Reconstruction and Analysis of Rollover Crashes of Light Vehicles
(Society of Automotive Engineers Course C1502)

Nathan A. Rose
Principal and Director
Kineticorp, LLC
nrose@kineticorp.com
Instagram – beautiful_evidence

Gray Beauchamp
Principal Engineer
Kineticorp, LLC
gbeauchamp@kineticorp.com

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# Table of Contents

Preface .......................................................................................................................................................... 5  

Chapter 1 – Overview of Rollover Reconstruction ................................................................................... 6  
  General Speed Analysis Approach ......................................................................................................... 8  
  Vehicle Geometry and Inertial Parameters ............................................................................................... 9  
  Overview of Rollover Test Procedures ................................................................................................... 10  
    Dolly Rollover Tests ............................................................................................................................. 10  
    Rollover Tests Using Modified Dolly Fixtures ...................................................................................... 14  
    Rollover Tests Using Automated Steering Control .............................................................................. 18  
    Controlled Rollover Impact System .................................................................................................... 21  
    Real World Rollovers ........................................................................................................................... 22  
References .............................................................................................................................................. 24  

Chapter 2 – Physical Evidence from Rollover Crashes .............................................................................. 27  
  Scene Evidence ...................................................................................................................................... 27  
  Vehicle Evidence .................................................................................................................................... 41  
  Scene and Vehicle Documentation Checklists ....................................................................................... 47  
    Site Inspection Checklist ...................................................................................................................... 47  
    Vehicle Inspection Checklist ................................................................................................................ 47  
  Photogrammetry ..................................................................................................................................... 48  
References .............................................................................................................................................. 54  

Chapter 3 – Analysis Methods – Rollover Phase ........................................................................................ 57  
  Average Deceleration Rates ................................................................................................................... 57  
  Determining Likely Number of Rolls from Roll Distance ......................................................................... 59  
  Typical Rollover Characteristics ............................................................................................................ 61  
  Mathematical Analysis of the Orientation of Scratches on the Vehicle .................................................. 64  
  Reconstruction Methodology – Roll Phase (Constant Deceleration) ..................................................... 64  
  Case Study – Asay’s Isuzu Rodeo Rollover Test (Test #2, 2010-01-0521) ........................................... 66  
References .............................................................................................................................................. 69  
Appendix ................................................................................................................................................ 71  
Exercises ................................................................................................................................................ 72  

Chapter 4 – Advanced Analysis Methods – Rollover Phase ...................................................................... 73  
  Vehicle-to-Ground Impact Model ............................................................................................................ 73  
    Impact Model Equations ....................................................................................................................... 73  
    Impulse Ratio ($\mu$) ............................................................................................................................... 76  
    Impact Angle ($\phi$) ............................................................................................................................... 79  
    Non-Constant Deceleration Models ...................................................................................................... 80  
    Validation of a Non-Constant Deceleration Model ............................................................................ 83  
    Reconstruction Methodology – Roll Phase (Linearly Decreasing Deceleration) ............................. 87
The Motion of Ejected Occupants......................................................................................................... 173
Applying the Occupant Ejection Model .............................................................................................. 175
Validation of the Occupant Ejection Model.......................................................................................... 179
References............................................................................................................................................ 179
Chapter 5 – Analysis Methods – Trip Phase

According to Martinez [1996], “the trip phase of [a rollover] accident is the portion of the event in which the lateral forces overwhelm the vehicle’s ability to maintain four wheeled contact on the ground.” During the trip phase, lateral forces applied to the tires or wheels on the leading side of the vehicle cause the trailing side tires to lose contact with the ground and the vehicle to begin tipping up. When the center of mass (CoM) of the vehicle rolls past the point of contact between the leading side tires and the ground, the trip phase ends and the rollover phase begins [Orlowski, 1989]. During the trip phase, the vehicle acquires an upward velocity and a roll velocity and loses a portion of its ground plane velocity. At the end of the trip phase, the vehicle becomes airborne. Once this occurs, the roll velocity that has been developing during the trip phase will level off and become constant until the next contact with the ground occurs [Asay, 2010].

Following are several examples of mechanisms that can supply the tripping force necessary to initiate a rollover: (1) a combination of tire forces and suspension effects from severe steering inputs [Larson, 2000; Garrott, 2001; Wilson, 2007a and 2007b; Stevens, 2011]; (2) interaction between a tire or wheel rim and pavement [Marine, 1999]; (3) wheels furrowing into soil or sod [Cooperrider, 1990 and 1998; Asay, 2009 and 2010]; and (4) wheels impacting a curb [Jones, 1975; Cooperrider, 1990; Hughes, 2002]. For each of these mechanisms, the magnitude and duration of the tripping force and the manner in which that force is applied through time will vary. The duration of the trip phase and the speed changes that occur during this phase depend on the characteristics of the trip force [Orlowski, 1989]. To the extent that the characteristics of the tripping force affect the characteristics of the vehicle motion during the trip phase, each tripping mechanism will produce trip phase vehicle motion characteristic of that mechanism. The features of the vehicle motion during the trip phase also depend on the vehicle characteristics, such as the weight, moments of inertia, center of mass height, track width, and suspension characteristics. This chapter will develop methods for calculating the tripping forces for a particular rollover and for determining the changes in over-the-ground, vertical, and roll speeds that occur during the trip phase.

Trip Modeling and Data

Cooperrider examined the trip phase both experimentally and analytically [1990]. He conducted a series of rollover tests that included both curb tripped and soil tripped rollovers. Two out of five curb trip tests resulted in rollover. Both soil trip tests resulted in rollover. The curb trip tests utilized a six inch (15 cm) square section of steel box tubing that was rigidly attached to the ground perpendicular to the vehicle’s travel direction. The test vehicles traveled laterally into the curb with their right sides leading at speeds around 30 mph (48 kph). In the two tests where rollover occurred as a result of the curb impact, the vehicles experienced average deceleration rates during the trip of 12.4g and 13.2g. These decelerations occurred over time periods of 60 milliseconds and 40 milliseconds. The soil tripped tests utilized a 40 foot by 20 foot (12.2 by 6.1 meter) area of loosely packed dirt and finely crushed granite. The test vehicles traveled laterally into this soil with their right sides leading. The initial speeds of the vehicles in these two tests were 33.7 and 27.0 mph (54.2 and 43.5 kph). The vehicle tires furrowed into the soil, causing the vehicles to rollover. In these two tests, the vehicle’s experienced average deceleration rates during the trips of 1.62g and 1.71g. These decelerations occurred over time periods of 513 milliseconds and 460 milliseconds.

Cooperrider also derived an expression for the lateral force necessary to initiate and sustain the trip phase. This expression was derived by first assuming that the tripping force is constant and then writing an approximate set of equations of motion for a model similar to that depicted in Figure 5-1. The vehicle depicted in this figure is in the midst of the trip phase, with the trailing wheels having lifted off the ground and with the lateral tripping force applied to the leading side tires. This model assumes a rigid vehicle, and so, suspension and tire compliances are ignored. The model depicted in Figure 5-1 is only different from the one Cooperrider used in the orientation of the coordinate system. Cooperrider had the y direction oriented to the right and the z direction oriented down. The model below reverses both of these, using a right-handed coordinate system with the y-axis pointed to the left and the z-axis pointed up. The angular position of the body is measured with positive rotation in the clockwise direction. With this coordinate system, the vehicle depicted in Figure 5-1 would be facing into the page and rolling with its passenger’s side leading.
The equations of motion that Cooperrider wrote for this model are as follows (using the coordinate system depicted in Figure 5-1):

\[ m\ddot{y} = F_T \]  
\[ m\ddot{z} = N - mg \]  
\[ I_{xx} \ddot{\theta} = F_T \cdot h - N \cdot \frac{T}{2} \]

In these equations, \( y \) is the lateral position of the vehicle’s center of mass, \( z \) is the vertical position, \( \theta \) is the vehicle’s roll angle, \( F_T \) is the tripping force, \( N \) is the normal force, \( h \) is the height of the vehicle center of mass, \( T \) is the vehicle track width, and \( I_{xx} \) is the roll moment of inertia. Equation (5.3) is approximate because the moment arms employed are only accurate when the roll angle is zero. This assumption is only true at the instant the trailing side wheels lift off. These moment arms become increasingly inaccurate as the roll angle increases. Thus, Cooperrider’s model is setup to predict the lateral force necessary to initiate trailing wheel lift off.

Cooperrider next wrote the following geometric constraint equation which imposes the condition that the leading side tires remain in contact with the ground during the trip phase:

\[ z = \frac{T}{2} \cdot \sin \theta \]
Cooperrider then employed a small angle assumption for Equation (5.4) reducing it to the following equation. This equation also only holds true at the instant the trailing side tires lift off and, from that point forward, it becomes increasingly inaccurate.

\[ z = \frac{T}{2} \cdot \theta \] (5.5)

Cooperrider used these equations in conjunction with the idea that in order to initiate a rollover, “the kinetic energy in the roll and vertical motions must equal the remaining change in potential energy needed to raise the [center of mass] to the neutral stability point…” Cooperrider reported the following expression that relates the trip force and the trip duration (\(\Delta t\)):

\[ \left( \frac{F_T}{mg} - \frac{T}{2h} \right)^2 \cdot \Delta t^2 + \frac{T}{2h} \cdot \left( \frac{F_T}{mg} - \frac{T}{2h} \right) \cdot \Delta t^2 = \frac{2I_{xx}}{mgh} \cdot \left( \sqrt{\left( \frac{T}{2h} \right)^2 + 1} - 1 \right) \] (5.6)

In this equation:

\[ I_{xx} = I_{xx} + \frac{mT^4}{4} \] (5.7)

Since a reconstructionist will typically be interested in the magnitude of the tripping force, which is directly related to the vehicle’s deceleration rate, this expression is not in an easily useable form. Nonetheless, Cooperrider reported that “the correlation of the four [curb and soil trip] data points provided by [the experimental data] with the constant force model is good, although additional comparisons are needed to conclusively evaluate the model.”

Brach examined options for modeling the trip phase in his accident reconstruction text [2005]. As a part of his discussion, he presented a reworked form of Cooperrider’s equation writing it in a more useable form that explicitly yields the tripping force in gravitational units. He also utilized the roll radius of gyration for the vehicle, noting that \(I_{xx} = mk_{xx}^2\). Brach’s equation is as follows:

\[ F_T = \frac{1}{2\Delta t} \left[ \Delta t \cdot \frac{T}{2h} + \sqrt{\Delta t^2 \cdot \left( \frac{T}{2h} \right)^2 + \frac{8h}{g} \left( \frac{k_{xx}^2}{h^2} + \left( \frac{T}{2h} \right)^2 \right) \left( \sqrt{\left( \frac{T}{2h} \right)^2 + 1} - 1 \right)} \right] \] (5.8)

Equation (5.8) was used to calculate lateral tripping forces for the two vehicles used in Cooperrider’s 1990 curb and soil tripped tests. In the two soil tripped tests, Cooperrider reported that the vehicle’s experienced average deceleration rates during the trips of 1.62g and 1.71g. These decelerations occurred over time periods of 513 milliseconds and 460 milliseconds. Equation (5.8) yielded trip forces of 1.84g and 1.82g. These calculated values would yield speed changes of 20.7 mph and 18.3 mph, whereas the actual reported decelerations would yield speed changes of 18.2 mph and 17.3 mph. In the two curb tripped tests where rollover occurred, with the same vehicle models as in the soil tripped tests, the vehicles experienced average deceleration rates during the trip of 12.4g and 13.2g. These decelerations occurred over time periods of 60 milliseconds and 40 milliseconds. For these durations, Equation (5.8) yielded trip forces or deceleration rates of 8.89g and 11.93g, lower than actual. These calculated values would yield speed changes of 11.7 mph and 10.5 mph, whereas the actual reported decelerations would yield speed changes of 16.3 mph and 11.6 mph.

In 1998, Cooperrider reported six additional soil trip tests that each utilized a 1984 Oldsmobile Cutlass Ciera. The first test was run with an initial speed of 13.5 mph (21.7 kph) and, as Cooperrider stated, “the subsequent test speeds were established based on the previous test results in an effort to closely bracket
the minimum trip speed.” Ultimately, the initial speeds for the vehicles in this test series ranged from 13.5 to 42.9 mph (21.7 to 69.0 kph). In the first four tests, the vehicle did not roll over. The initial speed of the test vehicle in the fourth test was 21.2 mph (34.1 kph). In the fifth and sixth tests, which had speeds of 23.0 mph and 42.9 mph (37.0 and 69.0 kph), the vehicle rolled over — ½ roll in the fifth test and 1 roll in the sixth test. When the trip phase was defined as the time over which the vehicle reached a roll velocity of 100 degrees per second, the average deceleration rate of the vehicles during the trip phase of both tests where rollover occurred was 1.23g (durations of 624 and 664 milliseconds). When the trip phase was defined as the time over which the vehicle reached a roll angle of 52 degrees, the average deceleration rates from the trip phase of the two tests were 1.05g and 1.39g (durations of 784 and 792 milliseconds). Neither of these definitions of the trip phase could be employed in practice.

Equation (5.8) was used to calculate lateral tripping forces for the two tests that resulted in rollover in Cooperrider’s 1998 article. When the trip phase was defined as the time over which the vehicle reached a roll angle of 52 degrees, the trip phase durations were 784 and 792 milliseconds. For these trip durations, Equation (5.8) yielded trip deceleration rates of 1.57g and 1.56g, higher than the values average deceleration rates Cooperrider reported. These calculated deceleration rates resulted in calculated speed changes of 26.9 mph and 27.1 mph, whereas the reported speed changes were 18.1 mph and 24.2 mph. Thus, for the four soil tripped tests reported by Cooperrider, his model overestimated the deceleration rate and for the two curb trip tests, it underestimated the deceleration rate.

Larson [2000] reported two soil tripped rollover tests – one with a 1991 Jeep Wrangler and another with a 1986 Chevrolet S-10 Blazer. Each vehicle was released without an initial roll angle and a slip angle of approximately 52 degrees across a dirt surface with a 15.5 degree downslope. The vehicles deposited tire and rim furrows in the dirt before beginning to roll over. Larson reported that the Jeep Wrangler was traveling 40.7 mph (65.5 kph) when it was released from the dolly, that it traveled 70.5 feet (21.5 meters) from release to rest, that it furrowed for approximately 20.7 feet (6.3 meters) before entering the roll phase, and that it completed one roll before coming to rest. The roll distance was approximately 49.8 feet (15.2 meters). Larson did not report a speed for the vehicle at the end of the furrowing. The average deceleration rate for the trip and roll phases combined was 0.78g. If we assume that the deceleration rate of this vehicle during the roll phase was around 0.45g, then the deceleration rate during the trip phase was around 1.59g.

Larson reported that the Chevrolet S-10 Blazer was traveling 41.1 mph (66.1 kph) when it was released from the dolly, that it traveled 74 feet (22.6 meters) from release to rest, that it furrowed for approximately 18.1 feet (5.5 meters) before entering the roll phase, and that it completed two rolls before coming to rest. The roll distance was approximately 55.9 feet (17.1 meters). Again, Larson did not report a speed for the vehicle at the end of furrowing. The average deceleration rate for the trip and roll phases combined was 0.76g. If we assume that the deceleration rate of this vehicle during the roll phase was around 0.45g, then the deceleration rate during the trip phase was around 1.73g.

Equation (5.8) was used to analyze Larson’s soil trip test of a Chevrolet S-10 Blazer based on the reported furrow distance and initial speed. This equation yielded a lateral tripping force of 1.64g. Here, though, the vehicle had a slip angle of approximately 52 degrees during the trip phase. Thus, only a portion of the lateral tripping force would have been opposing the velocity and acting to decelerate the vehicle. Equation (5.9) can be used to take this into account.

\[ f_{\text{trip}} = f_{\text{cooperrider}} \cdot \sin \alpha + f_{\text{long}} \cdot \cos \alpha \]  

(5.9)

In this equation, \( f_{\text{trip}} \) is the resultant deceleration during the trip phase, \( f_{\text{cooperrider}} \) is the normalized lateral tripping force calculated with Equation (5.8), \( \alpha \) is the average vehicle slip angle during the trip phase, and \( f_{\text{long}} \) is any longitudinal force (normalized by the vehicle weight) that was applied to the vehicle during the trip phase from braking or furrowing effects. Using this equation for Larson’s test with the assumption that there was no longitudinal force results in a deceleration rate of 1.29g, significantly below the actual value of 1.73g.
In 2002, Hughes reported a series of 8 curb tripped rollover tests conducted with 2001 Nissan Pathfinders. These tests used a modified dolly fixture that did not introduce an initial roll angle to the test vehicles. Each test vehicle was launched laterally from the dolly with a speed of around 31 mph (50 kph) and then the leading side tires impacted the curb, which was 15 centimeters high. Rollover occurred in all 8 tests. The roll distances ranged from 54 to 72 feet (16 to 22 meters) and the number of rolls varied between 1 and 2½ times. Peak accelerations during the curb impacts ranged from 14.5 to 20.5 g. Hughes did not report impact durations.

Asay reported some minimal analysis of the trip phase for each of the tests he published in 2010. He observed that, for all of the tests where rollover ultimately occurred, “the tripping phase was well defined by distinct furrows in the soil that terminated as the vehicle overturned.” He reported that the peak lateral acceleration experienced by the vehicles during the trip phase in these tests ranged from 2.76 to 5.70 g. The peak lateral accelerations while furrowing through the dirt for the tests that did not ultimately result in rollover were 1.76 and 1.71 g.

**Trip Modeling without Small Angle Assumptions**

Rose and Beauchamp reported modeling of the trip phase using the model depicted in Figure 5-1. Their modeling did not employ small angle assumptions, and so, it can be used to determine the effects of Cooperrider’s use of small angle assumptions. The external forces applied to the vehicle in this model again include the weight, the lateral tripping force and the leading side normal force. The vehicle weight is applied at the center of mass and the lateral tripping force and the normal force are applied at the leading side wheels. This has the following equations of motion:

\[ m a_y = F_T \]  \hspace{1cm} (5.10)  
\[ m a_x = N - m g \]  \hspace{1cm} (5.11)  
\[ I_{xx} a_r = F_T \cdot \delta \cdot c(\lambda - \theta_r) - N \cdot \delta \cdot s(\lambda - \theta_r) \]  \hspace{1cm} (5.12)

In Equations (5.10) through (5.12), \( a_y \) is the center of mass ground plane acceleration, \( a_x \) is the center of mass vertical acceleration, \( a_r \) is the roll axis angular acceleration of the vehicle, \( m \) is the vehicle mass, \( I_{xx} \) is the vehicle roll moment of inertia, \( F_T \) is the time dependent tripping force, \( N \) is the normal load at the leading wheels, \( \theta_r \) is the vehicle roll angle and \( \delta \) is a geometric parameter defined as follows:

\[ \delta = \sqrt{h^2 + \left(\frac{T}{2}\right)^2} \]  \hspace{1cm} (5.13)

Thus, \( \delta \) is the distance from the vehicle center of mass to the point of the application of the trip force. The \( s \) and \( c \) in Equations (5.10) through (5.12) refer to the sine and cosine. The angle \( \lambda \) in Equation (5.12) is defined as follows:

\[ \lambda = \tan^{-1}\left(\frac{T}{2h}\right) \]  \hspace{1cm} (5.14)

During the trip phase, the leading side wheels remain in contact with the ground, and thus, Equations (5.10) through (5.14) are supplemented with the following geometric constraint equation:

\[ z = \delta \cdot c(\lambda - \theta_r) \]  \hspace{1cm} (5.15)
Differentiation of Equation (5.15) twice with respect to time yields Equations (5.16) and (5.17), which give the vertical velocity and vertical acceleration of the center of mass, respectively. The symbol \( \omega_r \) represents the roll velocity.

\[
v_z = \delta \cdot \omega_r \cdot s(\lambda - \theta_r) \tag{5.16}
\]

\[
a_z = \delta \cdot a_r \cdot s(\lambda - \theta_r) - \delta \cdot \omega_r^2 \cdot c(\lambda - \theta_r) \tag{5.17}
\]

Substitution of Equation (5.17) into Equation (5.11) yields Equation (5.18), which describes the leading side normal load as a function of the vehicle weight, roll angle, roll velocity, and roll acceleration.

\[
N = mg + m\delta a_r \cdot s(\lambda - \theta_r) - m\delta \omega_r^2 \cdot c(\lambda - \theta_r) \tag{5.18}
\]

Substitution of Equation (5.18) into Equation (5.12) leads to Equation (5.19), a nonlinear, second-order ordinary differential equation describing the angular motion of the rigid vehicle.

\[
\alpha_r = \frac{F_r \delta c(\lambda - \theta_r) - mg \delta s(\lambda - \theta_r) + m\delta^2 \omega_r^2 s(\lambda - \theta_r)c(\lambda - \theta_r)}{I_{xx} + m\delta^2 s^2(\lambda - \theta_r)} \tag{5.19}
\]

Numerical integration of Equation (5.19) will yield a solution for any particular case. In this process, the trip duration is prescribed and the magnitude of the tripping force is iteratively changed until the normal load on the leading tires goes to zero at the prescribed trip duration. This analysis would usually begin with the initial conditions of a zero roll angle and zero roll velocity.

Figure 5-2 and Figure 5-3 show analysis conducted with both the Cooperrider model and the model just described to evaluate the effect of the small angle assumptions employed by Cooperrider. This analysis assumed a constant trip force throughout the trip phase. Figure 5-2 compares the normalized trip force obtained with the two models for parameters typical of a passenger vehicle. Figure 5-3 contains a similar comparison for parameters typical of an SUV. These results were developed using the following vehicle parameter sets:

**SUV Parameter Set**
- Vehicle Weight = 5000 lb (2268 kg)
- Roll Moment of Inertia = 750 lb-ft-sec\(^2\) (104 kg-m-sec\(^2\))
- Radius of Gyration = 2.2 ft (.67 meters)
- Track Width = 61.0 inches (155 cm)
- CoM Height = 28.0 inches (71.1 cm)

**Passenger Car Parameter Set**
- Vehicle Weight = 3600 lb (1633 kg)
- Roll Moment of Inertia = 500 lb-ft-sec\(^2\) (69.1 kg-m-sec\(^2\))
- Radius of Gyration = 2.1 ft (.64 meters)
- Track Width = 60.0 inches (152 cm)
- CoM Height = 22.0 inches (55.9 cm)
Figure 5-2 – Comparison between Trip Models With and Without Small Angle Assumptions (Passenger Car Parameters)

Figure 5-3 – Comparison between Trip Models With and Without Small Angle Assumptions (SUV Parameters)
In these figures, the trip duration was varied between 0.5 and 1.5 seconds, in ¼ second increments. The values depicted in these graphs show that the small angle assumptions employed by Cooperrider result in higher calculated trip forces than are obtained with a model that does not employ small angle assumptions. That said, the differences are small. The Cooperrider model – in the reworked form presented by Brach – is easier to implement than a model that does not employ small angle assumptions, so given the small differences between the models, it makes sense for accident reconstructionists to use the Cooperrider model for trip phase speed calculations, assuming that the rigid vehicle assumptions do not make it inaccurate. This possibility will be explored later.

Rose and Beauchamp also conducted analysis with both a linearly increasing trip force and with a sinusoidal shaped trip force. Figure 5-4 contains roll angle curves, Figure 5-5 contains roll velocity curves, and Figure 5-6 contains vertical velocity curves, obtained with both a tripping force that increases linearly with time, as described by Equation (5.20), and with a sinusoidal tripping force described by Equation (5.21). A version of Figure 5-6 containing SI units is included in the appendix to this chapter.

\[
F_T(t) = F_0 + (F_{max} - F_0) \cdot \frac{t}{\Delta t_{trip}}
\]  
(5.20)

\[
F_T(t) = F_{max} \cdot \sin \left( \frac{\pi \cdot t}{\Delta t_{trip}} \right)
\]  
(5.21)

In these equations, \(F_0\) is the initial tripping force magnitude, \(F_{max}\) is the maximum tripping force magnitude, \(\Delta t_{trip}\) is the trip duration and \(t\) is time. A linearly increasing trip force characteristic could potentially correspond to that which would be produced by a vehicle furrowing into soil, with the force increasing as the depth of the furrowing increases [Orlowski, 1989; Gopal, 2004]. A sinusoidal trip force shape could potentially represent an impact-type tripping mechanism where the tripping force starts out at zero, builds up to a peak value, and then returns to zero. For each curve in Figure 5-4, Figure 5-5, and Figure 5-6, the initial tripping force magnitude was set at the value of the static stability factor and the maximum tripping force magnitude was changed until leading wheel liftoff occurred at the prescribed trip duration.

\[\text{Figure 5-4 – Roll Angle v. Time, Linearly Increasing Trip Force (Left) and Sinusoidal Trip Force (Right)}\]
For the linearly increasing trip force, the trip duration was varied between 0.5 and 1.5 seconds, in ¼ second increments. For the sinusoidal tripping force, the trip duration was varied between 0.05 and 0.75 seconds. This range starts and ends at lower trip durations than those used for the linearly increasing trip force. The motivation for the low-end of this range is the fact that the sinusoidal-type tripping force would likely be associated with impact-type tripping mechanisms, such as curbs, and would generally occur with short duration applications [Jones, 1975; Cooperrider, 1990; Shi, 2006]. The high-end of this range was set so that there would be some overlap between the durations used for the linearly increasing and sinusoidal-shaped trip force shapes.

Figure 5-4 through Figure 5-6 lead to the following observations:

- An increase in trip duration results in an increase in roll angle and a decrease in the roll and vertical velocities at the end of trip.

- The linearly increasing trip force resulted in higher roll rates for a given trip duration than the sinusoidal trip force. This could mean that that soil tripped rollovers would generally begin with higher roll rates than soil tripped rollovers. That said, soil tripped and curb tripped rollovers are associated with very different trip durations, so the differences in roll rate for the two mechanisms may not be easily discernable in the real world. Ultimately, the physical evidence and the characteristics of a particular vehicle and trip mechanism should drive the reconstructionists determination of the roll rate with which the vehicle begins the roll phase.
The sinusoidal trip force resulted in higher vertical velocities than the linearly increasing trip force. This likely means that curb tripped rollovers would begin with greater vertical velocity than soil tripped rollovers. The vertical velocity curves peaked prior to the end of the trip phase. The vertical velocity then dropped off before leading wheel liftoff occurred.

For a sinusoidal trip force with a short duration trip phase, leading wheel liftoff occurs at low roll angles relative to what occurred for the longer linearly increasing trip forces. This likely means that curb tripped rollovers would begin with a lower initial roll angle than soil tripped rollovers. This finding is consistent with the roll angles observed at the end of the trip in the curb tripped rollovers reported by Hughes [2002]. In the eight tests that Hughes ran, these roll angles ranged from 5 to 12 degrees.

These results demonstrate that the vehicle orientation and velocity conditions that exist when a vehicle enters the roll phase depend on both the vehicle characteristics and the trip force characteristics. It stands to reason that the vehicle motion that occurs during the roll phase will depend on the orientation and velocity conditions with which the vehicle enters that phase. Therefore, the trip force characteristics will be influential in determining the vehicle motion that will occur during the roll phase. This is not to say that the analyst will be able to discern the relationship between the initial conditions for the roll phase and the roll motion that results.

Case Study – Asay’s Isuzu Rodeo Rollover Test (Test #2, 2010-01-0521)

This section analyzes the trip phase of the Isuzu Rodeo rollover test that Asay reported in 2010. The roll phase was analyzed in Chapters 3 and 4. In this test, the Isuzu was initially traveling a speed of 73.5 mph (118.3 kph). The vehicle was steered sharply to the left, causing the vehicle to travel across the roadway to the left and to yaw counterclockwise. The vehicle developed a slip angle of sufficient magnitude that tire marks were deposited on the road surface. Approximately one second later, the first steering input was followed by a significant steering input back to the right. The vehicle reversed its yaw direction, but continued off the left side of the road into the dirt, developing a significant slip angle as it yawed in a clockwise manner. The vehicle deposited furrows in the dirt and then began rolling over. Figure 5-7 contains frames from the video of this test that depict the final yaw motion of the vehicle that preceded the roll. The trip phase encompasses all or part of this final yaw.

Based on Asay’s survey data, the tire marks and furrows deposited by the Isuzu during the loss of control and trip phases were diagrammed and the vehicle positions depicted in Figure 5-8 were determined from that evidence. The trip phase occurs during the final yaw of the vehicle leading into the roll phase. This final yaw motion is depicted with 5 numbered vehicle positions in Figure 5-8. To determine the actual speed at each of these positions, the VBOX data was aligned to these positions in accordance with the alignment data reported by Asay. Arndt’s corrected speed of 44.4 mph was used for the fifth position. However, for the first four positions of the final yaw, the vehicle had not developed significant roll velocity, and so, correction of the speeds for these positions was not necessary. The process of aligning the VBOX data resulted in the following speeds for the five positions identified in Figure 5-8 – 68.9 mph at the first position, 67.6 mph at the second position, 64.3 mph at the third position, and 61.1 mph at the fourth position.

One issue that arises in modeling the trip phase is deciding what portion of the final yaw constitutes the trip phase. One approach would be to look at the tire mark evidence to determine when both trailing side tires have lifted off. In this case, that occurred at Position 4. This approach has the advantages of being tied to the physical evidence and of using the theoretical definition of the trip phase (trailing wheel lift-off to leading wheel lift-off). That said, it is not clear that this is the best approach. First of all, in the development of this rollover – and of most real-world rollovers with a yaw that precedes the roll – the two trailing side tires lift-off at different points in time. In addition, the modeling presented in Figure 5-4 and Figure 5-5 demonstrates that the development of roll angle and roll velocity during the trip phase can start off slow and then accelerate quickly at the end. Because the modeling of Figure 5-4 and Figure 5-5 was carried out with rigid vehicle assumptions, trailing wheel lift-off still occurred at the beginning of the trip phase. However, with real vehicles that have suspensions, the tripping forces could be at work developing roll of the vehicle body while the trailing side tires remain in contact with the ground due to relaxing of the suspension. Thus,
choosing the position within the evidence where the second trailing side tire lifts off may result in the analyst missing a portion of the trip phase.

Figure 5-7 – Frames from the Video of the Trip Phase

Another option would be to use the position at which the first trailing side tire lifts off as the beginning of the trip phase. In this case, this occurs at the passenger side rear wheel around Position 3 in Figure 5-8. This option suffers from the same potential problems as the first, though they would be less significant. Another option would be to treat the entire final yaw from Positions 1 through 5 as the trip phase. The calculations
reported in this section considered each of these possible definitions of the trip phase. The segment distance and slip angle measurements used in these calculations are depicted in Figure 5-9 and the first set of calculations are summarized in Table 5-1. These calculations used Equations (5.8) and (5.9), assumed a deceleration rate during the roll phase of 0.363 to match the speed at the beginning of the roll phase, and used the following values for the Isuzu Rodeo.

Weight = 4,266 pounds
Roll Moment of Inertia = 518 lb-ft-sec$^2$
Radius of Gyration = 1.98 feet
Track Width = 57.5 inches
Center of Mass Height = 25.9 inches

The calculations summarized in Table 5-1 were initially run assuming there was no longitudinal tripping force present during the trip phase. Then, the longitudinal tripping force that would be necessary to match the actual deceleration rate during the trip phase was calculated.

Several observations can be made based on the results reported in Table 5-1. First, with the assumption that there was no longitudinal force present during the trip phase, Equation (5.8) and (5.9) underestimated the actual deceleration rate regardless of how the trip phase was defined. There was no braking applied to the vehicle during this test, so whatever longitudinal forces were present must have been generated due to effects of furrowing and steering. Including these forces would improve the accuracy of the speed analysis.

Figure 5-10 shows the actual longitudinal and lateral (ground plane) accelerations measured during the initial counterclockwise yaw of the test vehicle and the subsequent clockwise yaw that includes the trip phase. This graph begins at a time of 0.5 seconds and continues until a time of 3.25 seconds. According to Asay’s article, time zero corresponds to release of the vehicle. The ground plane lateral accelerations shown in this graph were calculated from the measured vehicle frame lateral and vertical accelerations and
the roll angle. The extents of the final yaw in Figure 5-10 can be identified based on the roll velocity data from the test, the location of tire marks and furrows from the final yaw, and review of the video.

Figure 5-10 – Longitudinal and Lateral Acceleration During Trip (Corrected for Roll)

Figure 5-11 shows the relevant portions of the roll velocity data. As this graph shows, the roll velocity builds up during the trip phase over an approximately ½ second time frame from 2.5 to 3.0 seconds. The roll velocity then levels off to an approximately constant value of 400 degrees per second. This leveling off of the roll velocity is an indication that the trip phase has ended, the vehicle has become airborne, and the roll phase has begun. This plateauing of the roll velocity, which occurred at a time of approximately 3.005 seconds, was used to define the end of the trip phase. This point is identified in Figure 5-11.

Figure 5-11 – Roll Velocity Data
Next, the roll velocity data was integrated to obtain the roll angle of the vehicle body throughout the first 3.25 seconds of the test. Figure 5-12 shows these roll angles. The roll angle builds up over an approximate time frame of 2.5 to 3.0 seconds, achieving a roll angle of approximately 61 degrees at the end of the trip phase. Based on the determination that the trip phase ended 3.005 seconds after the vehicle was released, the video was reviewed and this time was identified. Figure 5-13 depicts a frame from the video showing the vehicle at the end of the trip phase. The video was also reviewed in conjunction with the physical evidence diagram of Figure 5-8 to identify Positions 1 (beginning of tire marks) and 4 (right front tire liftoff). These positions are shown in Figure 5-14 and Figure 5-15. The video was not conducive to determining when the right rear tire lifted off.
Based on the timing of Positions 1, 4, and 5 in the video, these positions were identified on the graph of the lateral and longitudinal accelerations during the trip phase (Figure 5-16). Based on the timing of these positions, the average lateral (ground plane) and longitudinal accelerations experienced by the vehicle were determined for the time frame from Positions 1 to 5 and for the time frame from Positions 4 to 5. It was concluded that from Positions 1 through 5, the vehicle experienced an average lateral acceleration of 1.09g and a longitudinal acceleration of 0.47g. Both of these were acting in a direction that would decelerate the vehicle. During the time frame from Positions 4 to 5, the vehicle experienced an average lateral acceleration of 1.69g and a longitudinal acceleration of 0.64g.
These measured accelerations demonstrate that there were longitudinal forces applied to the vehicle during the trip due to furrowing and steering affects (the significant rightward steer that was present throughout the trip phase is clearly visible in Figure 5-15). Such longitudinal forces have not customarily been included in analysis of the trip phase. Based on the analysis described in this section, however, their inclusion could potentially improve the accuracy of speed calculations related to the trip phase. Using the actual longitudinal accelerations, the speed analysis scenarios reported in Table 5-1 were recalculated and are reported in Table 5-2. The scenario that used Positions 3-5 as the trip phase used a longitudinal deceleration of 0.55g.

<table>
<thead>
<tr>
<th>Trip Phase</th>
<th>Actual Speed at Beginning of Trip (mph)</th>
<th>Trip Distance (feet)</th>
<th>Actual Deceleration Rate (g)</th>
<th>Calculated Normalized Lateral Tripping Force (g)</th>
<th>Average Slip Angle (deg)</th>
<th>Calculated Deceleration Rate Assuming Actual Longitudinal Force</th>
<th>Calculated Speed Assuming Actual Longitudinal Force (mph)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Positions 4-5</td>
<td>61.1</td>
<td>27.4</td>
<td>2.14</td>
<td>1.68</td>
<td>48</td>
<td>1.67</td>
<td>57.8 (-3.3)</td>
</tr>
<tr>
<td>Positions 3-5</td>
<td>64.3</td>
<td>45.5</td>
<td>1.58</td>
<td>1.38</td>
<td>44</td>
<td>1.35</td>
<td>61.7 (-2.6)</td>
</tr>
<tr>
<td>Positions 1-5</td>
<td>68.9</td>
<td>88.2</td>
<td>1.05</td>
<td>1.20</td>
<td>28</td>
<td>0.96</td>
<td>67.1 (-1.8)</td>
</tr>
</tbody>
</table>

Table 5-2 – Summary of Trip Phase Calculations with Actual Longitudinal Accelerations (Assumes Roll Phase Deceleration Rate of 0.363g)

One issue with the calculations reported in Table 5-1 and Table 5-2 is that they utilized a roll phase deceleration rate of 0.363g. This was the deceleration rate that Arndt reported for this test, and so, using this value allowed for the trip phase calculations to be isolated. However, this roll phase deceleration rate is outside the range that would typically be employed by a reconstructionist. A more typical value would be 0.44g. Table 5-3 summarizes calculations similar to those reported in Table 5-2 (using the actual longitudinal decelerations), but this time using a roll phase deceleration rate of 0.44g. Using 0.44g results in overestimating the speed at the beginning of the roll phase, but with any definition of the trip phase, it improves the accuracy of the calculated speed over calculations that ignore the longitudinal forces.
Table 5-3 – Summary of Trip Phase Calculations with Actual Longitudinal Accelerations
(Assumes Roll Phase Deceleration Rate of 0.44g)

<table>
<thead>
<tr>
<th>Trip Phase</th>
<th>Actual Speed at Beginning of Trip (mph)</th>
<th>Trip Distance (feet)</th>
<th>Actual Deceleration Rate (g)</th>
<th>Calculated Normalized Lateral Tripping Force (g)</th>
<th>Average Slip Angle (deg)</th>
<th>Calculated Deceleration Rate Assuming Actual Longitudinal Force</th>
<th>Calculated Speed Assuming Actual Longitudinal Force (mph)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Positions 4-5</td>
<td>61.1</td>
<td>27.4</td>
<td>2.14</td>
<td>1.68</td>
<td>48</td>
<td>1.73</td>
<td>61.8 (+0.7)</td>
</tr>
<tr>
<td>Positions 3-5</td>
<td>64.3</td>
<td>45.5</td>
<td>1.58</td>
<td>1.38</td>
<td>44</td>
<td>1.37</td>
<td>65.3 (+1.0)</td>
</tr>
<tr>
<td>Positions 1-5</td>
<td>68.9</td>
<td>88.2</td>
<td>1.05</td>
<td>1.20</td>
<td>28</td>
<td>0.96</td>
<td>70.3 (+1.4)</td>
</tr>
</tbody>
</table>

Source of the Longitudinal Forces

A 2017 *Collision* article explored the hypothesis that the longitudinal forces applied to the Isuzu during the trip phase were due to the significant rightward steering input that was present during the trip phase [Rose, 2017]. PC-Crash crash simulation software was used for this analysis. This software allowed us to easily incorporate the effects of steering on the speed loss during the trip phase and also to easily incorporate the three-dimensional characteristics of the terrain. A three-dimensional terrain was used for the simulation, which was based on Asay’s survey data. Manufacturer specifications provided the vehicle dimensions and the vehicle weight reported by Asay was used (4,266 pounds). Moments of inertia were estimated based on formulas in MacInnis [1997] and Allen [2003]. PC-Crash’s TM-Easy tire model was used with the default parameters. Representative shape parameters for the vehicle body of the Isuzu were measured from a computer model and entered for the simulation.

To generate the simulation in PC-Crash, the vehicle was placed at Position 1 (Figure 5-8) and a speed of 68.9 mph was entered. The yaw rotational rate and velocity direction at Position 1 were varied during the simulation process to optimize the match of the subsequent vehicle motion with the tire marks and furrows. During the simulation process, the results were judged based on how closely the calculated vehicle motion followed the tire marks and furrows of the final clockwise yaw that preceded the roll phase.

According to technical specifications for the Isuzu, the steering ratio was 18.7:1 and the minimum turning circle for the vehicle was 40.9 feet. For a rightward steering input, this implies a maximum input at the steering wheel of approximately 506 degrees. This would produce a rightward steer at the left front tire of approximately 27 degrees and a rightward steer at the right front tire of approximately 35 degrees. This steering input was entered into PC-Crash for the simulation. Brake factors of 1% were entered for each tire of the Isuzu to represent rolling resistance. No additional braking was applied during the simulation. Friction zones were constructed for the trip phase in order to generate the known speed loss over the trip phase while also matching the tire marks and furrows and the known roll angle and roll angular velocity of the vehicle at the end of the trip. Figure 5-17 is a screen capture from the simulation that shows the friction zones that we utilized. This approach is similar to the one described by Grimes in a 2006 study, though that study utilized HVE rather than PC-Crash.
Figure 5-17 - Friction Zones used in the PC-Crash Simulation

Figure 5-18 shows the vehicle motion in the optimized simulation. In this simulation, the speed of the vehicle at the end of the trip phase was 44.56 mph, close to the speed of 44.4 mph determined by Arndt. The roll angle at the end of trip was approximately 63.8 degrees and the roll velocity was 382 degrees per second, reasonably consistent with the values determined from the sensor data. In the simulation, the average lateral acceleration during the final clockwise yaw rotation of the vehicle was 1.24g and the longitudinal acceleration was 0.36g, both of which acted to decelerate the vehicle.
After optimizing this simulation with the rightward steering input, a simulation was generated with the rightward steering input removed. Without this steering input, we were not able to generate a simulation with a satisfactory and simultaneous match with the tire marks, the furrows, the vehicle speeds, and the roll angle and roll angular velocity at the end of trip. One attempt is shown in Figure 5-19. In this simulation, the match with the tire marks and furrows was acceptable, though not perfect. The speed loss during the trip, however, was significantly underestimated with the speed entering the roll phase being more than 11 mph too high. This indicates a significant drop in the overall deceleration level from the simulation that included the rightward steering input. The roll velocity at the end of the trip phase was also significantly underestimated at 230 degrees per second, rather than the actual 400 degrees per second.

The longitudinal accelerations in the simulation without steering were significantly less than in the previous simulation. The magnitude of the longitudinal accelerations in this simulation were consistent with the level that would be expected from rolling resistance and from traversing the slope of the roadway shoulder during the trip (less than 0.1g), but they were not anywhere near the level that actually existed during the test. Together, the simulation with steering and the one without steering confirmed that the longitudinal accelerations observed in the sensor data of the actual test were largely, though not completely, due to the significant rightward steering input that was present during the trip phase.

This finding indicates that there is not a need to develop any new models or formulas in order to incorporate these longitudinal forces into modeling of the trip phase. Accident reconstructionists already have a number of methods for incorporating the effects of steering on the deceleration rate of a yawing vehicle. Simulation is one of these [Rose, 2016]. Carter [2012] discussed several other calculation techniques that would allow a reconstructionist to calculate the effects of steering on the deceleration rate of a yawing vehicle. One difficulty would be knowing which direction, and the extent to which, a driver was steering during the trip phase. This is where simulation has a distinct advantage over spreadsheet calculations. In modeling the trip phase of the Rodeo test, the match with the physical evidence was significantly improved by including the rightward steering input. This is an indication that simulation can help detect the presence of a steering input during the trip phase. In cases where a vehicle begins yawing on asphalt before it enters the soil, striations in the tire marks on the asphalt may also be useful in determining which direction and the extent to which the driver was steering during the trip phase [Beauchamp, 2009 and 2016]. Techniques for striation analysis will be covered in the next chapter. In some instances, data from an event data recorder might also give the reconstructionist knowledge of the steering inputs.

Figure 5-19 - Vehicle Motion in Simulation without Rightward Steering Input
On-Road Rollovers

So far, this chapter has covered trip modeling appropriate for soil tripped and curb tripped rollovers. For calculating the speed loss during the trip phase, Equations (5.8) and (5.9) have been the centerpiece of the discussion so far. At this point, it is important to note that Equation (5.8) should not be applied to determine the speed loss for a rollover that is initiated on the roadway. Equation (5.8) will always yield a normalized lateral force that is greater than a vehicle’s static stability factor. However, the lateral forces generated between a vehicle’s tires and an asphalt or concrete roadway surface will nearly always be lower than the static stability factor. This means that, the lateral forces are not alone sufficient to initiate a rollover on the road surface. Suspension effects due to severe steering inputs are also playing a role in initiating the rollover. When a rollover occurs due to a combination of the lateral tire forces and suspension effects due to severe steering inputs, Equation (5.8) will overestimate the lateral forces, and thus, the speed loss during the roll initiation phase. In these instances, the roll initiation phase can be lumped into the loss of control phase and the speed loss calculated in the same way it is for the rest of the loss of control phase. Speed calculation methods for the loss of control phase are covered in Chapter 6.

References


Appendix

Figure 5-4 – Vertical Velocity v. Time, Linearly Increasing Trip Force (Left) and Sinusoidal Trip Force (Right) [SI Units]