THE INVESTIGATION AND MODELLING OF CORKSCREW ROLLOVERS

Nick Harle
Phil Glyn-Davies
Crashworthiness Laboratory
Millbrook Proving Ground Ltd
Millbrook, Beds, MK45 2JQ, England
Document Number: 157

ABSTRACT

Two increasingly important areas of vehicle crashworthiness are the study and modelling of rollovers, as next generation vehicles are being designed with rollover countermeasures This paper describes the corkscrew rollover phenomena, and the analysis techniques used to model it. In the ADAC corkscrew rollover test, two wheels of the test vehicle strike a 6 m long ramp which is 1.19 m high at the launch point. After travelling up the ramp, typically at around 80 km/h, the vehicle rolls onto its roof. Papers exist on the modelling of lateral rollovers, but only one reference is known to the authors on modelling a corkscrew rollover [1].

INTRODUCTION

Initially, a detailed description of the corkscrew event is given, using suspension displacement data along with vehicle roll and pitch velocities to explain the event. The main load path into the vehicle body is through the vehicle suspension.

Secondly modelling of the corkscrew event is described. Suspension behaviour at its stops can be difficult to determine, there can be contact between the suspension arms and the ramp, and there can be wheel/wheel arch contact. A bi-linear spring and bi-linear damper suspension model was used along with a representation of the wheel and tyre. An explicit timestepping procedure was used to solve the resulting equations of motion. This was implemented using the Excel spreadsheet.

Thirdly, a comparison of spreadsheet prediction and test results for corkscrew rollovers carried out with a mid-size vehicle, a convertible, and a sports utility vehicle is presented. The model requires suspension and inertial parameters. Where a parameter is estimated, its value can be varied to define corridors of expected vehicle response. Alternately, data from one test can be used to tune unknown vehicle parameters required for a similar vehicle at a different test speed.

Finally the spreadsheet results are discussed. Direct suspension/ramp contact is not modelled, and hence the results are most accurate for the sports utility vehicle where this effect is minimised.

THE CORKSCREW PHENOMENEN

The whole corkscrew event can be broken up into three phases: 1) the ramp phase, 2) the airborne phase, and 3) the ground/sliding phase. The key to understanding the first phase is a knowledge of the car suspension, and then access to two sets of data: 1) roll and pitch rates, and 2) suspension heights.

Vehicle Suspension

The mid-size and convertible vehicles used in the corkscrew rollover had a MacPherson Strut type suspension on the front wheels, with the damper placed inside the spring. A schematic of this is shown in Figure 1.

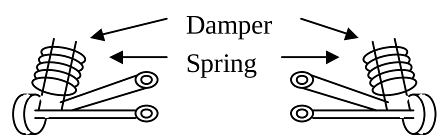

Figure 1. Schematic of the Mid-size and Convertible Vehicles Front Suspension

The important points concerning the front suspension are that the spring and damper are in parallel, and that when maximum travel on the suspension is reached, the spring and damper restrict movement.

The rear suspension had a spring and damper in parallel again, with a connecting anti-roll bar between both wheels. This time a stopper was positioned to restrict the maximum movement of the suspension springs. A diagram of the rear suspension is shown in Figure 2.

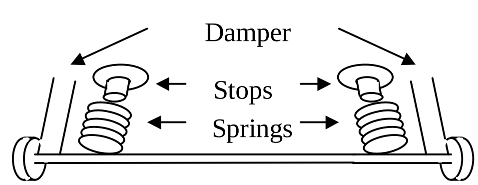

Figure 2. Schematic of the Mid-size and Convertible Vehicles Rear Suspension

Several points are to be noted. Firstly, the rear wheels move almost together as one unit, the stiffness of the anti-roll bar being the factor that determines how closely they act as one. Secondly, the maximum travel of the suspension is limited by the stops, not by the damper.

The sports utility vehicle had double wishbone suspension on the front and a five link arrangement at the rear. The front suspension stiffness was supplied through a torsion bar arrangement, and coil springs were used on the rear.

Suspension data, i.e. spring and damper properties, were supplied to the authors for the sports utility vehicle, but was unavailable for the mid-size and convertible vehicles.

Stage 1: The Ramp Phase

A sketch of the ramp used to roll the vehicle in a right hand corkscrew is shown in Figure 3.

Ramp 500 mm wide, left side of vehicle strikes ramp.
All dimensions in millimetres

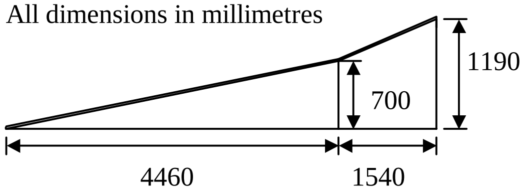

Figure 3. Ramp Used in Corkscrew Rollover Test

Figure 4 shows the pitch and roll rates for the convertible vehicle travelling up the ramp. These were gained from roll and pitch gyros placed on the vehicle. Also on the vehicle were string pots placed in the front left and rear right wheel arches to measure the suspension displacement. Superimposed onto the roll and pitch angular velocities are the vehicle suspension displacements as measured in the test. The vehicle leaves the ramp at about 0.28 seconds.

Before discussing the results one concept is worth bearing in mind. The suspension of the front left wheel, the leading wheel on the ramp, exerts a force on the car that gives positive roll and pitch velocity. The rear right wheel suspension exerts a force on the body that lowers the roll and pitch velocities. In essence the front left wheel aids the rolling and pitching motion, while the rear right wheel opposes them.

The front right and rear left wheels play little part in the process. As the vehicle pitches nose upwards, the front right wheel has little force on it. As the vehicle rolls moving the right sill downwards, the weight is taken off the rear left wheel.

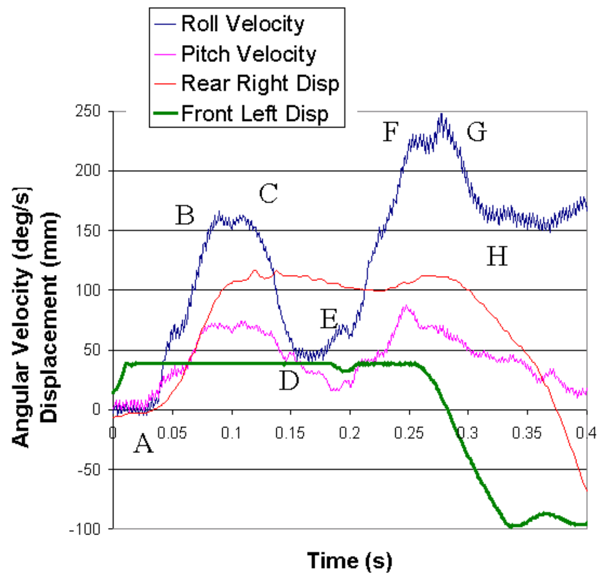

Figure 4. Vehicle Roll and Pitch Rates, & Suspension Displacement

The sequence of events as the vehicle travels up the ramp is:

1. Section A–B: Front left wheel hits the ramp, bottoms suspension giving high roll and pitch velocities
2. Section B–C: The rolling and pitching movement bottomed the rear right wheel suspension, halting the rise in roll and pitch velocity
3. Section C–D: The front left wheel, now moving broadly parallel to the ramp, moves off the suspension stop. Simultaneously, the high suspension force from the rear right wheel is present. The reduction in force from the front left wheel, and the high rear right suspension force, cause a sharp decrease in roll and pitch velocity.
4. Section D–E: A slight increase in roll rate, as the front suspension is not fully on its limit for all of this duration, giving a low roll and pitch rate. The rear right suspension is not resting heavily on its stop, which means there is nothing to oppose the small force increasing the roll velocity from the front left suspension mount
5. Section E–F: The front left wheel hits the second stage of the ramp, bottoming out the suspension, and giving high pitch and roll velocities.
6. Section F–G: The rear right suspension again moves onto its stop. This halts the rise in roll and pitch velocity.
7. Section G–H: The front left wheel leaves the ramp, removing the high force inducing roll and

pitch. The rear right wheel is momentarily in contact with the ground. This reduces the roll and pitch velocity.

To summarise the ramp stage, the vehicle seems to rock from the front left suspension point to the rear right and back again. When the front left suspension bottoms, the high resulting force causes both high roll and pitch velocities, both of these rotations increase the displacement on the rear right suspension mount. When the rear right suspension mount bottoms, it opposes the high rolling and pitching forces generated by the front left suspension mount bottoming out.

The rear right mount not only opposes the force from the front left but indeed overcomes it and produces a decrease in roll and pitch velocity. This may be from two effects. Firstly, after initially hitting the ramp, the front left wheel then begins to move parallel with the ramp, decreasing the force at that suspension mount. Secondly the force developed at the rear right suspension may be greater than that of the front left. Afterwards the rear suspension moves off its stops and then the whole process repeats, producing a second peak in the graph of roll and pitch velocities. What initiates the second rise in roll and pitch velocities is the front left wheel hitting the second, steeper, part of the ramp.

As far as modelling the corkscrew behaviour is concerned, several issues have now become apparent:
1. The vehicle suspension <u>before</u> it bottoms out has little effect on the roll and pitch velocities.
2. The main effect of the suspension spring and damper properties before they bottom, is to determine *when* the suspension bottoms, and so when the major forces at the suspension mounts are generated.
3. For the mid-size and convertible vehicles, the damper and spring at the end of their normal travel generate the front suspension force.
4. Again, for the mid-size and convertible vehicles, the rear suspension force is generated by suspension stops, not the damper.

While the major forces are transmitted to the vehicle body through the vehicle suspension, other load paths also exist. e.g all three vehicles put through the corkscrew test showed signs of direct suspension arm ramp contact.

Stage 2: The Airborne Phase

Once the vehicle's front left wheel has left the ramp, and the rear right wheel has cleared the ground, the vehicle is now airborne. The forces that acted on the vehicle body from the suspension, have now been removed. The major retardation force during this phase is likely to be the wind resistance acting on the vehicle. As will be discussed, the aerodynamic forces are likely to affect the vehicle's pitching motion more than its rolling motion.

Stage 3: The Ground Sliding Phase

With the vehicles nose-upward pitching motion, usually part of the rear structure strikes the ground first. Plastic deformation then occurs in the vehicle's roof structure, A, B and C Pillars. As the coefficient of friction is low between the vehicle roof and the Crash Laboratory floor, it slides, retaining much of its velocity. As plastic deformation occurs, only a complex model could capture the detailed pitching motion of the vehicle after it hits the ground. However, if it is assumed the vehicle slides keeping its velocity gained from the ramp, then an estimate of its trajectory can be obtained.

MODELLING THE CORKSCREW EVENT

The previous section has identified the important phenomena involved in a corkscrew rollover. A modelling strategy had then to be devised which could predict the vehicle roll and pitch angle up to the point of first ground contact. After writing the equations of motion, an explicit solution procedure was devised to solve them. This solution procedure was then implemented using the Excell spreadsheet. Various suspension and contact models were tried before a successful modelling strategy was found.

Firstly the general solution procedure will be described before the details of the suspension and ramp/ground contacts are discussed.

General Modelling Strategy

A nine degree of freedom model was found to give good correlation with the test results. Five degrees of freedom relate to the vehicle body, these are the car centre of gravity (c.g.) translation in three axes as well as roll and pitch. Four degrees of freedom relate to the vehicle wheels, these are translation movement of each wheel along its suspension path. The vehicle was assumed to start with a given initial velocity with its front left wheel at the base of the ramp. The basic solution procedure is then given below:

1. Assume full knowledge of vehicle c.g. displacement and velocities.

2. Allow the vehicle to travel forward one timestep with its current velocity starting from its current position.
3. Contacts: Calculate how far the vehicle wheels have travelled into the ramp or ground, then apply a force to remove the wheel from the surface it is penetrating.
4. Suspension Forces: Calculate the suspension forces exerted on the vehicle body.
5. Solve Equations of Motion: From the forces exerted on the car body calculate the c.g. acceleration in translation and rotation in roll and pitch.
6. New Velocities and Displacements: From the accelerations produced in step 5, calculate the translational and rotations velocities, along with the associated displacements and angles of rotation.
7. Iterate: Return back to step 1.

The above procedure can be implemented in a spreadsheet. Each iteration, steps 1 – 6, is carried out in a single row, with the columns in the rows carrying out the tasks involved in each steps. The procedure is "explicit" in that only data from the previous timestep is used to determine accelerations, velocities and displacement in the current timestep.

The aim of the model was to predict the roll and pitch angle of the vehicle as it hit the ground. Quantities such as car c.g velocity and displacement could easily be graphed against time, along with roll and pitch angles and angular velocities. To aid in determining when the model predicted the vehicle would contact the ground, several points were taken on the rear of the vehicle. Their height was then plotted against time. The final aid to data visualisation was to plot these rear points horizontal position against vertical position. This gave a cross-sectional view of the rear of the car as it rolled during the event. The inertial and geometric data required is shown in Table 1.

All of the inertial data was available for the mid-size and convertible vehicles in test condition. For the sports utility vehicle, only its mass and the c.g. longitudinal and lateral positions were available. However NHTSA has published a database of inertial properties, which included the five door version of this vehicle [2]. As the three door version was crash tested at Millbrook, the roll moment of inertia and c.g. height of that vehicle was used. The sports utility vehicle's pitch moment of inertia was estimated from experience gained from estimating other vehicle's moment of inertia in pitch.

Table 1.
Inertial & Geometric Data Required By Spreadsheet

Inertial Data	**Geometric Data**
Vehicle Mass	Wheel Base
Moment of Inertia in Roll	Wheel Track Front
Moment of Inertia in Pitch	Wheel Track Rear
Height of c.g.Above Axles	Longitudinal Position of Rear Points on Vehicle
Distance of c.g From Front Axle	Lateral Position of Rear Points on Vehicle
	Tyre Radius

Suspension and Contact Model

The suspension spring and damper, along with wheel mass and tyre stiffness are modelled. The schematic of this approach is shown below in Figure 5.

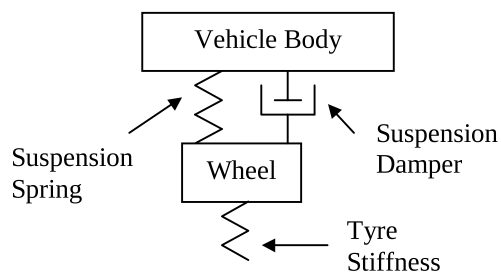

Figure 5. Suspension and Wheel Model

The tyre stiffness is modelled as the contact surface stiffness between the wheel and the ramp/ground. As the wheel digs into the ramp/ground, it is pushed out with the stiffness of the tyre. A given maximum travel was allowed before the wheel was made to move parallel with its contact surface. The wheel translation velocity has three components: 1. Translational motion along its suspension path 2. Angular velocity from the vehicle rolling about its c.g. 3. Angular velocity from the vehicle pitching about its c.g. The total vertical velocity of the wheel from these factors can be calculated along with the velocity it needs to travel parallel to its contact surface. The force the contact surface exerts when the wheel is at maximum compression, is the force required to accelerate the wheel so that in one timestep the wheel has the vertical velocity required to make it travel parallel to its contact surface.

Suspension Model – Mid-size and Convertible Vehicles As the suspension data was unavailable for these vehicles, it had to be estimated from 1) vehicle measurements, 2) test data and 3) a knowledge of suspension characteristics.

From the string pot measurements of the suspension displacement an estimate of the maximum normal suspension travel (d_t) can be made. Once the suspension reaches its maximum normal travel, there is a marked increase in the force resisting wheel movement, and hence an associated reduction in wheel travel. From Figure 4 an estimate of the maximum suspension travel could be made. The diagram of the suspension spring used is shown in Figure 6. Points to note concerning Figure 6 are:

1. Deflection is zeroed at ride height
2. Line 1 represents the end of the spring travel when the wheels have fully extended, deflection d_e in Figure 6, and has stiffness k_e. This was a very stiff value but was reduced after wheel instability occurred in the analysis (see following)
3. Line 2 has the suspension ride stiffness (k_s), up to deflection d_t in Figure 6.
4. Line 3 is the stiffness of the suspension stop (k_t)

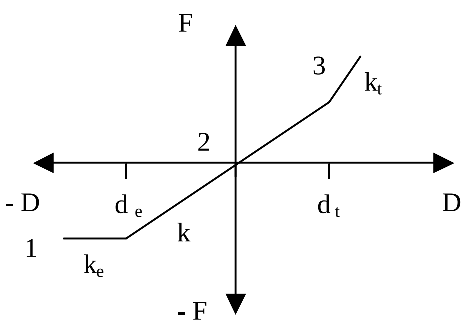

Figure 6. Suspension Spring Force Deflection Curve

A low stiffness value was used for the suspension spring at full extension, this was due to numerical instability. Normally a high value would be expected, as at full extension the suspension stops would prevent further travel. When the suspension applies a very large force to the wheel, this produces a very large instantaneous acceleration to the wheel (mass 35 kg). If this acceleration is applied for a long duration (with a large time step) large wheel velocity and displacements result. The solution is either to reduce the timestep or lower the stiffness of the spring. The latter action was chosen as the timestep of 1 ms kept the spreadsheet to a practical size, and the behaviour of the wheel once the suspension was fully extended did not have an important influence on the vehicle behaviour.

A bilinear damper was used for the front suspension, and a linear damper for the rear. As noted in the examination of the vehicle suspension, the damper aids in restricting the maximum travel in the front suspension. Hence one damping coefficient was used for the normal travel of the suspension, and another much larger value for travel beyond the normal range. As the stops restricted rear suspension travel, a linear damper was used for the rear suspension.

Parameters Used in Suspension Model – Mid-size and Convertible Vehicles The parameters required for the suspension model, and how they were derived, are shown in Table 2. Two points are worthy of further comment concerning the data in Table 2. These are the use of "Fundamentals of Vehicle Dynamics" by Thomas D. Gillespie [3,4], and the three "tuning" parameters. Gillespie deals with the analysis of suspensions, and gives sample parameters for various suspension components, and the relationships between these parameters. In the absence of knowledge of the damper properties, Gillespie's observations were used.

Having either measured or estimated all other parameters, there were three remaining which were still unknown. These were the coefficient of damping in the front suspension dampers beyond the normal suspension travel, and the stiffness of the front and rear suspension stops.

Determining Suspension Parameters – Mid-size and Convertible Vehicles The test with the convertible vehicle was conducted first. Hence the three unknown suspension parameters were determined by tuning the spreadsheet prediction to match the convertible test data.

The results after the tuning process are shown in Figure 7 to Figure 11. The roll and pitch velocities are shown in Figure 7, while the roll and pitch angles are shown in Figure 8. Figure 9 shows the suspension displacements, while vehicle c.g. velocity is given in Figure 10, and vehicle c.g displacement in Figure 11.

The spreadsheet values for the rear right suspension in Figure 9, do not fall to the same value as the test data due to the modelling of the rear suspension. As there is a connecting bar between the two rear wheels, the stiffness of the rear suspension, during normal travel, was given twice its estimated value. The stiffness of the suspension stopper was left unaltered. The connecting bar would allow the rear suspension to move as one during normal travel, but was not stiff enough to ensure both wheels severely compressed the stops. Because of the increased stiffness of the suspension spring, the wheel does not drop to the level of the same level of the test vehicle.

Table 2.
Suspension Parameters and Their Derivation

Suspension Data	Front	Rear
Spring Full Extension (d_e)	Measured from vehicle	Measured from vehicle
Spring travel to stop (d_t)	Estimated from test data	Estimated from test data
Spring Stiffness at Full Extension (k_e)	Numerical parameter	Numerical parameter
Suspension Ride Stiffness (k_s)	Estimated from vehicle measurements	Estimated from vehicle measurements
Stiffness of Suspension Stop (k_t)	Tuning parameter	Tuning parameter
Damper Coefficients		
Coefficient During Normal Suspension Travel	Estimated Using Gillespie [3]	Estimated Using Gillespie [3].
Coefficient At Stop	Tuning Parameter	N/A
Tyre Parameters		
Tyre Stiffness	Estimated Using Gillespie [4].	Estimated Using Gillespie [4]
Max Tyre Compression	Estimated from tyre geometry	Estimated from tyre geometry

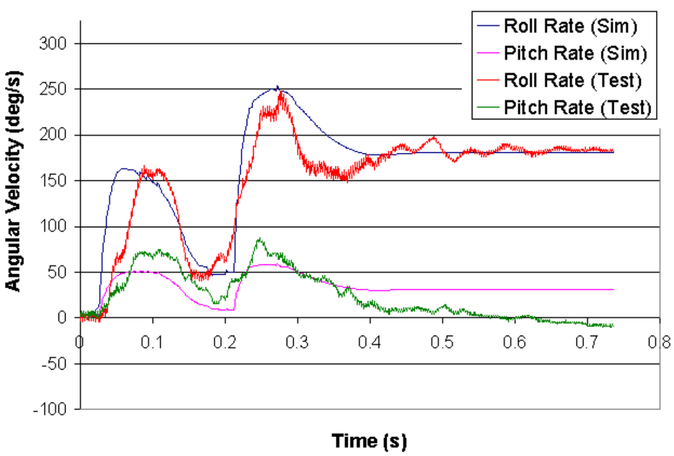

Figure 7. Spreadsheet and Test Results, Roll and Pitch Velocity – Convertible Vehicle

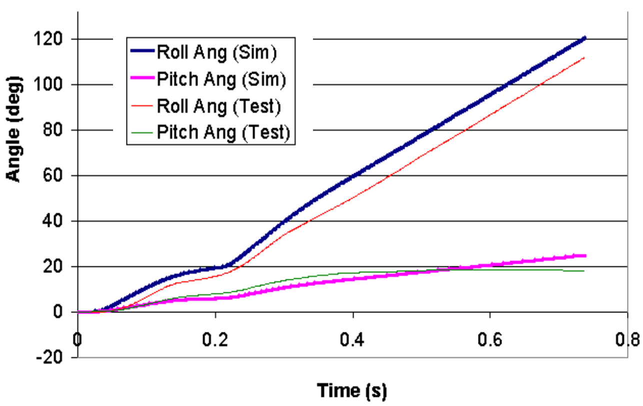

Figure 8. Spreadsheet and Test Results, Roll and Pitch Angles – Convertible Vehicle

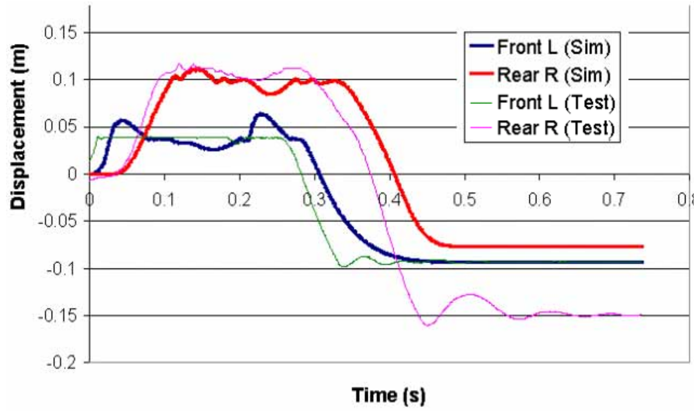

Figure 9. Spreadsheet and Test Results, Suspension Displacement – Convertible Vehicle

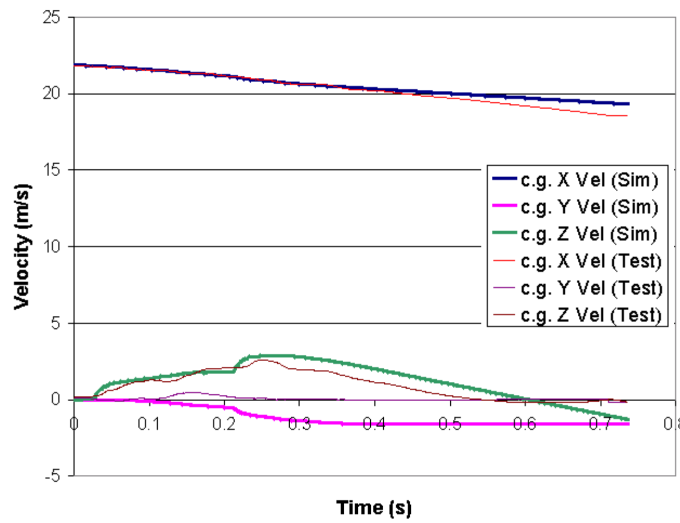

Figure 10. Spreadsheet and Test Results, Vehicle c.g. Velocity – Convertible Vehicle

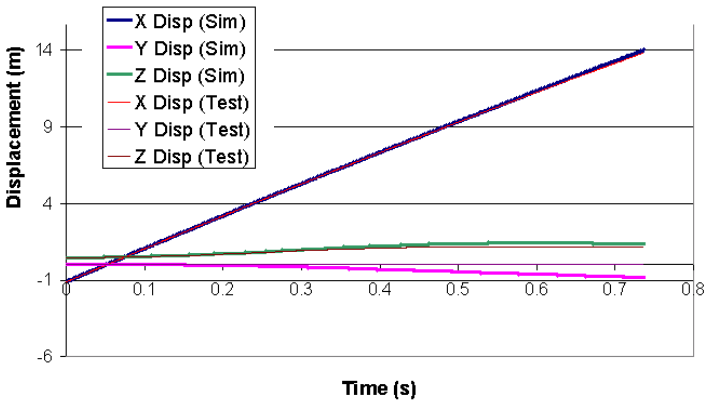

Figure 11. Spreadsheet and Test Results, Vehicle c.g. Displacement – Convertible Vehicle

The spreadsheet has a reasonable accuracy across the indices shown above. What is worthy of further comment is the front left suspension displacement. An already high force is being developed at the front left suspension mount. This is seen by the higher than test roll and pitch velocities, coinciding with the high front left suspension displacement. The vehicle c.g. vertical velocity also overshoots the test value when the front suspension displacement is high. If a greater force is applied at the front left suspension, by selecting appropriate suspension properties, this would lower the suspension displacement, but also mean greater overshoot in the roll, pitch & vertical velocities.

There is a fundamental issue here, the test vehicle has a lower front left suspension displacement than the spreadsheet, but also a lower front left suspension force. It is possible that linkages in the suspension are bending as the vehicle travels up the ramp. Thus the actual displacement would be higher that that recorded by the string pot, and lower forces generated due to the elastic/plastic deformation. What is worth mentioning here, and will be returned to later, is that the suspension arms directly contacted the ramp, creating a second load path into the vehicle.

<u>Determining Suspension Parameters – Sports Utility Vehicle</u> Full suspension data was supplied to the authors from the vehicle manufacturer. The stiffness data for both front and rear axles, was approximated by a tri-linear force deflection curve. A linear damper approximated the vehicle damper. A cautionary note should be sounded here. The manufacturer supplied damper information specified damping properties up to a wheel velocity of 1 m/s. This is very low for a corkscrew test, as the spreadsheet predicts wheel velocities of over 4 m/s. It is unlikely that the damper keeps its original linear characteristics at these wheel velocities.

SPREADSHEET PREDICTION OF VEHICLE BEHAVIOUR

The spreadsheet's predictive capacity can now be examined in two cases. What needs to be emphasised is that there is no tuning of spreadsheet parameters in the following discussion. For the two modelling cases now presented, the inertial, geometric and suspension are fully defined – there are no unknown parameters with which to "tune" the spreadsheet. Firstly, having determined the suspension parameters from the convertible test, they were used to predict the mid-size vehicle's behaviour. Secondly, for the sports utility vehicle, the NHTSA measurements of its inertial properties, and the author estimate of the moment of inertia in pitch, were used to predict its behaviour.

Spreadsheet Prediction – Mid-size Vehicle

The inertia and geometric parameters changed accordingly for the mid-size vehicle, along with the test speed. The convertible test speed was 78.9 km/h, while the mid-size vehicle hit the ramp at a lower requested speed of 77.1 km/h.

Figure 12 shows the roll and pitch velocities, and the pitch and roll angles are shown in Figure 13. Figure 14 shows suspension displacements, Figure 15 vehicle c.g. velocity, and the vehicle c.g. displacement is shown in Figure 16.

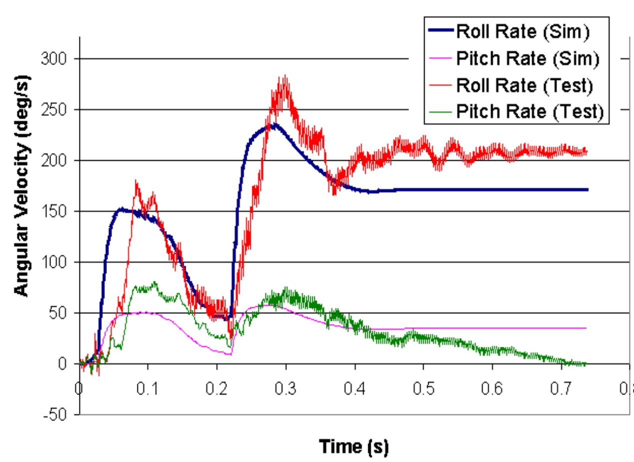

Figure 12. Spreadsheet and Test, Roll and Pitch Velocity – Mid-size Vehicle

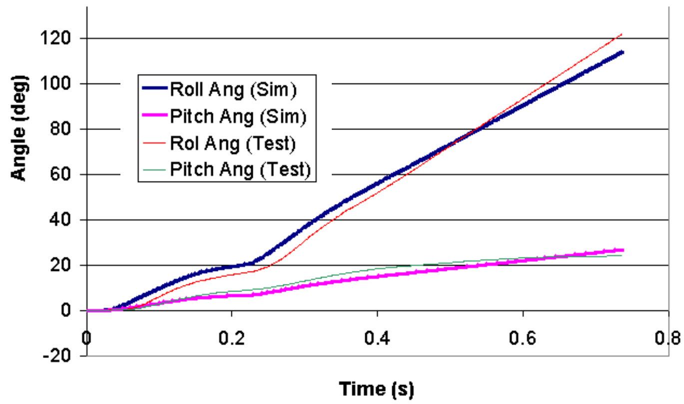

Figure 13. Spreadsheet and Test, Roll and Pitch Angles – Mid-size Vehicle

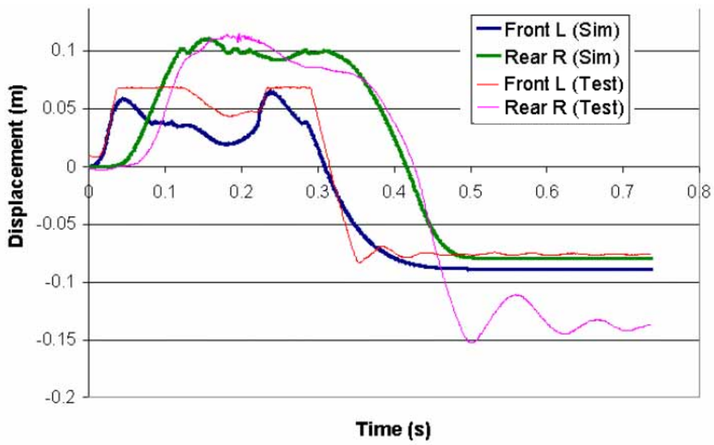

Figure 14. Spreadsheet and Test, Suspension Displacement – Mid-size Vehicle

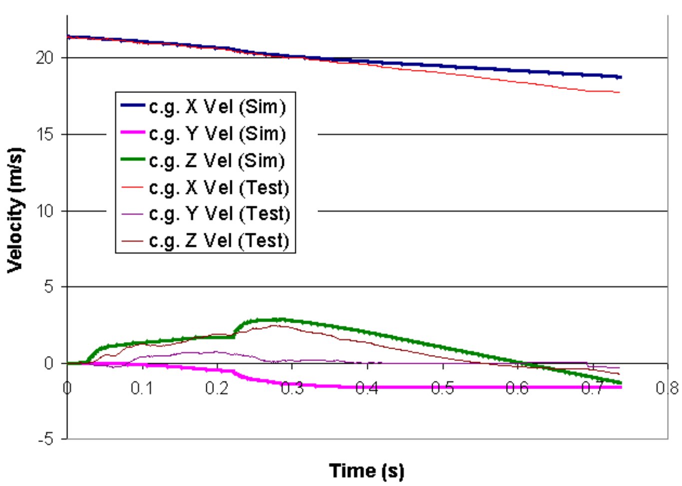

Figure 15. Spreadsheet and Test, Vehicle c.g. Velocity – Mid-size Vehicle

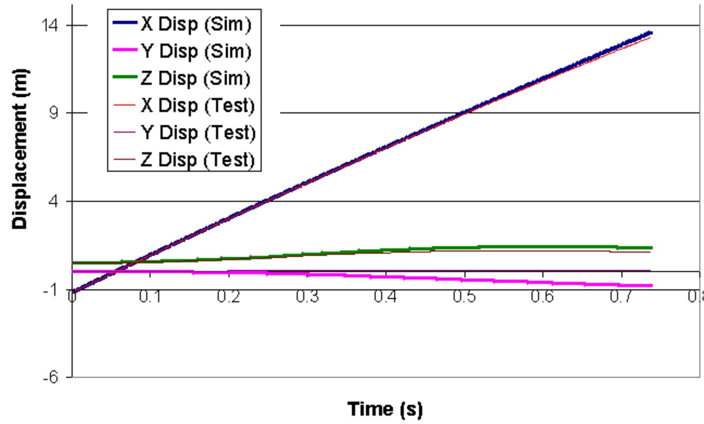

Figure 16. Spreadsheet and Test, Vehicle c.g. Displacement – Mid-size Vehicle

Spreadsheet Prediction – Sports Utility Vehicle

No tuning parameters have been used in these results. All the required suspension data was supplied to the author, while three inertial parameters were estimated. These were the moments of inertia in roll and pitch, and the c.g. height As mentioned earlier, the moment of inertia in roll and c.g. height were gained from from NHTSA's database [2], from results for a 5 door vehicle. The moment of inertia in pitch was determined from the author's experience of estimating this for other vehicles.

Figure 17 shows the roll and pitch velocities, while the pitch and roll angles are shown in Figure 18. Figure 19 shows the vehicle c.g. velocity, and the vehicle c.g. displacement is shown in Figure 20. The test speed in this case was 80 km/h.

Due to the uncertainty of estimate for the moment of inertia in pitch, its value was varied by plus and minus 15%. Increasing the pitch moment of inertia lowered the estimate of the roll angle, while decreasing the parameter increased the roll angle prediction. What was affected most was the roll velocity of the vehicle as it exited the ramp. The high and low roll angle predictions are shown in Figure 18. Figure 18 also reveals the value of the pitch moment of inertia used in the base line analysis agreed well with test data.

As the pitch moment of inertia increases, the vehicle takes longer to compress the rear right suspension, and once compressed, it stays compressed for longer. Thus when the front left wheel has left the ramp, the rear right suspension force acts for longer, decreasing the vehicle roll velocity as it leaves the ramp.

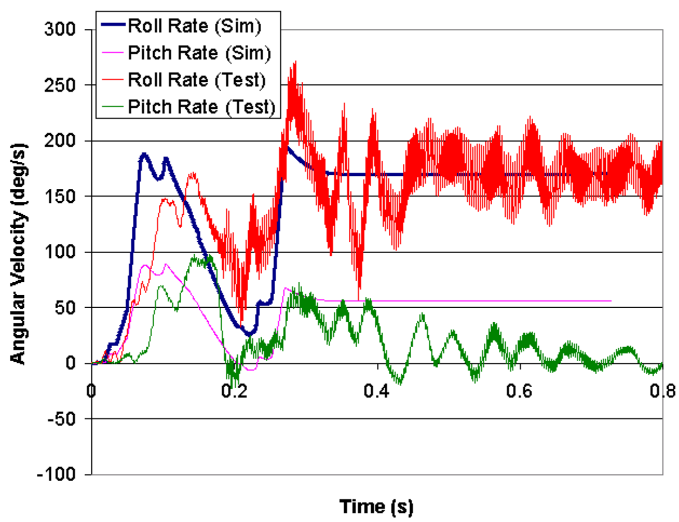

Figure 17. Spreadsheet and Test, Roll and Pitch Velocity - Sports Utility Vehicle

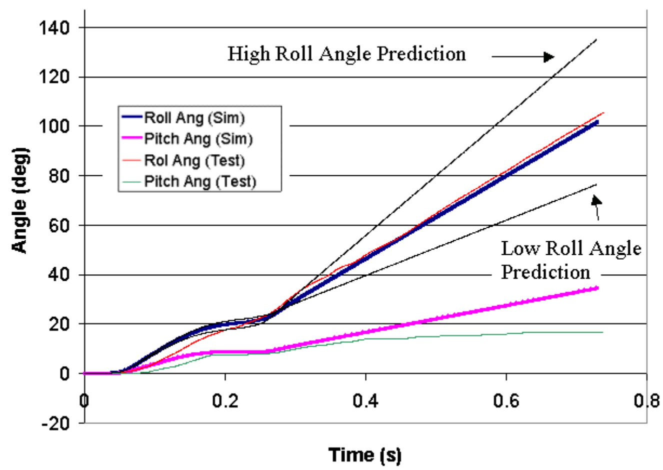

Figure 18. Spreadsheet and Test, Roll and Pitch Angles - Sports Utility Vehicle

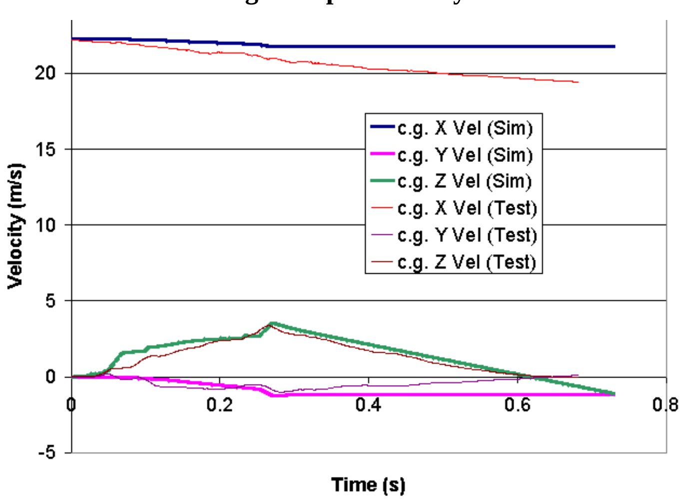

Figure 19. Spreadsheet and Test, Vehicle c.g. Velocity – Sports Utility Vehicle

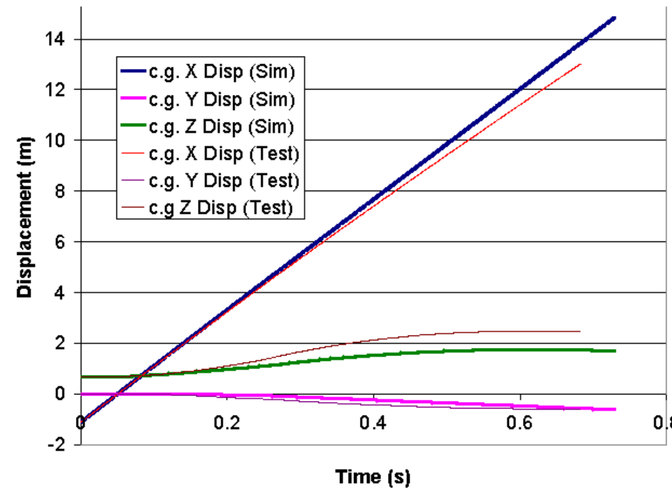

Figure 20. Spreadsheet and Test, Vehicle c.g. Displacement – Sports Utility Vehicle

DISCUSSION OF RESULTS

For a fixed corkscrew ramp, and fixed vehicle geometric and inertial properties, the vehicle behaviour is determined by suspension behaviour at its end limits of travel. Suspension parameters such as normal travel stiffness and damping determine the timing of the suspension bottoming out, and so the timing of the major forces on the vehicle.

Mid-size and Convertible Vehicles

The problem with the front left wheel displacement, and roll & pitch velocities being higher than test, has already been noted. Suspension contact with the ramp does occur. It's effect is clearly seen in Figure 21.

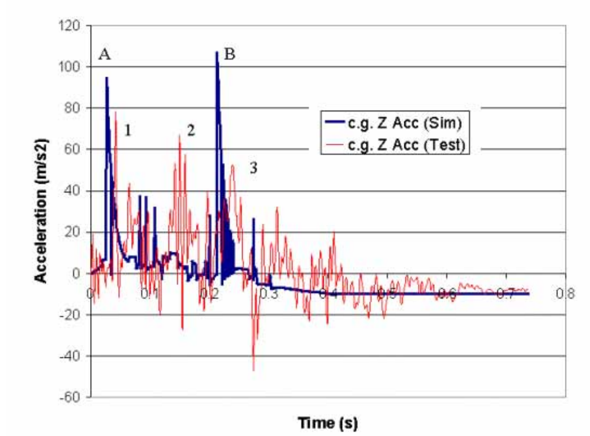

Figure 21. Spreadsheet and Test Results, Vehicle C.G. Z Acceleration

In Figure 21 the spreadsheet prediction has two very distinctive spikes, marked A & B. The test trace for the 2 door convertible has three smaller peaks, marked 1, 2 & 3. The simulation spikes correspond to the large forces developed when the front wheel bottoms out on its suspension, causing the initial rise in roll and pitch velocities. The first and third spikes from the test data are due to the front left wheel reaching the end of its stroke, but the second spike occurs from the direct suspension/ramp contact. This was confirmed by examining the time the second spike appears, and then calculating the distance along the ramp the vehicle had travelled at this point. Plywood had been placed along the side of the ramp, and showed scrape marks at the point the second spike occurred. Thus there was evidence of two load paths into the vehicle: firstly through the wheels compressing the suspension, and secondly the suspension contacting the ramp directly.

As the spreadsheet does not have the load input from the suspension/ramp contact, it has to generate a higher than test suspension force to generate the same vehicle behaviour. This can be done by either greater suspension displacement, or a stiffer suspension model. Conversely, should a suspension model be based on data supplied by a manufacturer, the spreadsheet could be expected to underestimate the vehicle roll behaviour. This is due to the second load path to the vehicle, suspension/ramp contact, not being represented.

Sports Utility Vehicle

The spreadsheet over-predicts both the longitudinal velocity on the ramp, and the pitch angle of the vehicle after its leaves the ramp. This is probably due to rolling friction and aerodynamic effects being excluded from the model.

Wind resistance could be a cause for the spreadsheet's over-prediction of the vehicle pitch angle. While the vehicle is on the ramp the spreadsheet follows the vehicle's pitch angle quite closely. The drop off in pitch velocity occurs while the vehicle is airborne. At 80 km/h (49.7 mph) it is expected that the aerodynamic effects on this vehicle would be significant. If an angled plate is put into a slipstream, the air resistance will try to rotate the plate to be parallel with the airflow. The same effect would work to reduce the upward pitching velocity of the vehicle while it is airborne. The mid-size and convertible vehicles also drop off in their pitch velocity after the ramp, but not as much as the sports utility vehicle. This is possibly due to these vehicles' more aerodynamic shape.

Both air resistance and rolling resistance could account for the spreadsheet over-predicting the sports utility vehicle's longitudinal velocity on the ramp. Neither of these effects were modelled in the spreadsheet. Again the convertible and mid-size vehicles showed less of a drop off in longitudinal velocity on the ramp, probably due to their more aerodynamic shape.

After the test the sports utility vehicle had witness marks where the lower front suspension arms had contacted the ramp during the test. Also noted were marks on the rear right wheel arch, indicating that the tyre had rubbed against it during the test. Hence evidence exists of load paths to the vehicle body, other than wheel movement on the suspension.

CONCLUSIONS

A spreadsheet has been produced which models with reasonable accuracy, vehicle behaviour during the corkscrew event up to the point of first ground contact. This allows the prediction of the vehicle speed to achieve a desired roll angle at ground contact. The current solution is cheap, it only requires a spreadsheet and fast, as there is only a momentary delay in calculating the results. When full inertial, geometric and inertial parameters are available the accuracy of the model increases. However in the absence of certain parameters, estimates can be made and parametric studies carried out to determine the effects of the unknown parameters. If greater accuracy is required for the model, areas of further work would be to include direct suspension/ramp contact, and the modelling of air resistance on the vehicle. This later point could be achieved by obtaining aerodynamic data such as drag coefficients and centre of pressure for the vehicle at various angles of pitch.

ACKNOWLEDGEMENTS

The authors would like to thank David Moseley of The Crash Safety Centre of TNO Automotive (UK), for his timely and fruitful advice.

REFERENCES

1. SAE 690804, Vehicle Rollover Dynamics Prediction by Mathematical Model, J. E. Ford and J. E. Thompson.
2. SAE 1999-01-1336, Measured Vehicle Inertial Parameters – NHTSA Data to November 1998.
3. Fundamentals of Vehicle Dynamics, Thomas D. Gillespie, SAE, 1992. Section on Vehicle Response Properties, p 146 - 148
4. Fundamentals of Vehicle Dynamics, Thomas D. Gillespie, SAE, 1992, p 163