the steer is found by running NHTSA's Slowly Increasing Steer test prior to the FH Maneuver to find the steer angle corresponding to 0.3 g of lateral acceleration. The steering amplitude is 6.5 times this value. Depending on results with this steer amplitude, a supplemental procedure that uses 5.5 times the 0.3 g steer angle is also performed. To improve results consistency and since the rate of steer is 720 degrees per second, this test is run with a steering robot. The maneuver entry speed, or speed just before the initial steer input, ranges between 56 and 88 km/h. The throttle is released at the start of the maneuver. The default procedure and all subsequent testing are complete when "tip-up" occurs at any speed below 76 km/h. Tip-up is defined as two inches (50 mm) of simultaneous two-wheel lift. The value of wheel tip-up and the slope of the envelope of the oscillatory roll response are metrics of interest. The phenomenon of sustained roll oscillations has been observed on several manufacturers' high CG vehicles with both independent and non-independent rear suspensions.

This paper describes the results of the sensitivity study of Fishhook response metrics to variations in chassis kinematic and compliance, tire, and mass characteristics. Although Electronic Stability Control (ESC) is currently accepted as the most effective auxiliary rollover mitigation method, proper chassis design can reduce the need for an intrusive ESC algorithm to pass the rollover NCAP.

This paper is organized as follows. The vehicle dynamics model and DOE analysis procedure are described. This is followed by the DOE results describing the main effects and significant interactions. The paper closes with conclusions.

VEHICLE DYNAMICS ANALYSIS

The main analysis tools are CarSim, for simulating the vehicle dynamics and iSIGHT, for automating the overall analysis and displaying results.

BASELINE CARSIM MODELING

A baseline CarSim model was built using suspension compliances test data at the NHTSA-specified five passenger FH test loading condition plus

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outriggers. This experimental test vehicle has independent front and rear suspensions. The tire was tested for the range of vertical loads, slip and camber angles observed in the FH test. The resulting tire test force and moment data was processed to generate data suitable for the CarSim tire model. Figs. 1 and 2, respectively, compare the CarSim and test roll velocity response and wheel tip-up results for an entry speed of 76 km/h with identical steer input. This figure shows that this CarSim model correlates well for this steer input and vehicle speed.

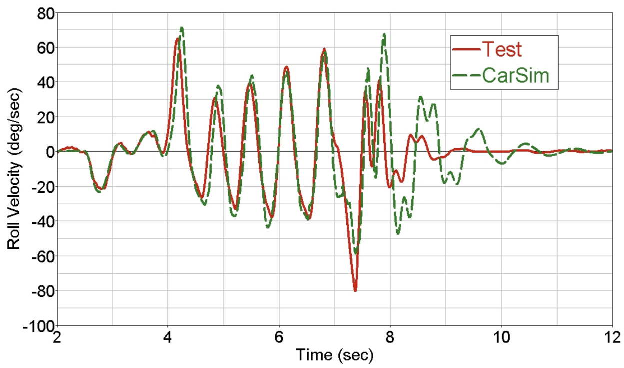

Fig. 1. Model correlation: Roll Velocity

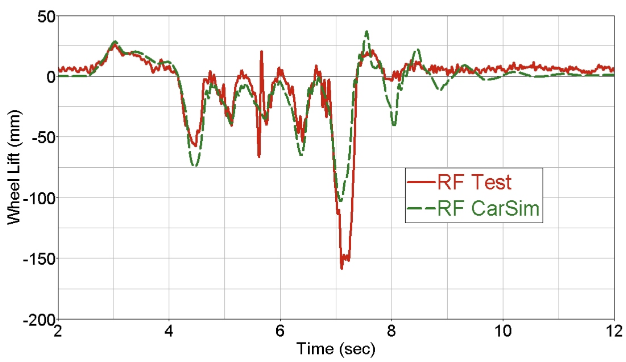

Fig. 2. Model correlation: Wheel Lift

Simulation facilitates the study of the influence of vehicle parameters on FH performance. FH test results are highly variable particularly due to sensitivity to surface friction coefficient. Eliminating the effects of this variation permits a more straightforward interpretation of the results.

The FH maneuver is a severe maneuver with a steering reversal which occurs when the roll rate transitions through zero. This maneuver is designed to excite resonances in the vehicle dynamics. The steering wheel angle input is large enough for the vehicle side slip angles to reach 90 degrees (without ESC). Tire cornering stiffnesses (i.e., slope of lateral force verses tire slip) at these slide slip angles can often be non-negative, promoting vehicle spin out. Since the sprung mass yaw, roll, pitch, and heave (i.e., vertical) motions are coupled, divergent oscillations in the roll, pitch, and heave directions can occur. These large vehicle motions result in large tire patch force variation.

A specially developed post-processing tool was used for the FH test or simulation results. Output includes metrics that describe the steering wheel amplitude, yaw and roll rates, and lateral acceleration first and second peaks, maximum wheel lift, frequency content, sideslip angle peaks, and yaw and roll rate decay constants and linear slopes. Fig. 3 illustrates the concept of the envelope slope metric: a Hilbert transform of a signal is applied during the counter steer dwell, followed by a linear fit to the resulting envelope curve.

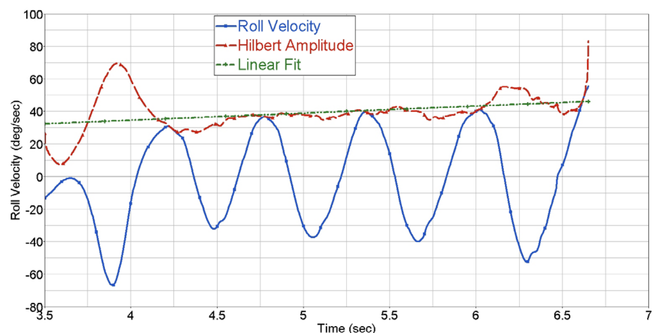

Fig. 3. Roll Velocity Envelope Slope (5.4 deg/sec^2)

Fig. 4 presents an example of CarSim time histories of different vehicle characteristics on one plot for an entry speed of 72 km/h. ISO sign convention is used for all response variables, i.e. X positive forward, Y positive left, and Z positive up. At entry speed of 72 km/h and higher, roll, pitch, yaw, and heave oscillations start to become divergent and the vehicle eventually spins out.

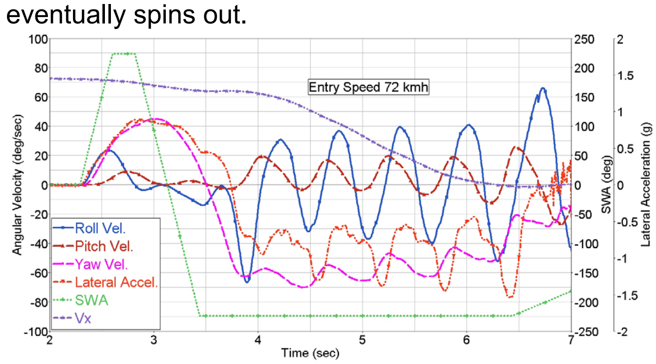

Fig. 4. Time Histories (Entry Speed of 72 km/h)

Figs. 5 and 6 show the wheel lift and sprung mass angular velocity envelope slope oscillation metrics for a range of entry speeds. Both figures show a definite increase in value for entry speeds beyond 72 km/h. Large amplitude oscillations, which are also evident for speeds beyond 72 km/h, diminish after 84 km/h, with the exception of the roll mode.

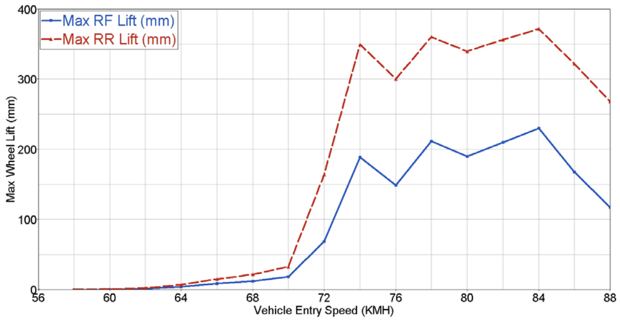

Fig. 5. Maximum Wheel Lift

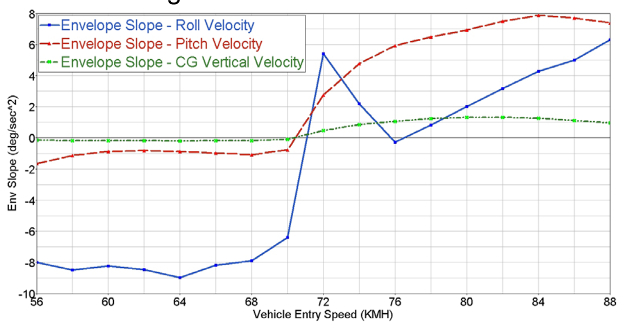

Fig. 6. Sprung Mass Angular Velocity Envelope Slopes.

DOE ANALYSIS

The iSIGHT program controls the execution of, and the flow of data between, CarSim and the Matlab post processor. iSIGHT uses *design variables* to define the *simulation model parameters* for CarSim. The analysis procedure within iSIGHT (e.g. main effects, Pareto plots, Latin Hypercube, etc.) determines the values of design variables and the number of runs needed.

DOE VARIABLES

Sensitivity of key elements of FH performance matrix to variation of suspension parameters, vehicle mass/inertia, tire characteristics, powertrain parameters is studied using iSIGHT and CarSim. Design parameters include:

Suspension

- Jounce travel
- Auxiliary roll moment (stabilizer bar)
- Damping
- Lateral force deflection
- Lateral force deflection steer
- Lateral force deflection camber
- Ride steer
- Static toe
- Roll center height

Steering

- Four wheel steer

Sprung mass

- Roll inertia
- Yaw inertia
- Yaw-roll product of inertia
- CG height
- CG lateral offset
- Weight distribution

Tire

- Peak coefficient
- Overturning moment
- Camber stiffness
- Longitudinal relaxation length / lag distance
- Lateral relaxation length / lag distance

Powertrain

- Coast down rate with engine braking

Roll center position is determined by the wheel center lateral position and camber angle versus jounce curves. These curves can be parameterized using quadratic functions. Linear coefficients determine roll center position whereas quadratic coefficients determine roll center movement with trim or roll. By varying these coefficients sensitivity to roll center position and migration can be analyzed.

The CarSim solver internally uses the curve: suspension spring force versus suspension compression/extension to define suspension ride rate and consequently ride travel. Such force includes spring contribution (mostly linear) and jounce bumper / rebound bumper contribution. An increase or decrease of wheel travel was achieved by moving the jounce bumper representation (nonlinear part).

MODEL PARAMETRIZATION

CarSim vehicle dynamics models are defined with both single parameters and tables of parameters. Examples of the former include sprung mass CG position and the roll inertia. These types of parameters are continuous in nature, and therefore fit well into iSIGHT's DOE analysis procedures. Many of the model parameters described above are defined by tables of data. Some of these tables (e.g. damping) can be easily generated using continuous parameters. Other tables require significant user intervention to construct, and thus, only limited numbers get generated. For this reason, these types of parameters are discrete. Examples include the tables that describe the tire cornering forces and the relationship between suspension force and jounce travel.

DOE RESULTS

This paper limits discussion to only some of the key main effects and full factorials for front and rear suspension jounce travel, the tire characteristics, roll and inertial axes, and suspension rear view geometry. Only tip-up and roll oscillation metrics will be examined.

The top eight parameters influencing fishhook performance, as measured by these two metrics, explain about eighty percent of two-wheel lift and sixty percent of roll oscillation. The significant parameters include front and rear jounce travel, front and rear auxiliary roll stiffness, tire lateral force peak, tire overturning moment due to slip, sprung mass center of gravity height, and four wheel steer.

Jounce Travel (Ride Rate Curve).

The main effects analysis shows increased front jounce travel causes a reduction in right front (RF) and right rear (RR) wheel lift. Note that the right side of the vehicle is on the inside of the turn. An increase in rear jounce travel of 15 mm or more causes a reduction in RR and RF wheel lift (see Fig. 7).

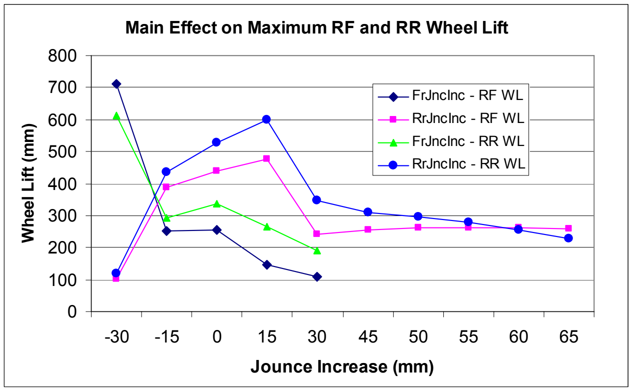

Fig. 7. Jounce Travel Increase: Main Effect on Wheel Lift

Recall that, one of the primary factors influencing roll oscillations is suspension ride rate nonlinearity due to jounce bumper and/or metal-to-metal contact. Engaging this high rate can cause a large variation in tire vertical force. Another consequence of suspension ride stiffness nonlinearity is coupling of the sprung mass CG vertical and roll motions. Limited wheel travel can result in more sprung mass CG vertical lift.

If the wheel jounce travel is unlimited, roll oscillations are convergent. However this causes the maximum roll angle to increase by almost 2 degrees, front wheel lift to decrease from 230 mm to 129 mm, and rear wheel lift to decrease from 371 mm to 183. Observed maximum jounce with unlimited travel is 156 mm for the front, whereas baseline jounce is limited to 103 mm; and maximum jounce for the rear is about 110 mm, whereas baseline jounce is limited to 63 mm. The full factorial study for the front (between -30 mm and 45 mm) and rear (between -30 mm and 65 mm) wheel travel increase showed that the most influential sources for RF wheel lift are front wheel travel increase followed by interaction between front and rear wheel travel increase. Similarly, for RR wheel lift, the most influential sources are rear wheel travel increase followed by interaction between front and rear wheel travel increase.

The effects of front and rear wheel travel change are not "additive". There is an optimal combination of front and rear wheel travel increase for improving FH performance. The largest improvement in wheel lift is seen in the cases where front jounce decrease is combined with some rear jounce increase. The best cases for roll velocity envelope slope are when the rear jounce travel increase is 15 or 50 mm.

Auxiliary Roll Moment

Sensitivity to auxiliary roll moment (i.e. roll stabilizer bar stiffness) is studied by uniformly scaling the corresponding CarSim curves. The TLLTD (tire lateral load transfer distribution) is defined as:

$$TLLTD = \frac{\Delta F_Z^{fr}}{\Delta F_Z^{fr} + \Delta F_Z^{rr}}$$

Where $\Delta F_Z^{fr} = abs(F_Z^{LF} - F_Z^{RF})$ is the front tire load transfer. Rear tire load transfer is defined analogously. In the simulation, TLLTD is computed at 0.3g from the slowly increasing steer maneuver. Separate scaling of the front or rear auxiliary roll stiffnesses results causes the roll gradient and TLLTD to also change.

There is a nonlinear relationship between TLLTD and the front and rear auxiliary roll moments. Scaling down the front auxiliary moment causes the RR wheel lift to slightly decrease and the RF wheel to show little change. It also causes the roll rate envelope slope to increase, degrading performance. The consequent reduction in TLLTD results in the vehicle spinning out at higher speeds. The effect of rear auxiliary roll moment is more pronounced. Both wheel lift and envelope slope are reduced considerably with increased rear roll stiffness.

To study the effect of TLLTD while maintaining total roll stiffness, front and rear scale factors were changed while simultaneously keeping the total auxiliary roll stiffness constant. This method, while easier to execute, does result in some variation in roll gradient. Figs. 8 through 10 demonstrate the effect of TLLTD on wheel lift. Decreasing the front bias of TLLTD improves wheel lift performance and reduces envelope slopes. However at this level of total roll stiffness roll oscillations are still divergent.

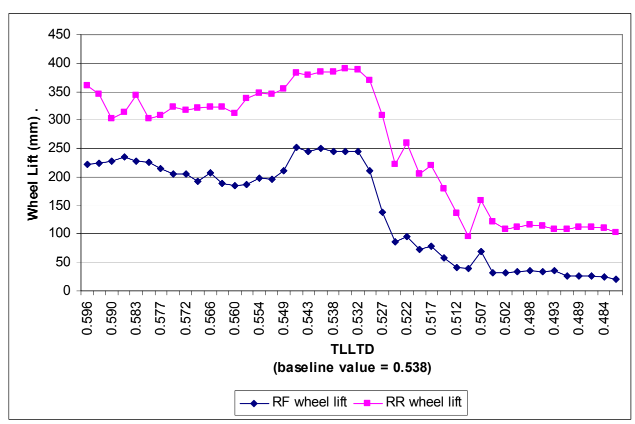

Fig. 8. Maximum Wheel Lift vs. TLLTD

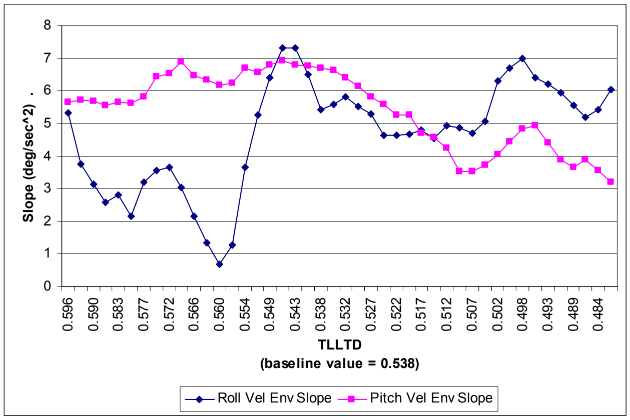

Fig. 9. Maximum Envelope Slopes vs. TLLTD

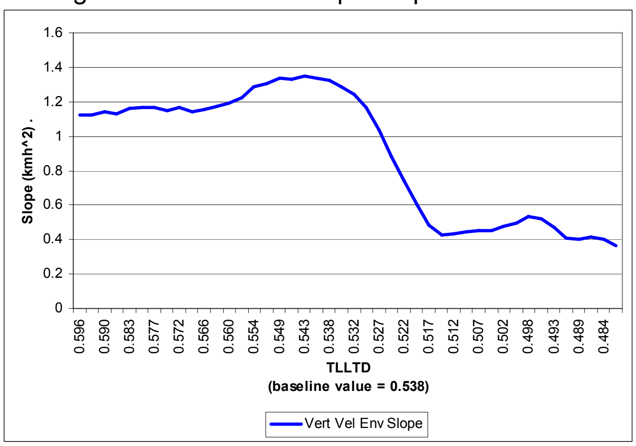

Fig. 10. Maximum Envelope Slope vs. TLLTD

Tires: Lateral Force (Fy).

Five properties are used to describe the lateral force as a function of slip angle for a particular vertical load: cornering stiffness, peak force, slip value of peak lateral force, slide force (i.e., force at 50 degrees of slip), and a curvature term. A relationship, based on the Magic Formula, was used to describe the lateral force as a function of slip angle (for details see [5]).

Design variables in the DOE scale the peak force, slip value of peak force, and slide force. Tables of data need to be generated for each combination of DOE parameters. Thus the tire lateral force versus slip angle characteristics were studied by means of three parameters:

- peak value (+/- 10%)
- slip at peak (+/- 20%)
- slide value (+/- 10%)

The slip angle at peak lateral force is varied while attempting to maintain essentially the same cornering coefficient or lateral force at 1 deg slip. Fig. 11 compares the curves at a vertical load of 4940 N. Note that the change in lateral force as a function of camber, and overturning moment (M_x) and aligning moments (M_z) as a function of slip and camber were not studied in this case.

Results of the full-factorial tire lateral force characteristic study are shown in Figs. 12 and 13. Note that two of the Fy Peak/Fy Slide/Slip-at-Fy Peak configurations (0.9/1.1/0.8 and 0.9/1.1/1.2) could not be generated because there is no systematic way to move the nonexistent peak. For simplicity of executing DOE these cases were assumed to be identical to the case 0.9/1.1/1.0.

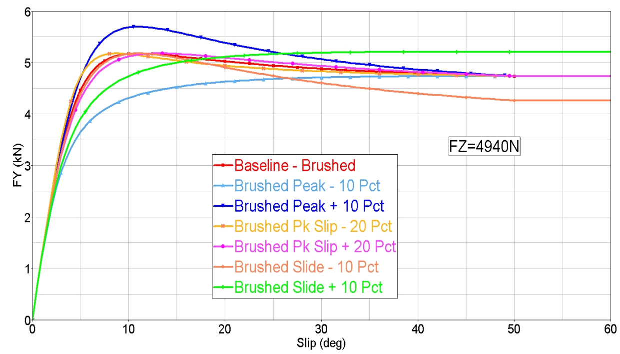

Fig. 11. Tire Lateral Force vs. Slip Angle: DOE parameters

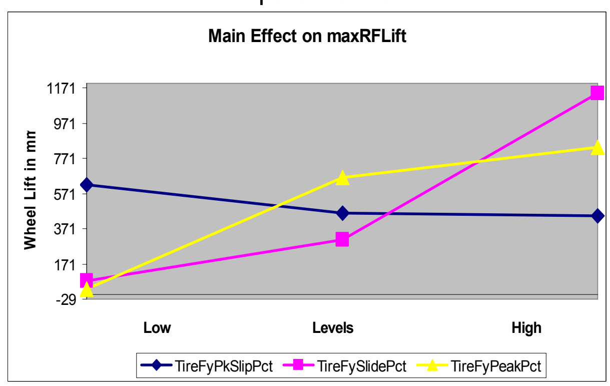

Fig. 12. Tire Fy Full-Factorial Study – Main Effect for RF Wheel Lift

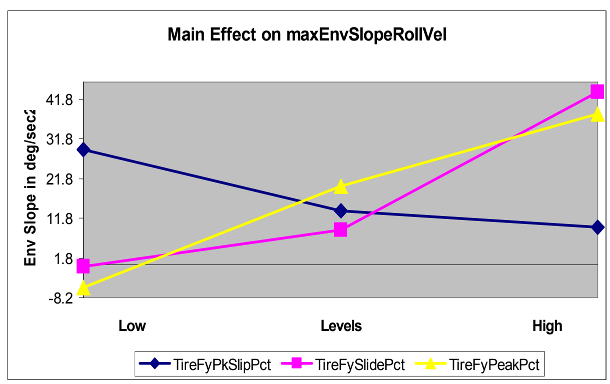

Fig. 13. Tire Fy Full-Factorial Study – Main Effect for Roll Velocity Envelope Slope

By changing the tire Fy peak value, the negative slope of the portion of curve Fy(slip) between peak and slide, is either increased in magnitude or decreased. Changing the Fy peak strongly affects the nature of the roll oscillations; making them either divergent or convergent. The value of slip angle at peak determines the start of the region of the negative slope of the curve Fy(slip). This variable influences the critical speed for roll oscillation.

According to the Pareto diagrams, the most influential factors are peak and slide value. As peak values increase (-10%, 0, +10%), roll oscillation becomes divergent and wheel lift monotonically increases. The effect of the slide value is similar: higher values degrade FH performance. The higher the slip angle at peak force, the more gradually Fy builds up as a function of slip, pushing the negative slope region toward higher slip angles. Higher values for slip at peak are beneficial for FH, reducing wheel lift and envelope slopes. However, by itself this factor was not found to make roll oscillations convergent.

Tires: Overturning Moment (Mx). The effect of overturning moment was studied by simultaneously scaling (between 0 and 2) two curves: overturning moment versus slip angle and overturning moment versus inclination angle. Zero overturning moment would represent an effective infinite tire lateral stiffness which would reduce apparent track width. This improves dynamic T/2H and reduces jacking forces.

Reducing overturning moment decreases wheel lift and envelope slopes leading eventually to convergent roll oscillations and less than 50 mm of wheel lift. Increasing overturning moment degrades roll oscillation and wheel lift in a nonlinear manner. For example, between 1.25 and 1.3 there is a bifurcation point and the vehicle roll becomes divergent for a scaling factor more than this value. Therefore, FH performance is quite sensitive to overturning moment.

Sprung mass CG height

Height of the CG of the sprung mass (HCGSU) is a first order factor for rollover stability since it directly impacts static stability factor (i.e., T/2H). FH performance depends nonlinearly on HCGSU (see Figs. 14 and 15). The baseline value of HCGSU is 627 mm. Wheel lift decreases almost linearly with decreasing HCGSU until about 600 mm, and then it stays constant. Roll oscillations become convergent for CG height's less than about 592 mm.

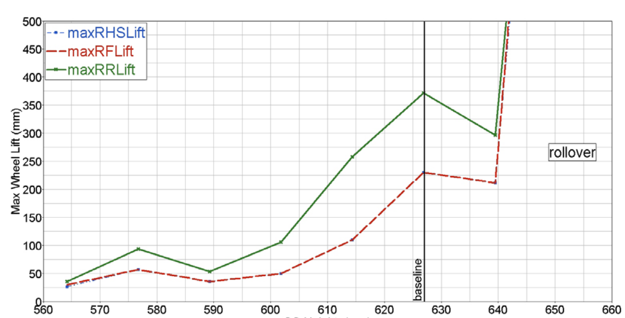

Fig. 14. Maximum Wheel Lift vs. Sprung Mass CG Height.

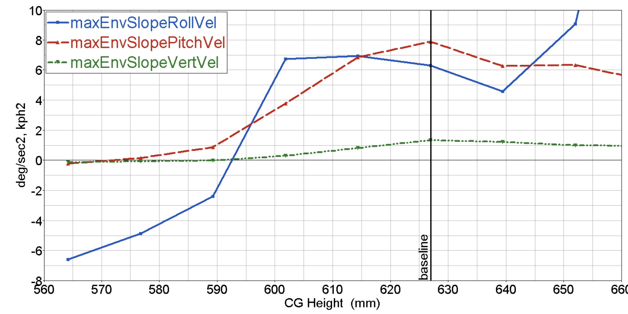

Fig. 15. Maximum Envelope Slopes vs. Sprung Mass CG Height.

Four Wheel Steer.

The effect of rear steer was analyzed by modifying the relationship between rear steer gain and speed. First, case investigated is a constant rear steer gain. The analysis shows that in-phase rear steer is very beneficial for FH, whereas out-of-phase steer leads to a spin-out situation. With a 14.6% rear to front steer ratio, roll oscillations are convergent and wheels barely lift the ground. This requires about 2 degree of rear steer. Increasing the fraction of rear steer to 22 percent of the front steer results in convergent roll oscillations, little wheel lift, and less than 50 degree of vehicle sideslip. Maximum rear steer for this case is about 3.4 degrees. Further increasing the fraction of rear steer to 30 percent of front steer results in tire slip angles that do not exceed 13 degrees, roll oscillations that are convergent, and wheels that do not lift.

Next, the curve rear steer gain versus speed was generated based on linear bicycle model and condition of zero slip in steady state turning. In this case, wheel lift is very small, angular oscillations are convergent, and tire slip angles do not exceed 30 degrees.

Interrelation between roll axis and inertia axis

To better understand the influence of relative position of the roll axis with respect to longitudinal inertia axis on FH performance, a two parameter full-factorial study was performed.

1. Rear RCH by means of b_2 - linear term in quadratic fit of wheel center lateral position versus jounce curve (25 levels)
2. Yaw-roll product of inertia Ixz (7 levels: -100.0, -69.0, 0.0, 50.0, 100.0, 143, and 200.0 kg-m^2)

The baseline yaw-roll product of inertia is 143 km-m^2. Positive sign indicates an inertial axis that tilts down toward the front. The analysis is based on 175 runs. Pareto diagrams revealed that the biggest percentage of the total effect is the cross term RCH by Ixz or the relationship between the inertia axis and roll axis. On the other hand, envelope slope is mostly dominated by rear RCH.

Figs. 16 through 18 show wheel lift and envelope slope as a function of the angle between (principal) inertia axis and roll axis. Sign convention is for a positive angle if the inertial axis is rising more than the roll axis toward the front of the vehicle. In the baseline configuration this angle is about -3.7 degree. It follows from these plots that, in general, FH performance depends not only on relative angle but on both Ixz and rear RCH. It is seen that with more positive relative angle, wheel lift and roll oscillation are reduced. However there is a saturation point, so that beyond a certain angle, performance does not change much.

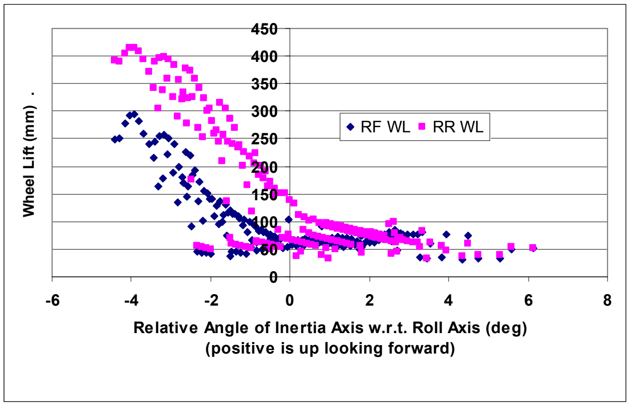

Fig. 16. Max Wheel Lift vs. Angle Between Inertia and Roll Axis

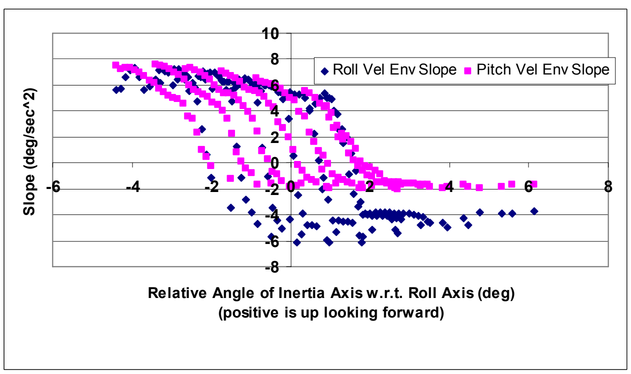

Fig. 17. Max Roll Velocity and Pitch Velocity Envelope Slope vs. Angle Between Inertia and Roll Axis

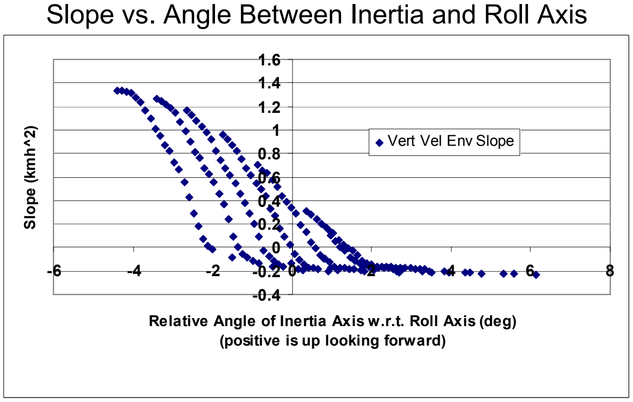

Fig. 18. Max Vertical Velocity Envelope Slope vs. Angle Between Inertia and Roll Axis

Figs. 16 through 18 scatter plot all the data, however the curves corresponding to different values of Ixz can be easily visualized. For all these figures, curves of constant Ixz move to the left as Ixz is increased.

In summary, increasing the relative angle helps FH performance; however there is an optimal point beyond which there is no improvement. The optimal angle is not necessarily zero (perfect alignment of two axes), for high values of Ixz this angle could be negative, for low values of Ixz it could be positive. For the test vehicle configuration, aligning the inertial axis nearer the roll axis is beneficial.

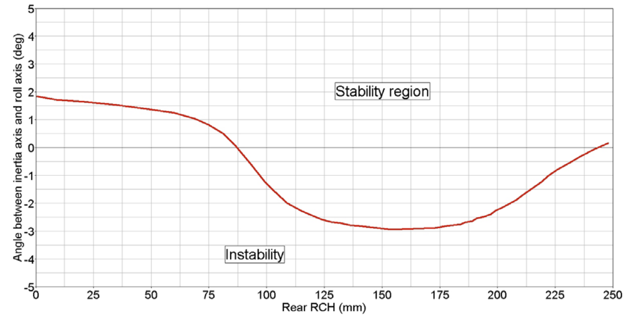

Fig. 19. Stability region

Fig. 19 shows stability region in the plane of rear RCH and relative angle of inertia axis with respect to roll axis.

Suspension Rear View Kinematics Optimization

Integration of CarSim and iSIGHT allows performing optimization using different optimization criteria and in different design space. By parameterizing the front and rear wheel center lateral position and camber versus vertical displacement curves, an optimization could be run to improve FH performance. The parameters are defined as follows: a_1 is the linear and a_2 the quadratic coefficient for the roll camber versus trim curve, while b_1 and b_2 are the linear and quadratic coefficients for the wheel center lateral position versus trim. Two solutions are presented here; they differ by the range of design parameters. Optimal solutions are defined in the following table:

Parameter	Baseline	Optimal1	Optimal2
a_1 front (deg/mm^2)	0.00021	0.000158	0.000166
a_2 front (deg/mm^2)	0.0023	0.001725	0.001452
b_1 rear (deg/mm)	-0.03334	-0.04168	-0.057865
b_2 rear (deg/mm)	0.1829	0.137175	0.171942

For the Optimal 1 solution, maximum wheel lift is about 43 mm and maximum roll velocity envelope slope is about -3 deg/sec^2. For the Optimal 2 solution, maximum wheel lift is about 38 mm and maximum roll velocity envelope slope is about -7 deg/sec^2. Note, that for both optimal solutions rear roll center is raised (RCH is 40 mm and 87 mm respectively) and front roll center movement (vertical and lateral) with ride and roll is reduced in magnitude.

CONCLUSION

Note that the conclusions reached are based on a vehicle without ESC. ESC is a very effective rollover mitigation technology that essentially supersedes the impact of the vehicle factors studied. However, as stated in the introduction, applying good basic design as studied here may allow the ESC algorithm to be less intrusive.

Note that CarSim can be effectively used to study Sine with Dwell (SWD) maneuver. NHTSA has identified the 0.7 Hz SWD as a good maneuver for evaluating the lateral stability and responsiveness of ESC-equipped vehicles. Hardware-in-the-loop testing is often used to test the electronic control unit.

The following conclusions apply specifically to the vehicle modeled.

1. Non-convergent roll oscillations observed in the FH maneuver appear to be energized by longitudinal energy from vehicle deceleration being transferred into the roll and heave modes. Oscillatory instability can occur in yaw, roll, and heave. Tire peak and slide characteristics, suspension ride rate nonlinearity (roll-heave coupling), and tire overturning moment are the primary factors causing the oscillation.
2. Of the parameters studied, rear wheel steer (in-phase) is the most effective way to stabilize FH performance.
3. Sprung mass CG height is a first order factor for rollover stability since it impacts static stability factor.
4. Suspension ride nonlinearity greatly affects roll-heave coupling. Limited wheel travel forces the sprung mass CG to lift more during severe roll motion and can cause a large variation in tire vertical force reducing the stability region. However, there appears to exist a particular combination of front and rear jounce travel for improved FH performance.
5. The distance between the sprung mass CG height and roll axis is one of the primary factors in roll stability, since it represents the lever arm for the inertial force due to lateral acceleration.
6. Raising the rear roll center has a beneficial effect on FH performance. When rear RCH exceeds 125 mm, roll oscillations become convergent and wheel lift does not exceed 50 mm.
7. The interrelation between Ixx (the longitudinal principal inertia axis aligned closest to the horizontal plane) and the roll axis is an important factor in FH performance. A principal inertia axis that tends to be less declining toward the front of the vehicle than the roll axis improves FH performance. However, there is an optimal angle (not necessarily zero) between inertia and roll axes beyond which there is no improvement. This angle is dependent upon the specific values of Ixz, Ixx, and Izz. For the test vehicle configuration, aligning the principle inertia axis nearer the roll axis is still beneficial because it makes these axes closer to the optimal angle.
8. The most influential tire properties for FH performance are tire lateral force peak and slide coefficients, and overturning moment. A smaller peak lateral force value improves FH performance. Reducing the magnitude of the tire overturning moment decreases wheel lift and envelope slope, and depending on the vehicle configuration, may lead to convergent roll oscillations. Lower values of tire slide improves wheel lift and envelope slope, but it also increases vehicle tendency to spin out indicating a degradation in the overall handling.
9. Reduction in TLLTD towards 50 percent front bias improves FH performance.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

FH	Fishhook
NHTSA	National Highway Traffic Safety Administration
NCAP	New Car Assessment Program
ESC	Electronic Stability Control
DOE	Design Of Experiments
VHF	Vehicle Handling Facility
RCH	Roll Center Height
CG	Center of Gravity
HCGSU	Sprung mass CG height
YCGSU	Sprung mass CG lateral offset
RF	Right Front
RR	Right Rear
rr	Rear
fr	Front
TLLTD	Tire Lateral Load Transfer Distribution