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DEPARTMENT OF MECHANICAL AND NUCLEAR ENGINEERING

A COMPARATIVE ANALYSIS OF STATIC AND DYNAMIC TRANSIT BUS
ROLLOVER TESTING USING COMPUTER SIMULATION

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ABSTRACT

Rollover occurs when a vehicle rotates ninety degrees or more about its longitudinal axis. Studies have shown that although it is the type of accident that occurs least often by a wide margin, it is a close second in terms of accident types in which fatalities result. Although much research has been conducted on rollover in passenger vehicles and heavy trucks, one type of vehicle that scarcely enters the rollover discussion is the transit bus. Although buses aren't generally in a situation conducive to rollover, the severity of such an accident would be very high, especially in terms of passenger injuries. The National Highway Traffic Safety Administration (NHTSA) has led efforts to develop a rollover warning system based on the Static Stability Factor (SSF), a parameter that predicts the rollover threshold based on the vehicle's track width and the height of its center of gravity. In attempts to generate more accurate rollover predictions, static tests such as the Tilt Table test, and other dynamic tests, such as the NHTSA Fishhook test and the International Standards Organization (ISO) 3888 Double Lane Change test, were implemented.

In this analysis, a Tilt Table test is simulated with MSC.ADAMS/View software, along with the Fishhook 1a maneuver and the 3888 Double Lane Change maneuver. Inputs to the bus model presented include a driving force and a steering input. A comparative analysis is performed on the simulation results. Both the SSF and Tilt Table test yield inflated rollover threshold values. The Fishhook 1a test proves unsuitable for bus testing, while the 3888 Double Lane Change maneuver produces the most realistic results.

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CHAPTER 1: INTRODUCING ROLLOVER

1.1 The Problem of Rollover

Vehicle rollover is the single most dangerous occurrence on the road today. Statistics gathered by the Fatality Analysis Reporting System (FARS) and depicted in a National Highway Traffic Safety Administration (NHTSA) report [1] show that in 1999, 41,717 people were killed in light vehicle (under 10,000 lbs) accidents. In the same year, 10,142 were killed in light vehicle rollovers, approximately one-fourth of all light vehicle related deaths in that year. Between 1995 and 1999, only seven percent of light vehicle tow-away crashes involved rollover, but these accidents accounted for thirty-one percent of light vehicle occupant deaths [1]. **Figure 1** summarizes the distribution of light vehicle crashes in 1999. Frontal crashes occur more often than any other type of accident, nearly twice as often as side and rear accidents. Rollover accidents, on the other hand, occur approximately one-nineteenth times as often as frontal crashes.

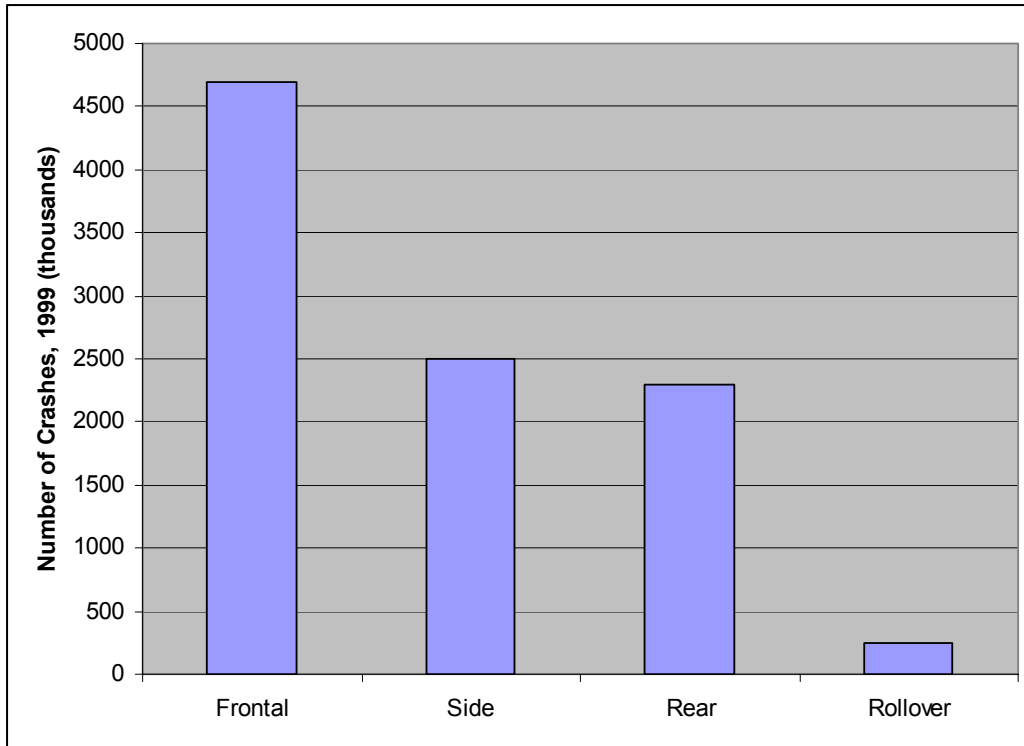


Figure 1. Distribution of light vehicle accidents in 1999 (NHTSA General Estimates System, 1999)

Further demonstrating the high fatality rate of rollover accidents, **Figure 2** represents breakdown of fatal crashes also in terms of crash type. While rollover is the least frequent of any type of accident, it is a close second to frontal crashes in terms of accidents in which a death resulted. Single vehicle rollover is also statistically more dangerous than multiple vehicle accidents. In 1999, single vehicle rollover was responsible for 43 occupant deaths per million registered passenger vehicles, while multi-vehicle accidents accounted for 10 deaths per million registered vehicles. An average of 19,000 people suffered injuries due to rollover from 1995 to 1999 [1].

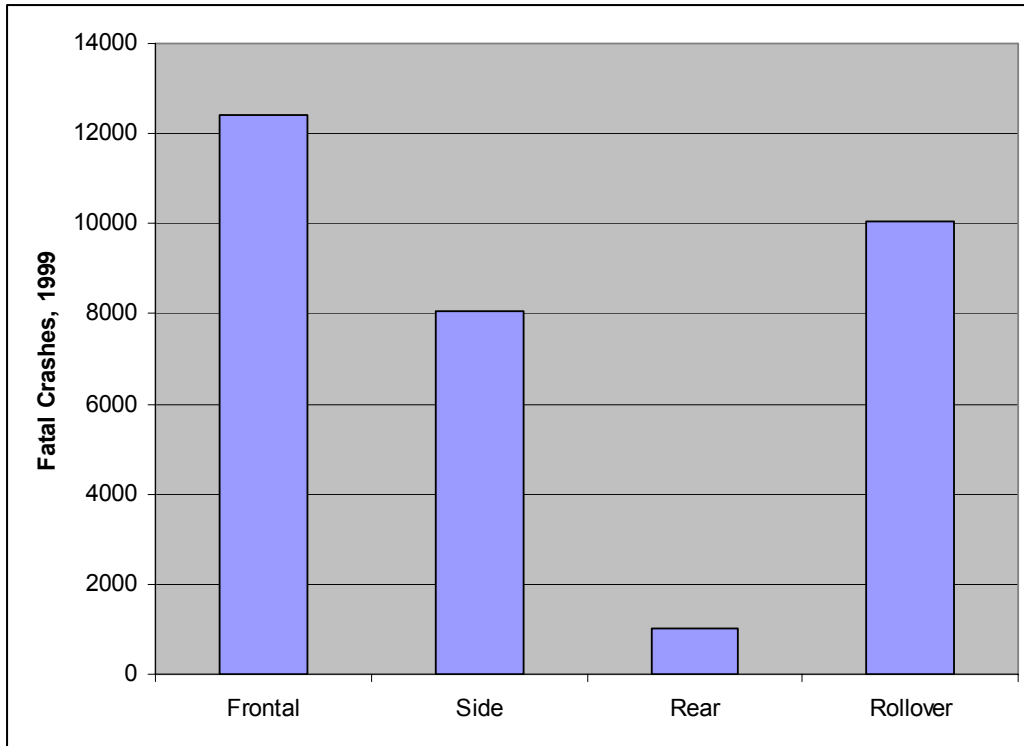


Figure 2. Distribution of fatal accidents in 1999 (NHTSA Fatality Analysis Reporting System (FARS), 1999)

While the aforementioned data pertained solely to the United States, a study by Richardson, et al., [2] in Australia showed that rollover has become a problem in Australia as well. Between 1996 and 1999, 22.78 percent of road fatalities involving cars, sport utility vehicles (SUVs), vans, four wheel drive trucks (4x4's), and motor homes were linked to rollover, and 12.17 percent of fatal crashes involved rollover. Richardson et al. claim that these numbers work out to about 1.87 fatalities for each fatal rollover event. In most accidents, serious or fatal injury is directly related to crash severity. In rollover, fatal or serious injury can still occur in low energy rollover accidents due to partial occupant ejection [2].

1.2 Transit Bus Rollover

Rollover in transit buses is a topic not often brought up in the discussion of vehicle rollover. Because light passenger vehicles are driven in a much more aggressive fashion that is more conducive to rollover, these vehicles are the focus of almost all rollover studies. An example of a statistical analysis of bus rollover was done by a group in Hungary. In this study, one of the main methods of obtaining information about the type of rollover in each specific accident was by checking local media coverage on the accident in question. Although this may be a good start to trying to tackle the bus rollover problem, it certainly has its flaws.

Other actual rollover tests in place today deal mostly with dropping a bus into a ditch to assess structural damage. By creating the safest possible cabin, manufacturers hope to reduce passenger injury. Although this is also a reputable strategy, there aren't many studies being done to assess the cause of bus rollover. Although rollover in passenger vehicles is becoming more well understood than in the past, buses aren't often in the same situations as passenger vehicles, but still experience rollover.

1.3 Rollover Vehicle Dynamics

1.3.1 Rollover Dynamics of a Rigid Vehicle on Flat Ground

Gillespie defines rollover as “any maneuver in which the vehicle rotates ninety degrees or more about its longitudinal axis such that the body makes contact with the ground [3].” In order to understand rollover, it helps to understand what

causes rollover. The understanding of the physics of rollover involves the study of vehicle dynamics and force balances which use models of varying complexity.

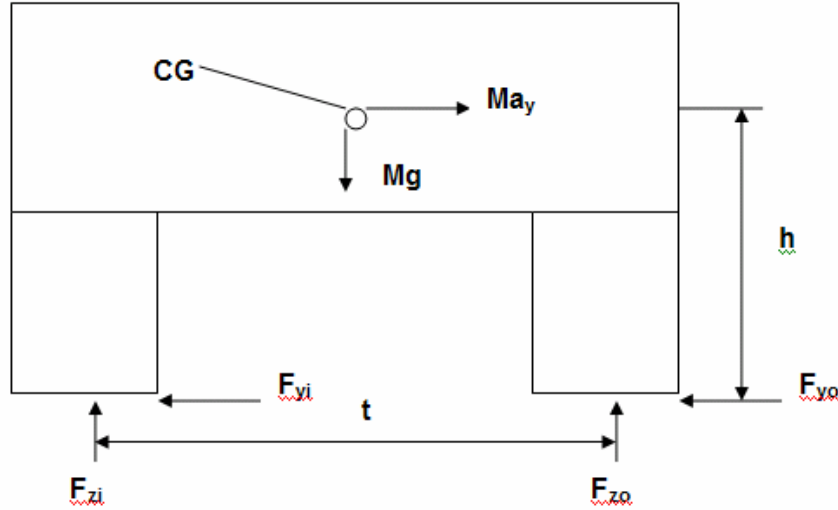


Figure 3. Free body diagram of rigid vehicle

Figure 3 shows the free body diagram of a rigid vehicle. In this case it is assumed that there is no deflection in the vehicle's suspension and tires, and the entire vehicle rolls about its longitudinal (front to rear) axis as a unit. In addition, it is assumed that the vehicle is in a steady turn, with no roll acceleration, and that the inner (F_{yi} and F_{zi}) and outer (F_{yo} and F_{zo}) tire forces, are, respectively, the combination of the inner and outer forces at the front and back tires. Cornering maneuvers are very favorable to untripped rollover because they produce a lateral acceleration perpendicular to the longitudinal velocity vector of the vehicle, acting at the vehicle's center of gravity (CG). This lateral acceleration, a_y in the diagram, is equal to the vehicle's forward velocity squared divided by its turning radius. A force couple is created on the vehicle between the centripetal force resulting from the vehicle's lateral acceleration, and the lateral forces acting on the tires of the vehicle,

F_{y0} and F_{yi} , which form to counteract the centripetal force in the lateral direction. Other forces shown are F_{zi} and F_{zo} , which can be thought of as normal forces counteracting the weight of the vehicle.

Assuming the vehicle is on flat ground, and taking moments about the outer tire's point of contact with the ground:

$$F_{zi} + Ma_y h - \frac{1}{2} Mgt = 0 \quad (1)$$

where h is the height to the vehicle's CG and t is the track width of the vehicle, the above equation can be solved for lateral acceleration in g's and simplified to:

$$\frac{a_y}{g} = \frac{t}{2h} - \frac{F_{zi}t}{Mgh} \quad (2)$$

1.3.2 Rollover Dynamics of a Rigid Vehicle on a Banked Road

Figure 4 shows the free body diagram of the same rigid vehicle, only this time on an incline. Much of the analysis is the same, only components of non-vertical and non-horizontal forces must be taken into account.

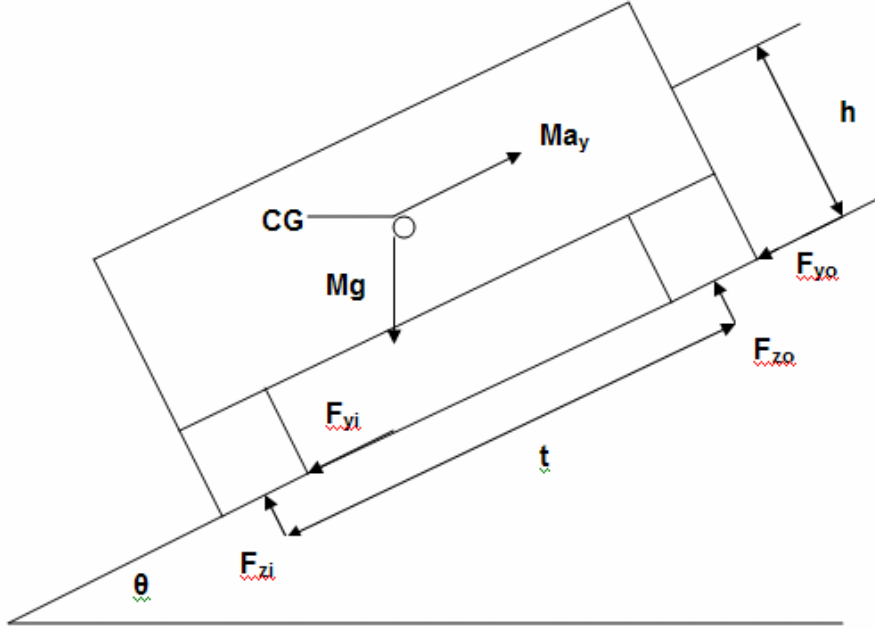


Figure 4. Free body diagram of rigid vehicle on banked road

Once again, taking moments about the point of contact of the outer tire with the ground:

$$Ma_y h + F_{zi} t - Mgh \sin \theta - \frac{1}{2} Mgt \cos \theta = 0 \quad (3)$$

Assuming small bank angles ($\sin \theta = \theta$ and $\cos \theta = 1$),

$$Ma_y h + F_{zi} t - Mgh \theta - \frac{1}{2} Mgt = 0 \quad (4)$$

Rearranging and solving for the dimensionless form of acceleration in terms of g's:

$$\frac{a_y}{g} = \frac{t}{2h} + \theta - \frac{F_{zi} t}{Mgh} \quad (5)$$

Equations (1) and (5) above represent the magnitude of the lateral acceleration for a steady state turn, without a bank angle (θ) and with a bank angle, respectively.

At the moment that a vehicle begins to roll outwards, the normal force, F_{zi} , becomes zero. For the initial case, on flat ground, equation (1) can be simplified to:

$$\frac{a_y}{g} = \frac{t}{2h} \quad (6)$$

and equation (5) can be simplified to:

$$\frac{a_y}{g} = \frac{t}{2h} + \theta \quad (7)$$

Assuming the vehicle is rigid, if the lateral acceleration of a vehicle in g's is equal to $t/2h$ on a flat surface, the vehicle will begin to roll. If the lateral acceleration is sustained at $t/2h$ for a long enough period of time, the vehicle will roll over. This value, $t/2h$, is also called the Static Stability Factor (SSF) or Static Rollover Threshold (SRT), and is often used to predict the roll stability of vehicles. The SSF will be discussed in more detail later, along with arguments for and against its widespread use.

Figure 5 shows the instability curve of a rigid vehicle. The curve is actually a representation of equation (7). It is a straight line graph connecting the two possible extreme situations, a_y equaling zero, or θ equaling zero. Because the vehicle is rigid, the angle at which the vehicle's vertical axis is offset from the normal vector to the ground (roll angle) will be equal to the angle at which the road is banked. Only in the case of a rigid vehicle can the bank angle also represent the vehicle's roll angle.

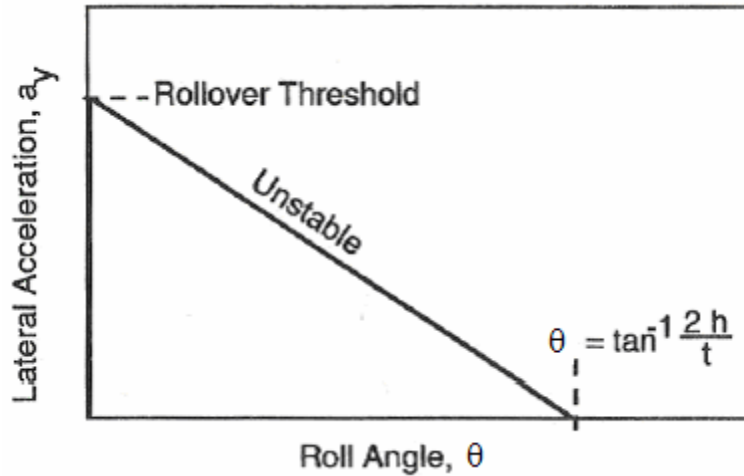


Figure 5. Equilibrium curve of a rigid vehicle [3]

The graph shows that while driving with no bank angle, the vehicle becomes unstable at lateral accelerations equal to the rollover threshold. With no lateral acceleration, the bank angle must be equal to the arctangent of $2h/t$ for the vehicle to roll. This point corresponds to the point during rollover when the vehicle's CG passes the plane perpendicular to the road containing the line connecting the two wheels still touching the ground. At this point, rollover is irrecoverable [3].

1.3.3 Problems with Rigid Body Modeling

Because in the previous model the effects of suspensions and tires were neglected, the rigid body model predicts vehicle rollover at a higher lateral acceleration than is true in reality. In some cases, mostly vehicles like sports cars or smaller cars with low CG heights and larger track widths, the rigid body model suggests that the necessary lateral acceleration required for rollover is outside the friction limits of the tires of a vehicle. This means the vehicle will slide instead of roll, which isn't always the case. This idea does help describe why trucks and

vehicles with higher CG heights are more susceptible to rollover than smaller cars with lower CG heights. A vehicle with a high CG height is able to reach the necessary lateral acceleration ($t/2h$) for rollover within the friction limits of its tires, before the vehicle begins to slide [3]. This concept helps explain the recurring trend shown in accident data, that SUVs have a much higher rollover propensity than other light vehicles. SUVs have a high CG height compared to many other vehicles, which lowers their rollover threshold, making them easier to roll. **Figure 6**, taken from [1], shows the rollover rate of vehicles by type. SUVs and trucks, vehicles with high CG heights, roll much more often than smaller cars, with a little over one more SUV rolling per 100 crashes than pickup trucks [1].

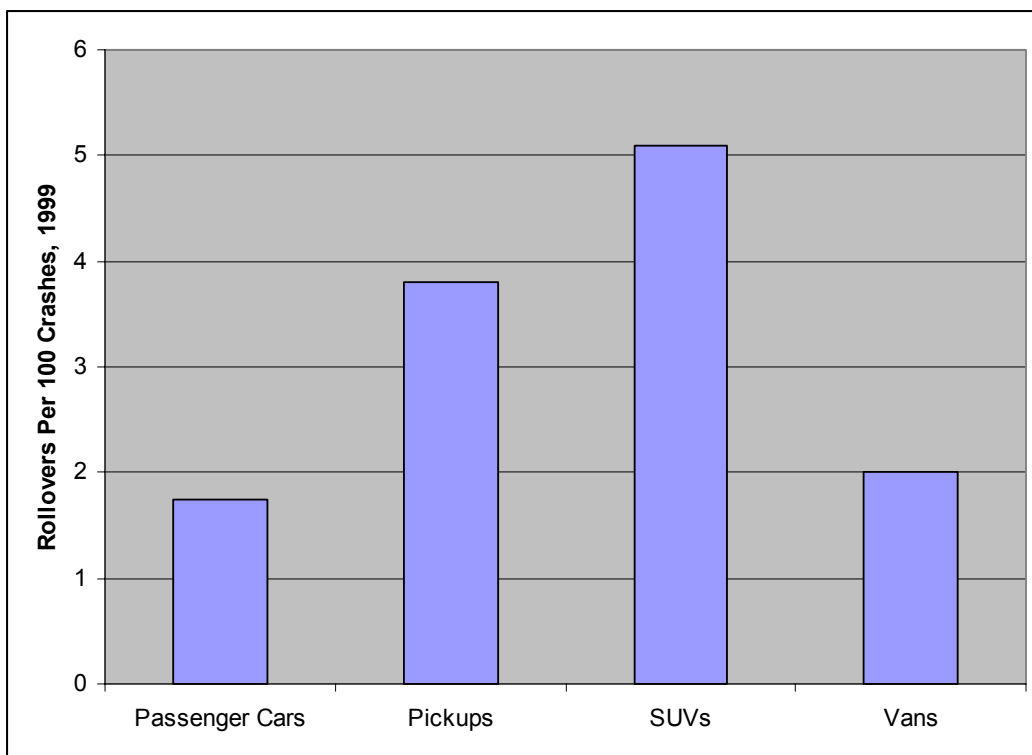


Figure 6. Rollover rate by vehicle type (NHTSA General Estimates System, 1999)

1.4 Rollover Dynamics of a Suspended Vehicle

Since modeling a vehicle as a rigid body ignores the effects of tires and suspensions, which in turn overestimates the necessary lateral acceleration to roll the vehicle, a better model would include the possibility of the sprung mass (the mass of the vehicle supported by the suspension) rolling before the wheels lift. In the real world, during cornering, weight is shifted to the outside wheels. As this happens, of course, the center of gravity of the vehicle also moves toward the outside wheels as the sprung mass begins to roll in that direction. As shown in **Figure 7**, the free body diagram of a suspended vehicle in a cornering maneuver, the weight of the sprung mass, acting at the vehicle's CG, is the only force acting against rollover. As the sprung mass begins to roll due to cornering, and the CG begins to shift, the moment arm from the point of contact between the road and outside wheel and the CG begins to shrink. As the moment arm gets smaller, the overall moment resisting rollover due to the vehicle's weight decreases as well. This effect tends to increase the vehicle's rollover propensity, decreasing the lateral acceleration necessary to roll the vehicle [3].

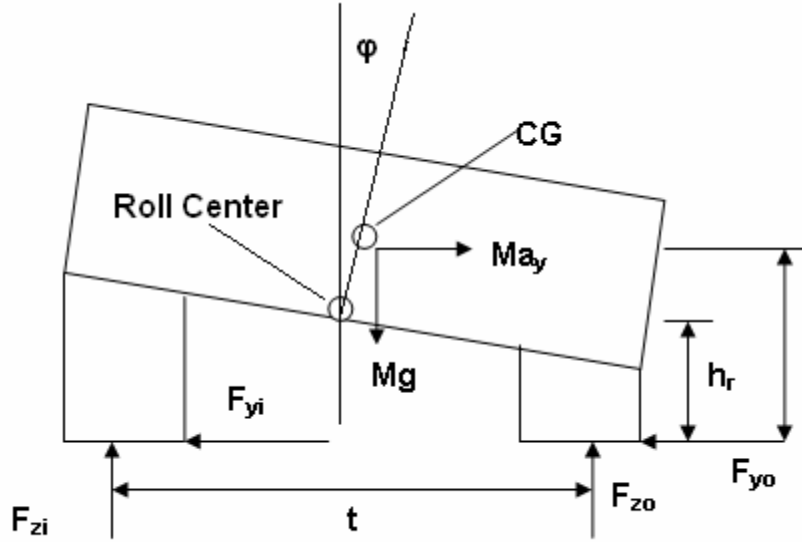


Figure 7. Free body diagram of a suspended vehicle during cornering

Another new development in this model includes an imaginary point called the roll center. This point represents where the mass of the vehicle's body is connected to either axle (there are two roll centers, one in front and one in back). The body is said to roll or pivot around the roll center. It is also the point at which lateral forces are transferred from the axle to the sprung mass. A new variable, h_r , the height to the roll center, also arises. In this case, ϕ is the angle that the body has rolled due to cornering. Once again, neglecting differences between front and rear, and summing moments about the point where the outer tire contacts the ground and assuming small angles:

$$Ma_y - Mg \left[\frac{t}{2} - (h - h_r)\phi \right] = 0 \quad (8)$$

Solving for a_y/g :

$$\frac{a_y}{g} = \frac{t}{2h} \left[1 - \frac{2}{t}(h - h_r)\phi \right] \quad (9)$$

The lateral acceleration necessary to roll a suspended vehicle is therefore equal to the rollover threshold previously discussed, reduced by $\left[1 - \frac{2}{t}(h - h_r)\phi\right]$.

Although this model accounts for body roll prior to wheel lift, it is only a better model than the previously discussed rigid body model. There still are better but far more complex models available. A more accurate rollover model would include a roll moment of inertia about the CG. This type of model is useful for modeling a step input of lateral acceleration. With a sudden lateral acceleration input, the roll angle of the vehicle responds like a second-order system, as shown in **Figure 8** [3].

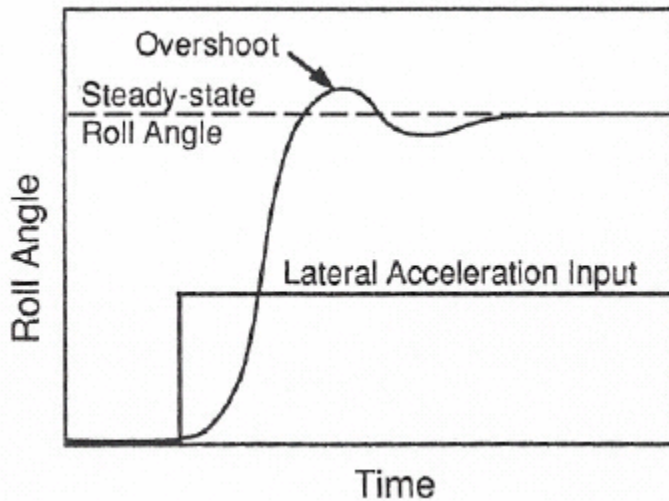


Figure 8. Suspended Vehicle Subject to a Step Lateral Acceleration Input [3]

Like an underdamped system, the roll angle approaches its steady-state value, overshoots, then settles on the steady-state value after a few oscillations. This overshoot also accounts for the lower rollover threshold that exists in reality [3]. The amount that suspension and tire inclusion actually affect the magnitude of the

necessary lateral acceleration to produce rollover is often dependent on the vehicle. Typical overestimations of this acceleration range from 10 percent [3] to 15 percent [1].

1.5 Rollover Studies and Preventative Actions

1.5.1 Factors Contributing to Rollover

There have been a few statistical studies done to help find which factors or combinations of factors lead to a situation where rollover is probable. One such study was done by Donelson, et al [4]. It was found that factors associated with high relative risk of rollover as compared to that of a single vehicle accident included: posted speed limit greater than 45 mph, rural driving environment, risky driver behavior, and vehicle rollover threshold ($t/2h$). It was also found that in general, the driver's behavior and the surrounding environment had a stronger correlation to rollover than vehicle parameters or characteristics did [4].

1.5.2 Derivation of the Static Stability Factor

Studies such as this one show that there are far too many factors that make it difficult to control rollover, and tracking down all the combinations that lead to a rollover situation is most likely fruitless. While the driver's reactions and the road geometry are impossible to control, automobile manufacturers can control the vehicle's parameters and geometry to try to reduce rollover as much as possible. One previously mentioned group that has been involved in the study of rollover is the NHTSA, which has been investigating rollover since 1973. Some of NHTSA's

milestones in rollover research include studies with the National Academy of the Sciences in the mid-1990's dealing with the high roll propensity of SUVs, which was brought center-stage around that time due to the sudden SUV sales explosion. After a few years of research, SUV warning labels were issued to vehicles under a certain SSF starting in 1999. In 2001, NHTSA came up with a five star vehicle rating system based on a particular vehicle's SSF, and data gathered over time correlating the probability of rollover per single vehicle crash with that particular SSF. The scale breakdown is as shown in **Figure 9**.

	<u>Rollover Risk</u>
Five Stars	Less than 10%
Four Stars	Between 10% and 20%
Three Stars	Between 20% and 30%
Two Stars	Between 30% and 40%
One Star	Greater than 40%

Figure 9. Summary of NHTSA rollover rating system

1.5.3 Limitations of the Static Stability Factor

These rollover ratings have been incorporated into NHTSA's New Car Assessment Program (NCAP), the main source of vehicle safety made available to consumers by the US federal government [1]. Although SSF has a strong correlation with vehicle rollover, automobile and consumer groups have expressed concern that SSF ratings do not take the dynamic behavior of vehicles into account. In addition, Richardson, et al. [2] brings up a valid point of concern with the five star system, as can be see in **Figure 10**.

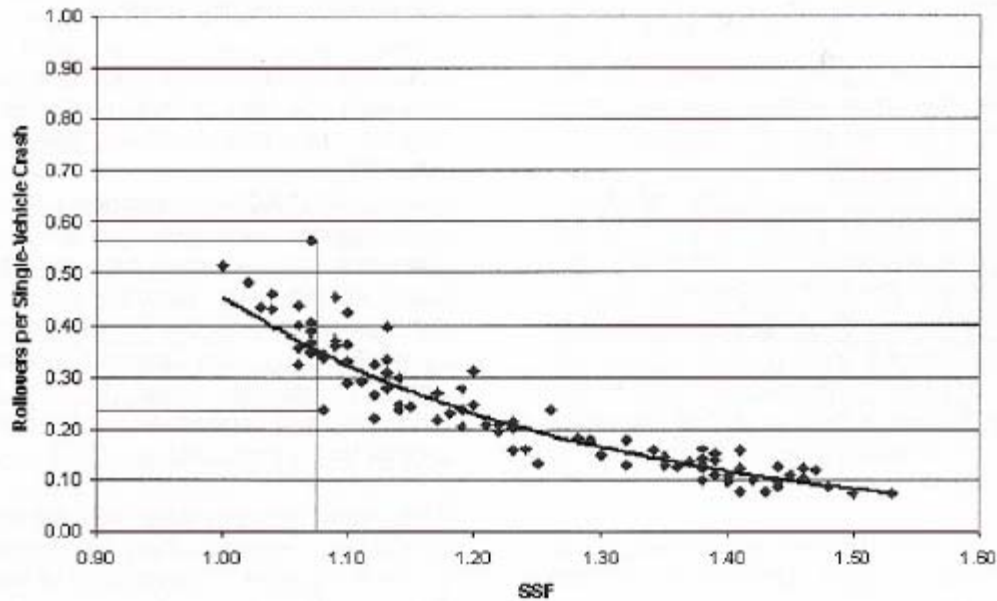


Figure 10. NHTSA Data on Rollovers per Single Vehicle Crash [2]

As can be seen from the graph, the data corresponds pretty well to the best fit line shown, which is the basis of the SSF system. There are certain points, however, such as at an SSF of 1.07, where the five star system can be deceiving. The graph shows a point far above the best fit line and a point far below the best fit line. This means that although both vehicles have the same SSF and receive the same rollover rating, one vehicle rolls over thirty-three percent more often than the other. Although the five star rating system seems to be a step in the right direction in terms of informing consumers, there are still some flaws in the system [2].

Fortunately for those who feel SSF is insufficient as a rollover rating method, working in parallel with the SUV warning label effort, NHTSA initiated a project to develop a series of tests that would evaluate the dynamic rollover propensity of a given vehicle, based on actual vehicle performance. In a project that was partly initiated by a Consumer's Union request for more information on rollover to made

available to the public, it was found that “several maneuvers appear to be able to discriminate between vehicles [that have] low static and dynamic rollover propensity measures and those that do not [1].” NHTSA’s work with dynamic testing has continued in response to the Transportation Recall Enhancement, Accountability, and Documentation (TREAD) Act of November 2000, which has deemed it necessary to continue to develop dynamic tests for consumer information on rollover. Some of the maneuvers referred to in the project will be described later, such as the J-Turn and Fishhook.

1.6 Defining Static and Dynamic Rollover Tests

Vehicle parameters do not always tell the whole story when it comes to rollover, as previously discussed. There is no better way to actually understand how a vehicle will respond to certain conditions than to physically test the vehicle itself. According to NHTSA [1], rollover testing can be broken down into two types: Static and Dynamic.

Static Testing is performed in the laboratory. It may involve the measurement of vehicle parameters (e.g., center of gravity height, track width) that are then combined to yield static metrics related to a vehicle’s rollover propensity – for example, static stability factor (SSF). Alternatively, static tests of entire vehicles, such as tilt table and side pull tests may be performed to obtain data that can be correlated with a vehicle’s rollover propensity.

Dynamic Testing is performed on a test track and involves driving maneuvers. Although dynamic tests are potentially helpful in understanding the events

immediately preceding rollover, they are expensive and require safety precautions for test drivers. Furthermore, repeatability may be difficult to achieve. In view of the challenges associated with dynamic testing, computer simulations have been undertaken using mathematical models to predict vehicle behavior associated with rollover [1].

1.7 Static Rollover Testing

While static testing includes some of the previously mentioned parameter analysis, there are also a few static tests that require the physical presence of the vehicle. Some of these include the side-pull test, the centrifuge test, and probably the most common static test on a vehicle, the tilt table test. All of these tests are intended to determine the rollover threshold of a vehicle, but all achieve the objective in different ways. Many agree that tests such as these are more realistic because effects such as tire and suspension compliance are included, but because these tests are still simulations of the real thing, there are still some trade-offs.

1.7.1 Side-Pull Test

Figure 11 shows a possible set up for a side-pull test. In order to simulate lateral forces acting at the vehicle's CG during cornering, the vehicle is pulled laterally, perpendicular to its longitudinal axis at its CG height. The pulling force is increased until the wheels opposite the side being pulled lift off the ground. By determining the force required to lift the wheels, the necessary lateral acceleration to

cause rollover can be determined. One drawback of the side-pull test is that it is difficult to perform [1].

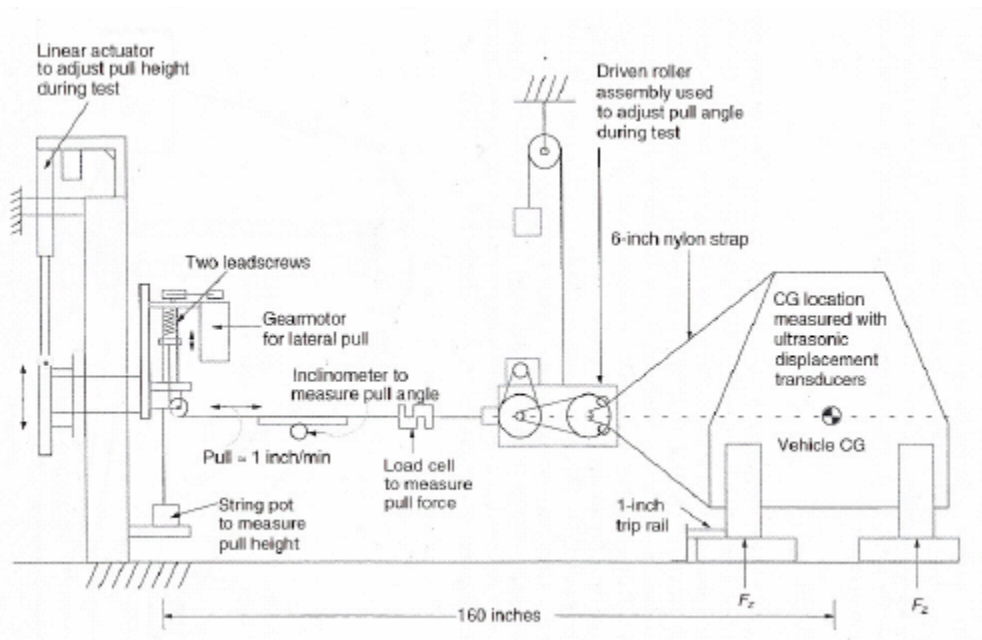


Figure 11. Possible Side-Pull Test Set Up [1]

1.7.2 Centrifuge Test

Another static test is the centrifuge test. This test uses a motorized shaft, connected to a platform that revolves around the shaft. The vehicle is placed on the platform, and the angular velocity of the shaft is increased until the vehicle's lateral acceleration causes it to tip over on its outside wheels. In this case, the lateral acceleration is found by converting angular velocity to longitudinal velocity, which is then squared and divided by the distance from the shaft to the center of gravity of the vehicle [1]. A possible set up for this test can be seen in **Figure 12**.

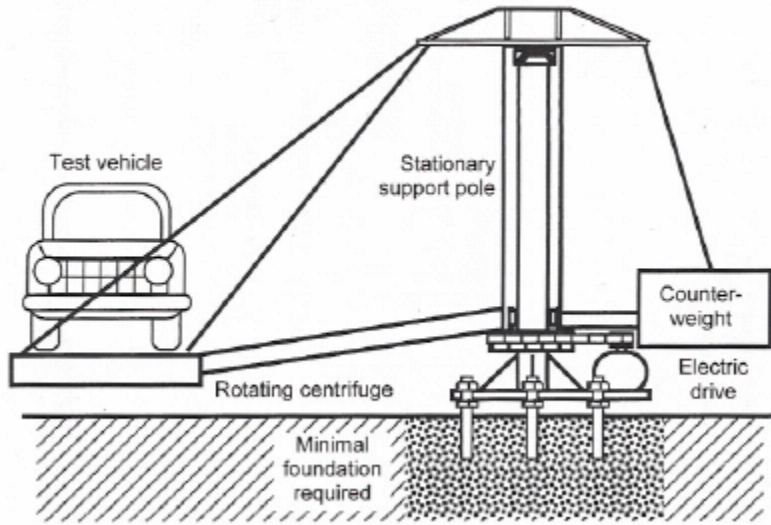


Figure 12. Possible Centrifuge Test Set Up [1]

1.7.3 Tilt Table Test

One of the better known static rollover tests involving the testing of an actual vehicle is the tilt table test. In this test, the vehicle in question is placed on a horizontal platform, parallel with the ground that slowly rotates about one of its edges. **Figure 13** shows a simplified diagram of how a tilt table test would look. When the test starts and the vehicle is sitting on flat ground, the weight of the vehicle is directly perpendicular to the plane of the ground. As the table begins to tilt, and the weight begins to have components both perpendicular and parallel to the rotating platform. The component of weight perpendicular to the platform represents the weight of the vehicle, while the component parallel to the platform represents the lateral force experienced during cornering. The increased tilting of the platform represents a vehicle going through an increasingly severe turning radius [5]. The platform is tilted at a constant rate until the uphill wheels of the vehicle lift off the platform. Since $Mg \sin \theta$ represents lateral force and $Mg \cos \theta$ represents the weight

of the vehicle, the lateral force acting on the bus divided by gravity is equal to $\tan \theta$. According to a Transport Engineering Research New Zealand Limited (TERNZ) study [6], the tilt rate of the platform should be no greater than 0.25 deg/s.

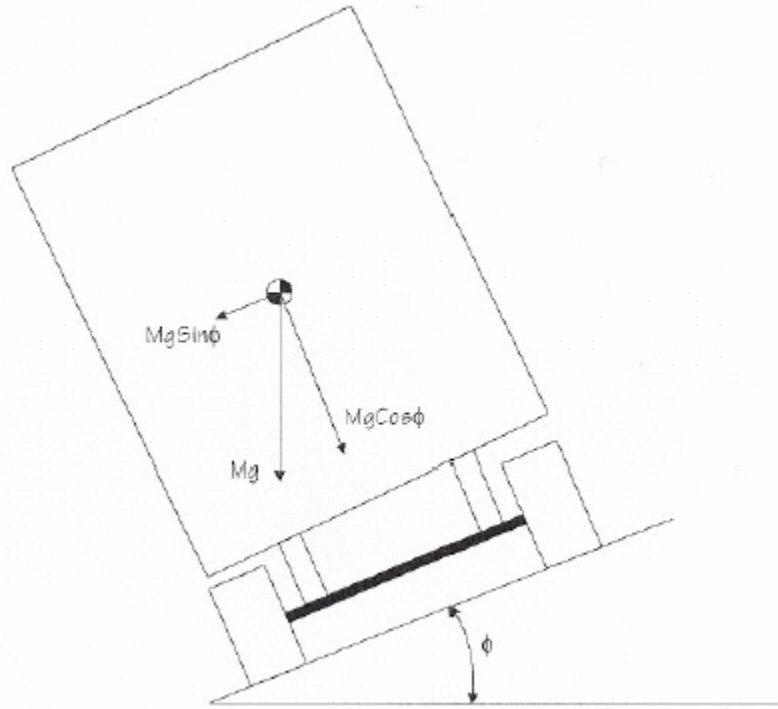


Figure 13. Forces Acting on the CG of a Truck During a Tilt Table Test [6]

According to the University of Michigan Transportation Research Institute (UMTRI) study entitled *Rollover of Heavy Commercial Vehicles*, both the lateral and vertical forces acting during the tilt table test are smaller than they would be in reality, and the magnitude of this scaling depends on the degree of tilt present in the table. The study also states, however, that the scaling “has multiple effects which, although they tend to cancel one another, can nevertheless reduce the accuracy of the experiment [5].”

In realistic situations, the weight of the vehicle is always perpendicular to the plane of the road. In the tilt table test, the closer the component of the weight perpendicular to the tilting platform to the actual weight, the more accurate the results will be. This means that the tilt angle should be as close to zero as possible, since $\cos 0 = 1$. In this way, the tilt table test generally better suits heavier vehicles, that tend to roll under lower lateral accelerations, and therefore lower tilt angles [5].

Although tilt table tests work well in the right conditions, there are a few arguments against their use as a valid roll predictor. Some argue that as the table is tilted, the total weight put on the tires drops, along with the loads on the suspension, which causes the vehicle's center of gravity to move farther away from the tilt platform, which wouldn't happen on flat ground. This in turn causes the vehicle to roll sooner than would happen in reality.

There is a fundamental argument present against all of the static tests. In order to achieve the best results in each test, the front and rear roll stiffness should be balanced in order to have both wheels on the same side lift at the same time. The problem with this is that actual vehicles aren't set up this way, because such a set up sacrifices vehicle handling [1].

1.8 Dynamic Rollover Testing

Since even physical static testing of a vehicle, although quite beneficial to research, creates controversy, the only method of rollover testing that shows how a vehicle would react in reality is dynamic testing. In dynamic testing, the vehicle is how it would be in the real world. One of the biggest problems with dynamic testing,

however, is creating a standardized test that fairly distinguishes between vehicle performances that can be repeated. A study by Forkenbrock, et al. [7], examines various dynamic tests proposed by NHTSA to serve as a standard by which vehicles are measured. In each rollover test, the maneuver is first performed at a lower speed, such as 35 mph, and the speed is increased in intervals of 5 mph until the vehicle experiences wheel lift off. At this point, the maneuver is performed again at speeds reduced by one mph until the vehicle no longer experiences wheel lift-off, at which point the exact speed at which rollover begins can be pinpointed [7].

1.8.1 The Slowly Increasing Steer Maneuver

The “slowly increasing steer” maneuver is the maneuver suggested by NHTSA to characterize the vehicle’s lateral dynamics, and is not considered an actual rollover test. A graphical representation of this maneuver can be seen in **Figure 14**.

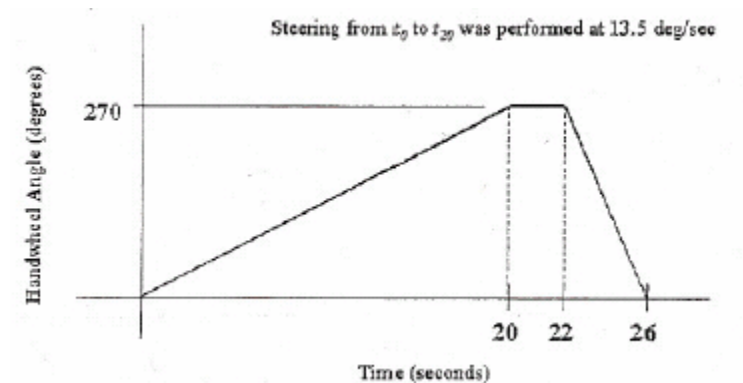


Figure 14. Handwheel Angle vs. Time for Slowly Increasing Steer Maneuver [7]

In this test, the vehicle is driven in a straight line at 50 mph, when at a given instant, the steering wheel’s angular position is increased linearly from 0 to 270 degrees at 13.5 degrees per second. The wheel is then held constant for two seconds, before being returned to zero degrees at the driver’s convenience. A few trial runs are

performed, and the average steering angle at which the vehicle experiences 0.3g's of lateral acceleration, the Slowly Increasing Steer angle (SIS angle), is recorded and used as input for the NHTSA rollover tests, such as the J-Turn test and the Fishhook tests.

1.8.2 The J-Turn Maneuver

A graph of steering wheel angle against time for the J-Turn maneuver, which has a long history of industry use, can be seen in **Figure 15**.

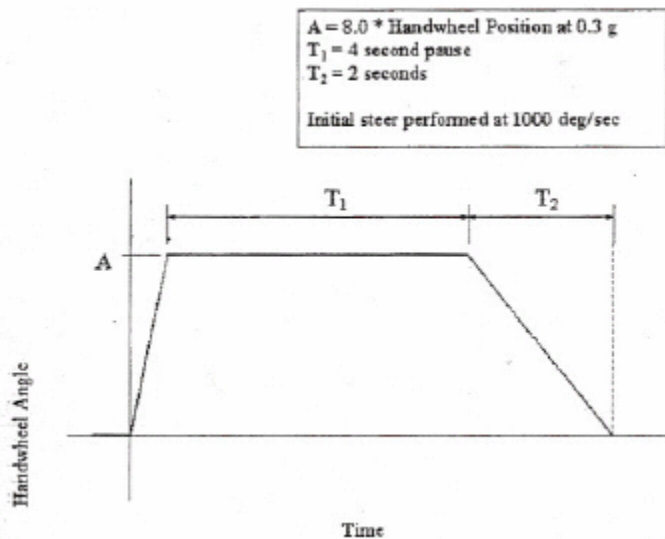


Figure 15. Handwheel Angle vs. Time for J-Turn Maneuver [7]

In this maneuver, the vehicle is driven in a straight line, until the steering wheel is ramped to 8.0 times the SSI angle at a rate of 1000 degrees per second. The wheel is held at this angle for four seconds, and is then turned back to zero degrees at a steady rate during the following two seconds. Forkenbrock, et al. [7] found that the J-Turn maneuver was the most repeatable of all rollover resistance maneuvers performed in

the study, but the maneuver does not always get vehicles to tip within the given speed limits. This maneuver is similar to driving on cloverleaf exit or entrance ramps or tightly curved roads at substantial speeds. One drawback to this test, however, is that drivers would rarely perform a maneuver like this in reality without hitting the brakes [7].

1.8.3 The Fishhook 1a Maneuver

The next maneuver proposed by NHTSA is the Fishhook 1a test, or the Fixed Timing Fishhook, which can be seen in **Figure 16**.

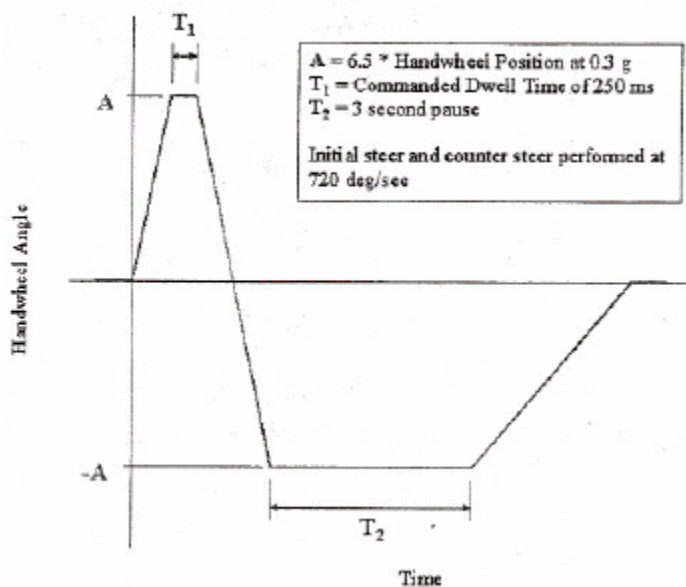


Figure 16. Handwheel Angle vs. Time for Fishhook 1a Maneuver [7]

After driving straight, the steering wheel is ramped up to 6.5 times the SSI angle at a rate of 720 degrees per second. This angle is held for 250 milliseconds, and then the wheel is ramped to -6.5 times the SSI angle in the opposite direction (a total angular

displacement of 13 times the SSI angle), again at 720 degrees per second. This angle is held for three seconds before the wheel is brought back to zero degrees. This maneuver had “excellent” repeatability according to Forkenbrock, et al.[7], and was one of only two maneuvers to cause two wheel lift off for vehicles in the above 1.13 SSF range. One downside is that some vehicles may need more than a 250 millisecond dwell time after the first steering input to achieve optimum rollover data. This test is similar to the steering input a driver would produce in trying to regain lane position on a two lane road after dropping two wheels off onto the shoulder [7].

1.8.4 The Fishhook 1b Maneuver

In trying to correct the insufficient dwell time problem that the Fishhook 1a test provided, NHTSA suggested the Fishhook 1b test, or the Roll Rate Feedback Fishhook, seen in **Figure 17**.

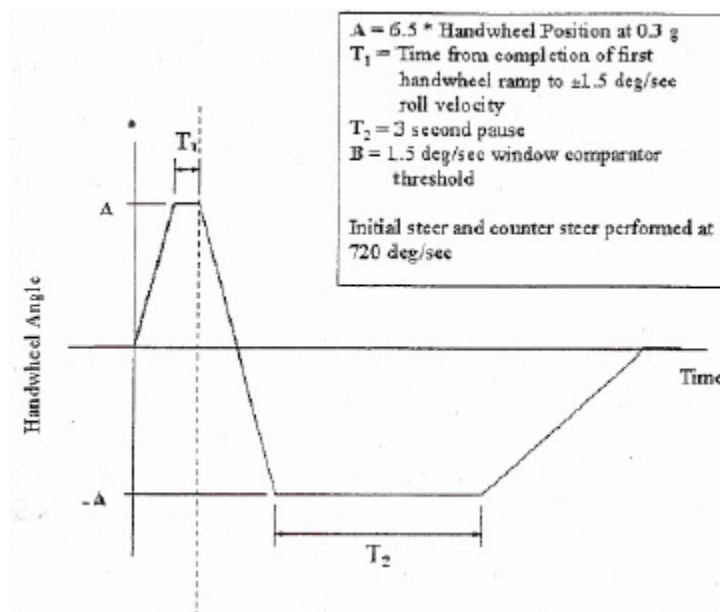


Figure 17. Handwheel Angle vs. Time for Fishhook 1b Maneuver [7]

The steering input in this maneuver is exactly the same except for the initial dwell time after the first steering input. In this case, the steering reversal is initiated after the first wheel ramp when the roll rate of the vehicle is less than or equal to 1.5 degrees per second. With this feedback system, the test is able to adjust to changes in parameters such as suspension between each vehicle tested. This test is the other maneuver known to NHTSA to cause to wheel lift off for vehicles in the above 1.13 SSF range [7].

1.8.5 The ISO 3888 Double Lane Change Maneuver

Although NHTSA is one of the leaders in rollover testing, other organizations outside of NHTSA are experimenting with dynamic testing. The International Standards Organization (ISO) has created a maneuver known as the ISO 3888 Double Lane Change Maneuver. Unlike maneuvers proposed by NHTSA, which are based on steering inputs based on time, the ISO 3888 maneuver is based on steering inputs based on road geometry, in the form of a standardized lane change course. Geometry for the ISO 3888 maneuver can be seen in **Figure 18**.

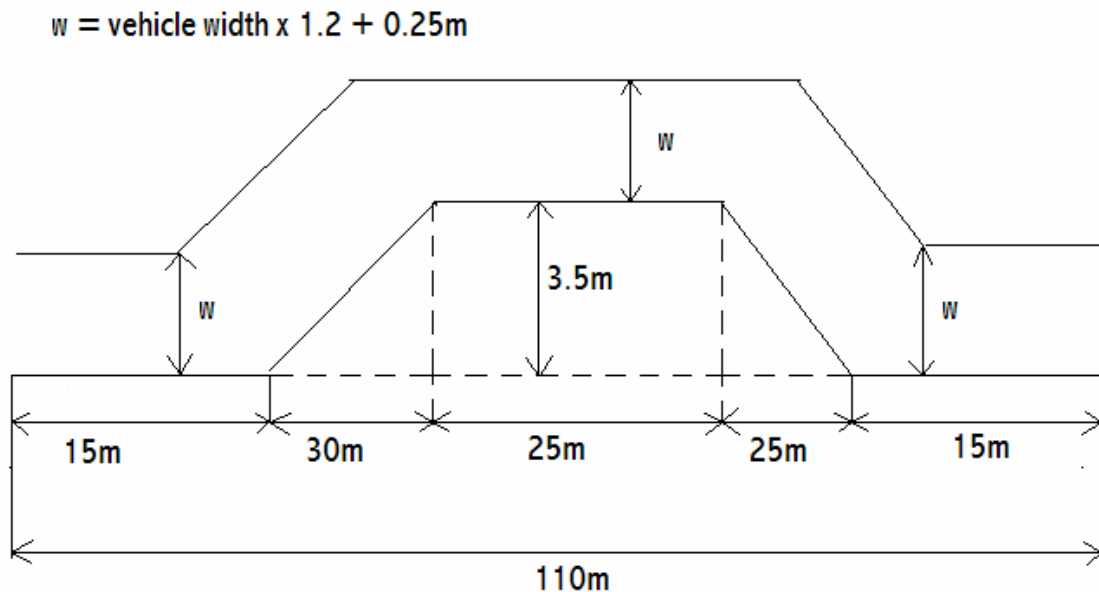


Figure 18. ISO 3888 Double Lane Change Maneuver Geometry

This maneuver has its benefits in that it is modeled after a maneuver often duplicated in real life, and is a test consumers would be able to identify easily with. One downside is that the course geometry is fixed, and vehicles with low roll propensity may need to reach high speeds before roll is imminent.

1.9 Benefits of Simulation

As fast and efficient computing hardware and software become available, computer simulation of vehicle dynamics related topics is becoming more and more common. Software such as MSC.ADAMS and Carsim are becoming so powerful that every part of a vehicle that has an effect on the outcome of a test in real life can be accounted for and simulated on a computer. Although real world testing has many advantages, software simulation is making a great case for itself. Some of the advantages of simulating with software over performing an actual test include:

- Cost
- Driver Safety
- Test Repeatability
- Changing of Vehicle Parameters

Although testing at an actual test track may be necessary in some cases, it is very expensive, with each test costing more money. Although simulation software may be expensive, it is a one time cost, after which infinite free test runs are available. In addition, vehicle damage is not a concern when dealing with simulations. In the real world, tests take time to set up and perform, and if there are vehicle parameters that need to be changed between tests, even more time is necessary. Running additional tests and changing vehicle characteristics is simply a matter of clicking and typing when using a software simulation. One of the most important things simulation brings to the table is the lack of a driver. Not only does this take out any human error present, it doesn't put a test driver at risk when performing one of these risky maneuvers.

1.10 Simulating Transit Bus Rollover

In researching vehicle rollover, one type of vehicle rarely encountered is transit buses. There could be a variety of reasons for this. With the issue of SUV rollover becoming so prevalent in recent history, light passenger vehicles, those bought by everyday consumers, have been one of the main focuses of rollover research. This goes hand in hand with public demand for information on light passenger vehicle rollover, since the majority of people would most likely request

rollover information pertaining personal vehicles rather than rollovers involving public transportation. The environment transit buses operate in also affects the research done on them. In most cases, buses are not put in situations conducive to rollover, such as city driving. City or town driving does not often allow the speeds combined with the cornering maneuvers-lateral accelerations- necessary to create a rollover situation. Buses, however, are like any other vehicle, and have the ability to roll given the right conditions. If they do rollover, however rarely, the consequences (injuries and fatalities) per accident could be much worse since they will likely involve many passengers.

In light of the lack of rollover information pertaining to transit buses, the remainder of this paper will deal with static and dynamic on-road, untripped simulation of transit buses. The model used in simulation was created in MSC.ADAMS/View by graduate students at the Pennsylvania Transportation Institute at The Pennsylvania State University. Simulation software will be used to make comparisons between the rollover propensity of a transit bus determined by one static test and two dynamic tests. The static test used will be the Tilt Table Test, in which roll angle and simulated lateral acceleration will be measured. In order to compare tests from different organizations, the dynamic tests used will be the Fishhook 1a maneuver and the ISO 3888 maneuver. Variables such as roll angle, lateral acceleration, speed, and yaw rate will be compared. These results will also be compared to the SSF derived in **Section 1.3.2**.

CHAPTER 2: ROLLOVER SIMULATION

2.1 Static and Dynamic Simulation

With so much discussion about the advantages and disadvantages of static and dynamic rollover testing, the preferred way to compare the two is by actually performing the tests and analyzing the results. Although the best data would be collected from taking measurements from a real vehicle on an actual test track, simulation provides an often more feasible method of testing a vehicle, in a way that is highly repeatable. As mentioned in **Section 1.10**, the vehicle to be tested through computer simulation is one that is rarely brought up in rollover discussion, a typical transit bus. The bus has a mass of approximately 17,200 kg, a track width of 2.2 m, and a CG height of 0.842 m.

2.2 Tilt Table Test

2.2.1 Creating the Tripping Force

The tilt table test was performed with a constant platform angular velocity of 0.2 degrees per second, within the suggested range. In reality, many vehicles slide before rolling, so when an actual tilt table test is performed, the platform contains a tripping mechanism to ensure that the vehicle does indeed roll and doesn't slide off the platform before it tips. Often the tripping mechanism is simply a barrier that is a few inches in height. During simulation, it was discovered that like many vehicles, the bus did slide before rolling, and something would be needed to prevent lateral translation. A standard tripping barrier could not be used in simulation, however,

because the tires used on the ADAMS bus model could only take into account one point of contact. From that point of contact, the wheel-road interface, forces and moments on the tire are calculated.

Preventing the bus from sliding down the ramp was accomplished through force balance. The bus began to slide because the component of its weight parallel to the platform overcame the lateral forces on the tires due to friction at the tire-road interface. Realizing that the tripping barrier simply provided a force on the tires that increased directly with the increasing platform angle, a similar force could be created to simulate that force, keeping the bus in place until it tipped. A free body diagram is shown below in **Figure 19**, including only forces in the direction parallel to the tilt platform. In order for the bus to remain in place before rolling, the newly created force, F_{barrier} had to equal the difference between $Mg\sin\theta$ and $F_{y \text{ total}}$, the sum of the lateral forces acting on all six tires. $F_{y \text{ total}}$ could be calculated by summing the individual lateral forces on each tire, which were already defined in the simulation.

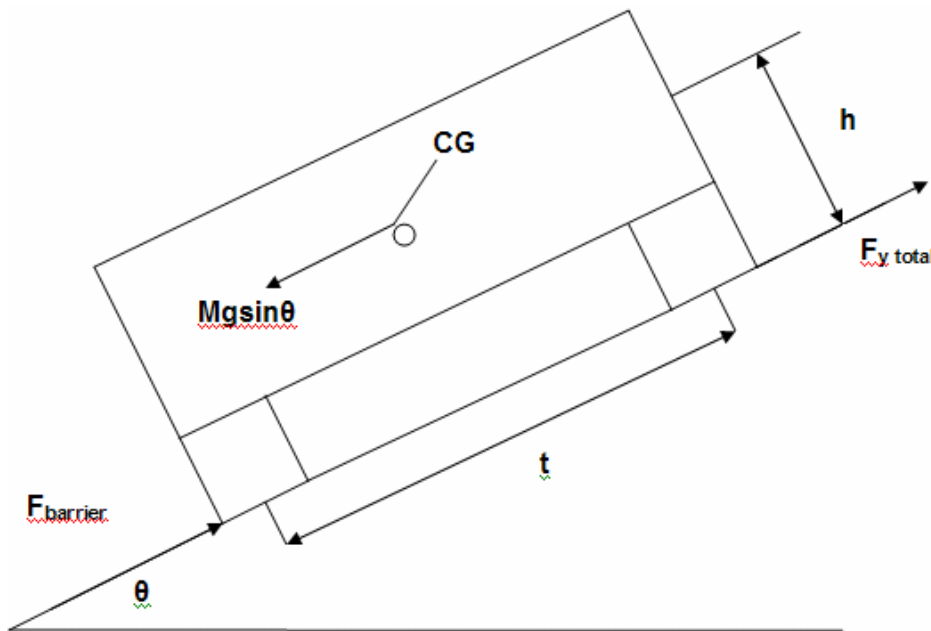


Figure 19. Free body diagram representing barrier force.

The tripping force, F_{barrier} , was divided into two forces, one acting on the front left wheel and the other acting on the rear left outside wheel. They were divided based on their distance from the bus's CG location, to ensure they didn't create a moment that would cause yaw. These tripping forces are then applied on the rims of the previously mentioned wheels, and were placed level with the tilt platform. In this way, the forces on the bus in the direction parallel to the tilt platform would be in equilibrium, but the force couple created by the tilting platform would still allow the bus to tip. The following expression was used to define F_{barrier} :

$$F_{\text{barrier}} = \frac{x}{L} [Mg \sin \theta - F_{y\text{total}}] \quad (10)$$

where, for the rear barrier force, x is equal to the distance from the CG to the front axle and for the front force, x is equal to the distance from the CG to the rear axle and, L represents the wheelbase of the bus.

2.2.2 Tilt Table Simulation

Figure 20 shows the normal forces acting on the tires on the right side of the bus, which happened to be the side higher up the incline of the tilt platform upon lift-off. From the graph, it can be seen that lift-off from the platform (the instant that the normal force, F_z , becomes zero) for the front and rear wheels occurs at different times, which can be attributed to unequal front and rear roll stiffness, as mentioned in **Section 1.7.3**. An interesting point brought up by **Figure 20** is that the rear right outside tire experiences lift-off about fifteen seconds before the rear right inside tire.

Once the outside tire lifts, however, the normal force on the inside tire begins to drop at a faster rate.

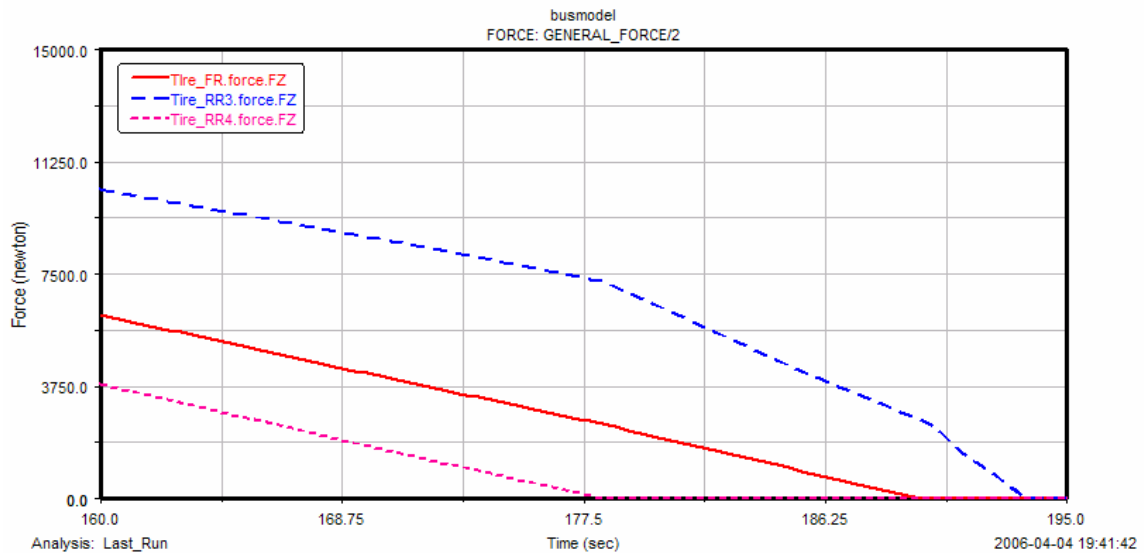


Figure 20. Normal force on right side tires during tilt simulation

The rear right outside tire is the first to lift off the platform, happening 178.2 seconds into the test. With a tilt rate of 0.2 degrees per second, the bus experienced lift-off when the tilt angle was approximately 35.6 degrees. Since the tangent of the tilt angle at lift-off represents the lateral acceleration in g's required to roll a vehicle, according to simulation, this particular bus requires a lateral acceleration of 0.72 g's in order to tip. A simulation snapshot of the bus tipping is shown below in **Figure 21**.

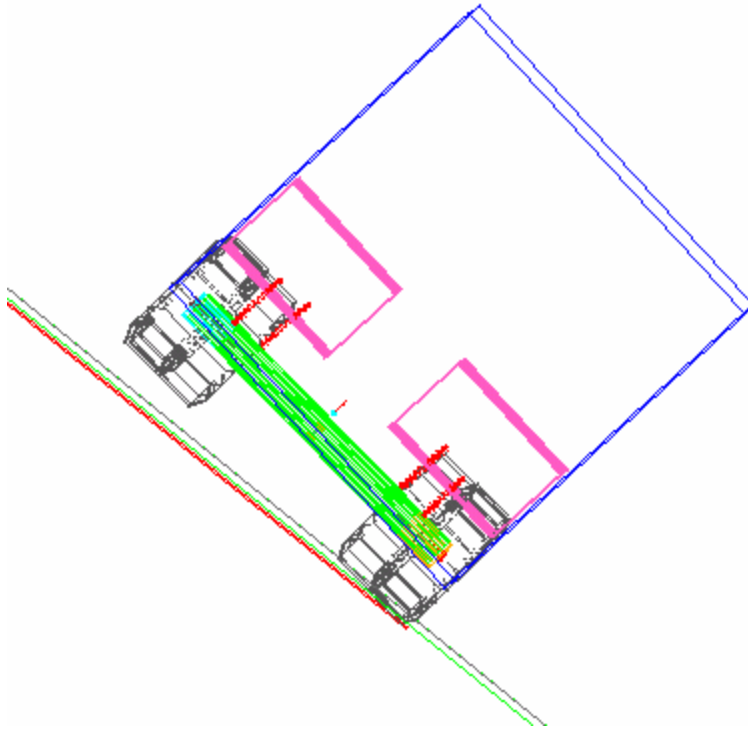


Figure 21. Bus beginning to roll during Tilt Table Test

2.3 Slowly Increasing Steer Maneuver

The Slowly Increasing Steer Maneuver isn't considered a rollover test in itself, but is the basis for many NHTSA rollover tests. The vehicle is brought to constant speed at 50 mph, and the steering wheel is turned at 13.5 degrees per second until its angular displacement reaches 270 degrees. The steering wheel angle at which the vehicle's lateral acceleration is equal to 0.3 g's is recorded and some derivative of it is used for actual NHTSA tests. Because the bus model in the present study is actually steered by controlling the front road wheels directly, not by a steering wheel, a few assumptions must be made. In order to correlate steering wheel angle with actual road wheel angular displacement, a steering ratio of 18:1 was assumed. Because of this assumption, the front wheels in this test were turned 15

degrees over 20 seconds upon reaching 50 mph. **Figure 22** depicts the bus's lateral acceleration during the maneuver.

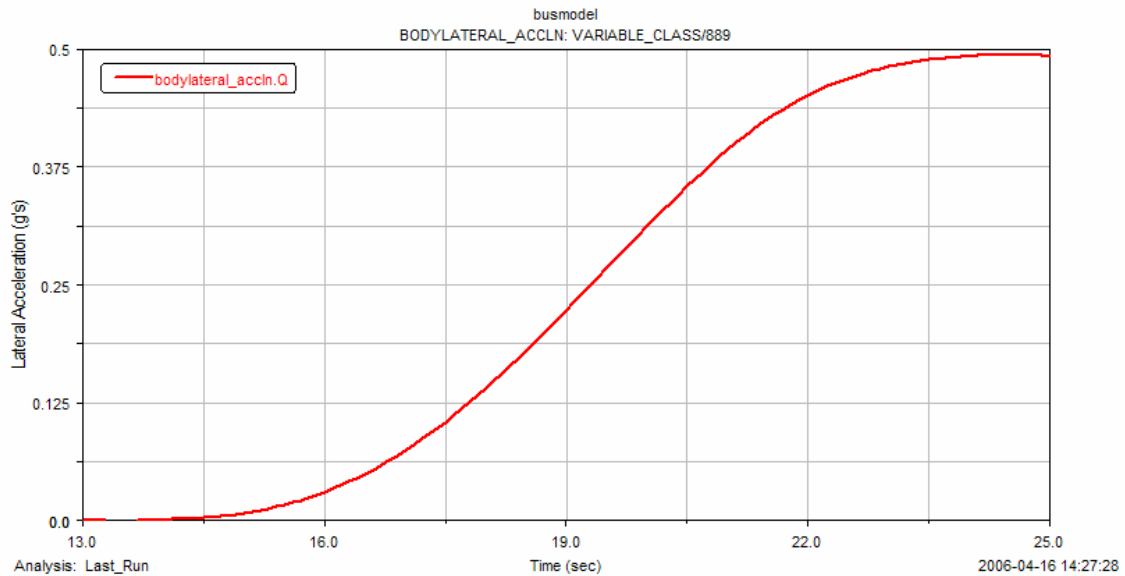


Figure 22. Lateral acceleration of bus during Slowly Increasing Steer maneuver

The bus reaches 0.3 g's at approximately 19.04 seconds, 7.04 seconds after the steering input begins. At 13.5 degrees per second, this is equivalent to a steering wheel angle of 94.97 degrees.

2.4 Fishhook 1a Test

2.4.1 Fishhook 1a Test (35 mph)

The Fishhook 1a test, as described in **Section 1.8.3**, requires a steering input 6.5 times the steering wheel angle found in the Slowly Increasing Steer maneuver. If the necessary steering wheel angle at 0.3 g's during the previous maneuver was 94.97 degrees, assuming an 18:1 steering ratio, the actual front wheels would be rotated 5.28 degrees. Multiplying this value by 6.5, the front wheels must be ramped to the

left 34.30 degrees, and then to the right 68.60 degrees in the opposite direction during the maneuver.

The first run of the fishhook maneuver was performed at a relatively low speed of approximately 35 mph. Shown in **Figure 23** and **Figure 24** are the normal forces acting on the left and right front tires and the left and right rear outside tires respectively. From the graphs it can be seen that the bus initially turns left, due to the spike in the normal force on the right tires, and then after the necessary dwell time, turns right, shown by the sudden increase in normal force on the left tires. Judging by the graphs, it seems that if the bus were to experience lift-off, it would be on the right side during the second steering input. In both the front and rear tires depicted, the normal force drops to about 10,000 N at the lowest point. The bus did not experience any wheel lift-off during this test, which can be seen by no normal force on any tire reaching zero. It should also be noted that for each set of opposing wheels, the graph of their normal forces are mirror images, since the weight on both tires must add up to the weight on the tires at equilibrium.

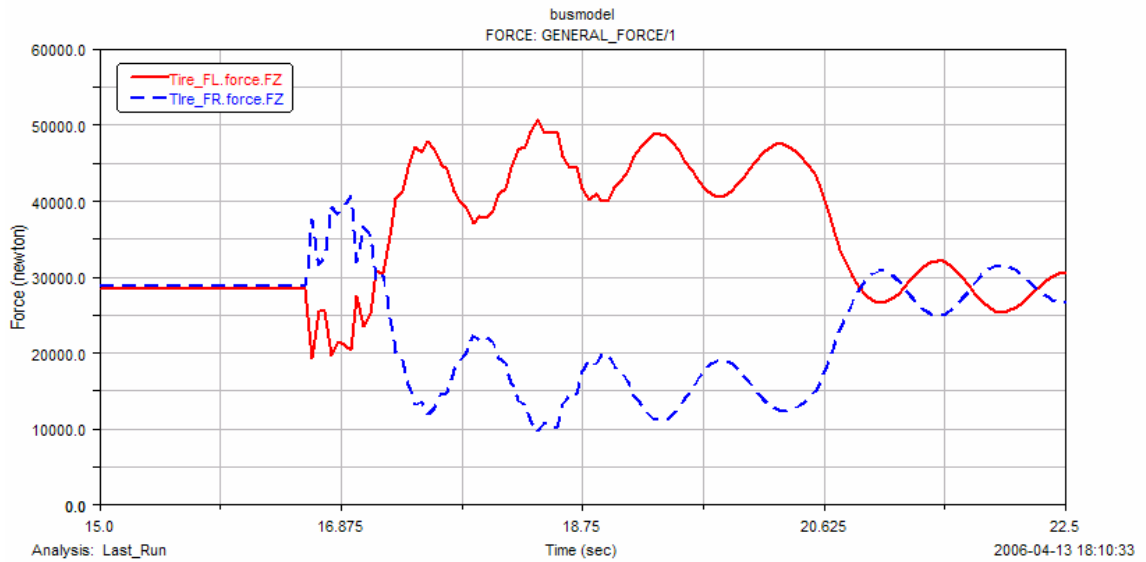


Figure 23. Normal forces on left and right front tires during the 35 mph Fishhook 1a test

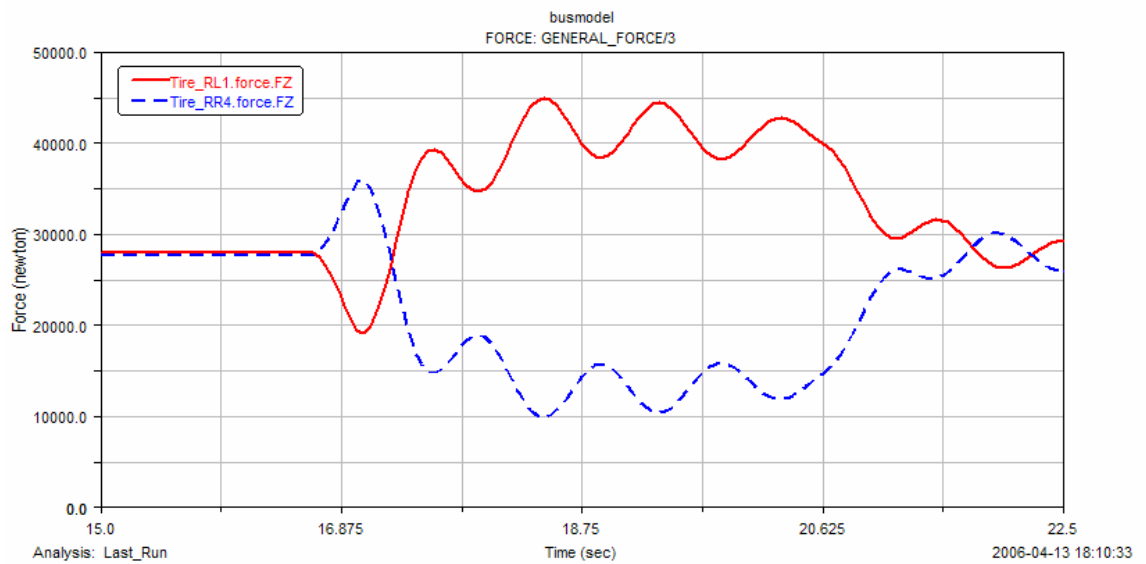


Figure 24. Normal forces on left and right rear outside tires during the 35 mph Fishhook 1a test

Another phenomenon seen is the tendency of the bus to oscillate during cornering. This is displayed not only in the graphs above, but could be seen during simulation. Upon completion of the maneuver, the normal force still oscillates between the right and left side, eventually returning to equilibrium. This effect can

also be seen in **Figure 25**, in which roll angle, the angular displacement between the body's vertical axis during cornering and its vertical axis at rest, is depicted against time. While the bus reaches a maximum roll angle of about 3.0 degrees, it oscillates during the maneuver about 1.5 to 2.0 degrees during the most severe cornering.

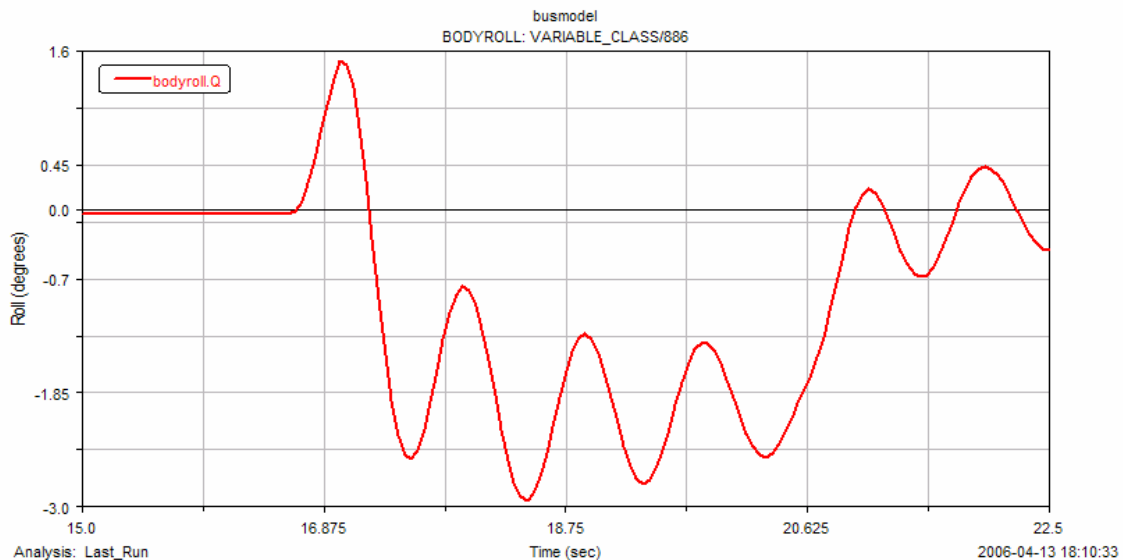


Figure 25. Roll angle of the bus during the 35 mph Fishhook 1a test

The oscillation described above is an effect of a lightly damped suspension with low roll stiffness. While this type of suspension may create oscillatory responses during rollover testing, in which severe cornering maneuvers are performed, it is a more ideal suspension for city driving. When designing transit buses, passenger comfort is a high priority, and cornering such as that described here isn't often experienced.

Shown below in **Figure 26** is the bus's yaw rate, the rate of the bus's angular displacement about its vertical axis, plotted against time. An interesting correlation can be made when comparing yaw rate to lateral acceleration, shown in **Figure 27**. While not identical, both curves are very similar in shape. The bus seems to

experience the greatest lateral accelerations during times when it is rotating about its vertical axis most rapidly. Lateral acceleration spikes during the first steering input, and reaches a maximum of about 0.45 g's during the second steering input. This value is much smaller than the simulated lateral acceleration experienced upon tipping by the bus during the Tilt Table test.

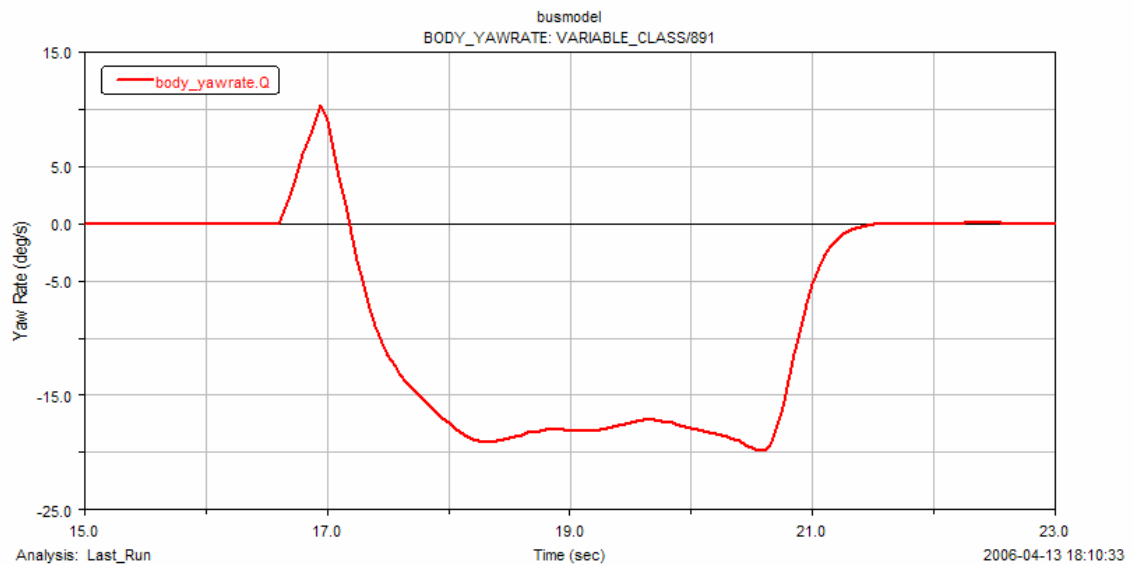


Figure 26. Yaw rate of bus during 35 mph Fishhook 1a test

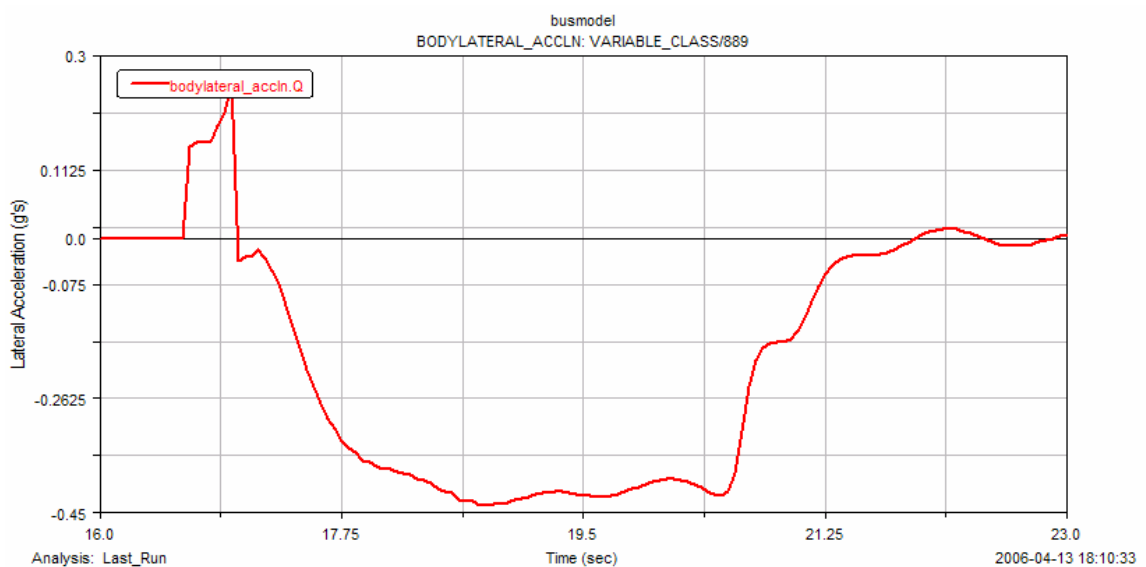


Figure 27. Lateral acceleration of bus during 35 mph Fishhook 1a test

2.4.2 Fishhook 1a Test (45 mph)

The normal force curves for the left and right front tires and the left and right rear outside tires look very similar to the 35 mph test, as seen in **Figure A1** and **Figure A2**. There isn't much difference between the normal force graphs for the front tires from the 35 mph test, except for the minimum normal force on the right tire dropping more noticeably below 10,000 N. In contrast, the rear right outside tire seems to have significantly less weight on it during the second steering input. In this case, there seems to be at least a 1,000 N decrease on the tire from last test, indicating more weight transfer. This concept at first seems strange, considering the front experienced almost no change in weight transfer. One possible cause to this effect is that at during simulation at higher speeds, the bus seemed to lose some stability in the rear, causing a fishtail effect. In addition, there is a greater proportion of the bus's weight acting on the rear axle. Together, these effects may have caused a slight increase in lift on the back end of the bus.

The oscillatory effects of the bus suspension are still visible, which can be seen in the roll angle plot, **Figure A3**. The plot is almost identical to the 35 mph plot, with only slight variation. In this case the roll angle appears to reach a maximum again at about 3.0 degrees. The oscillations, however, are becoming slightly smaller in magnitude, as the bus never oscillates past 2.0 degrees as in the 35 mph test. This may be attributed to the last test being more in tune with one of the modes of the suspension.

Although the plot of yaw rate below in **Figure 28** is very similar to that of **Figure 26**, there are some slight differences. During the first and second steering

input, the magnitude of the yaw rate in the 45 mph test is less than that of the 35 mph test. The graph below shows the yaw rate of the bus never going above 10.0 degrees per second and during the first turn and never going above 17.5 degrees per second during the second turn, two values that were exceeded during the last test. This seems counterintuitive, since a car moving around a curve at a higher speed should have a greater yaw rate. This is also most likely due to skidding. During skidding, the tires are not perfectly rolling, and thus do not have maximum directional stability, causing the bus to deviate from the intended path. During the second steering input, the yaw rate decreases from about 17.5 degrees per second in magnitude to about 10 degrees per second in magnitude. This is most likely caused by the bus skidding. Afterwards, upon regaining control, the yaw rate again increases before the maneuver ends. At this point, it should be noted that the bus's lack of directional control at high speed is becoming a problem in itself.

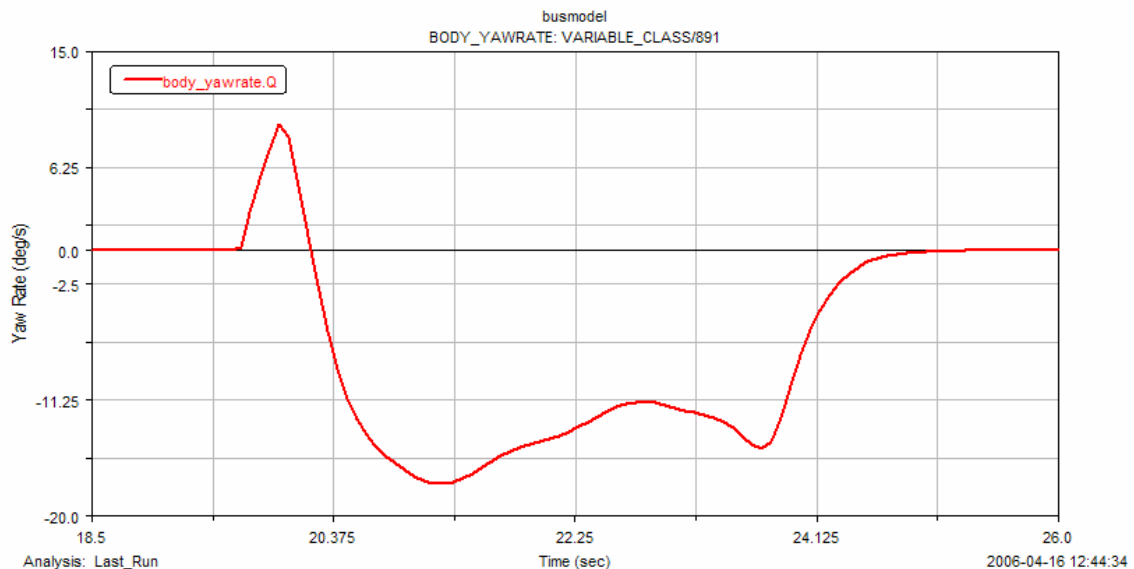


Figure 28. Yaw rate of bus during 45 mph Fishhook 1a test

The longitudinal acceleration of the bus also stays relatively constant from the last test, and as shown in **Figure A4**, it still correlates well with yaw rate. One difference is that while the yaw rate changed in magnitude due to sliding during the 45 mph test, the skidding of the bus did not effect the bus's lateral acceleration. As yaw rate stays constant, lateral acceleration should also stay constant since the tires keep directional stability. At a certain point, if lateral acceleration increases due to a decreased turning radius, the vehicle may begin to slide due to the overcoming of tire lateral forces. At this point, the yaw rate will decrease, as will lateral acceleration slightly. When the tires eventually stop skidding and regain stability, the yaw rate will again increase if the wheels are still turned, and the lateral acceleration will again increase until the process is repeated.

2.4.3 Fishhook 1a Test (55 mph)

As the Fishhook 1a test is done at higher and higher speeds, it is evident that there is definitely an increase in weight transfer, especially between the right and left rear wheels. As shown in **Figure A5**, the normal force on the right rear outside tire drops well below 10,000 N, to approximately 8,300 N. Simultaneously, what seems to be more and more evident, is that the bus is much more likely to slide under high lateral accelerations than it is to roll, as can be seen in the yaw rate plot in **Figure 29**. The faster the bus travels during the fishhook maneuver, the more inclined it is to slide instead of corner. **Figure 29** shows an approximately 10.0 degree per second drop in yaw rate during the second steering input. This drop in yaw rate is directly associated with the bus sliding instead of cornering. This is a prime example of the

frictional coefficient and thus the frictional force between the tires and the ground being too small for the bus to roll under realistic driving conditions. Because the bus is sliding, the weight transfer to the outside wheels is not able to increase dramatically enough for the bus to roll.

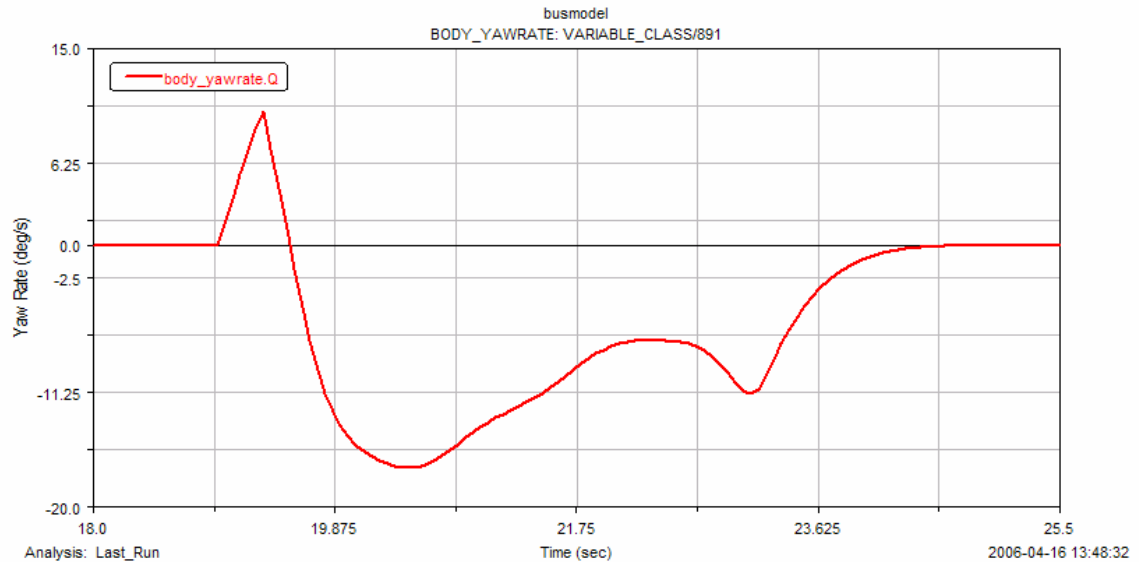


Figure 29. Yaw rate of bus during 55 mph Fishhook 1a test

Most vehicle rollover studies are based on the maximum lateral acceleration a vehicle can withstand before it begins to tip. As seen in **Figure A6**, although the bus is traveling faster and faster through the maneuver, its lateral acceleration continues to remain relatively constant, not growing much larger than 0.45 g's, again, directly related to the sliding of the bus. If the frictional forces between the tires and ground were stronger, the bus would be able to corner more tightly, and lateral acceleration would then have a chance to increase to the point of rollover. Further supporting this point is bus's roll angle, shown in **Figure A7**. The magnitude of the largest roll angle experienced is slightly larger than before, accounting for the increase in weight shift.

Although there is a slight change in roll angle, it is still about 3.0 degrees at maximum.

2.4.4 Fishhook 1a Test (65 mph)

Even at 65 mph there is not much change to the previously gathered data. The normal force curves for the front and rear tires do not change significantly, as shown in **Figure A8** and **Figure A9**.

The effect of an aggressive steering input at high speeds can once again be seen in **Figure 30**, a yaw rate plot of the bus during the maneuver at 65 mph. There is an even greater decrease in yaw rate, this time about 14.0 degrees per second. This large decrease is directly related to lateral acceleration, a value that's largest magnitude has yet to change significantly, even during cornering at 65 mph, as shown in **Figure A10**.

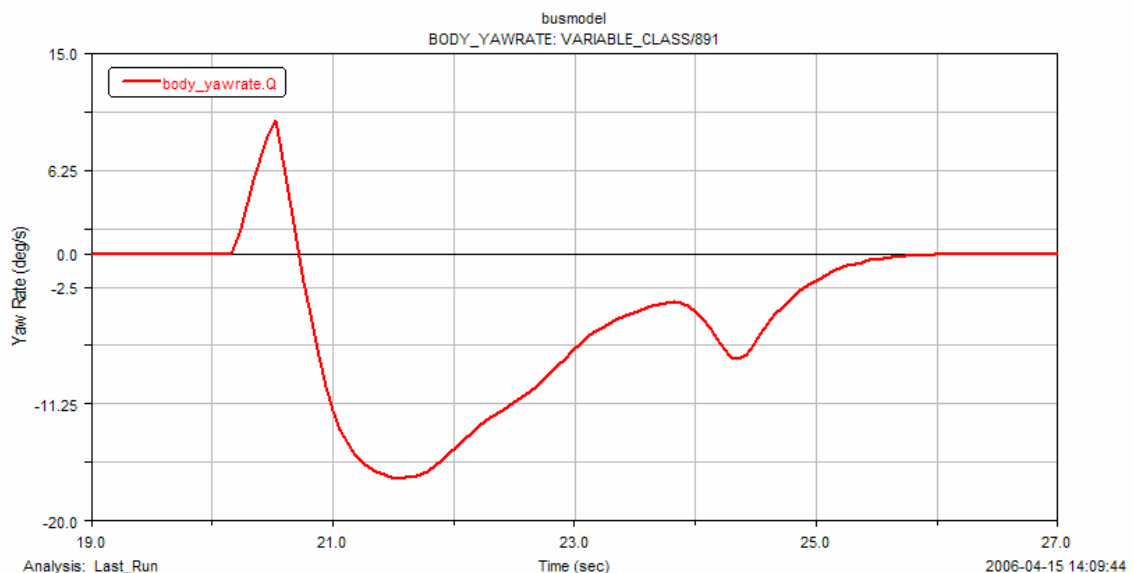


Figure 30. Yaw rate of bus during 65 mph Fishhook 1a test

Figure 31 shows a snapshot of the Fishhook 1a test simulated at 65 mph. Even at 65 mph, the bus is not close to rolling over. It is easy to see the weight transfer as the bus corners, but tipping cannot be seen from this angle.

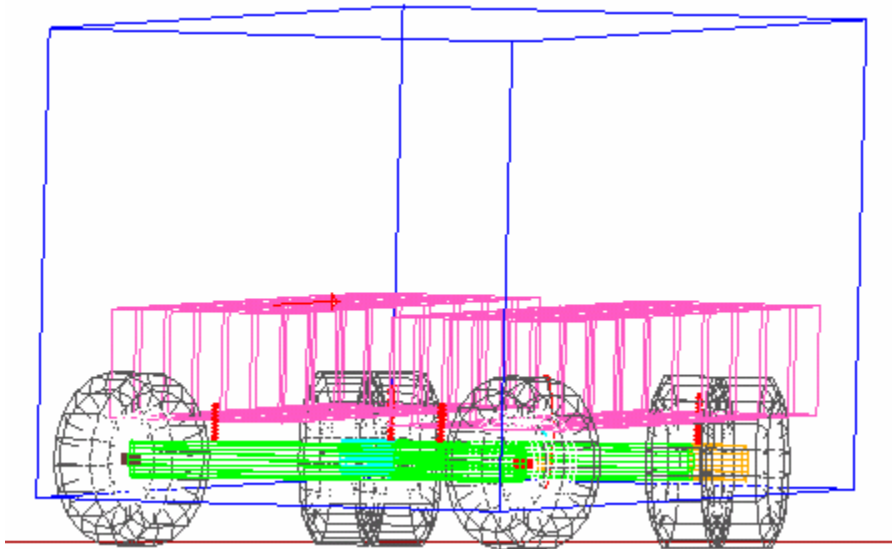


Figure 31. Side view of Fishhook 1a simulation at 65 mph

2.5 ISO 3888 Double Lane Change Maneuver (65 mph)

The ISO 3888 Double Lane Change maneuver is different than the NHTSA tests in that it is geometry-based instead of time-based. A simulation snapshot of the bus driving through the lane change course is shown in **Figure 32**. Considering the bus seemingly will not roll under realistic driving conditions, this test was only simulated at one speed, 65 mph. This test is much more difficult to simulate than the previous one. Because the fishhook maneuver is based only on timing, the steering input simply needs to be programmed for the correct times. Since the ISO maneuver is based on a suggested distance for a lane change, the user must determine the

correct times for the steering inputs through trial and error. Needless to say, correcting the steering input to counter the skidding of the bus was also an issue. In this test, the steering inputs (8.5 degrees left, 25.5 degrees right, 42 degrees left) were not as drastic as in the NHTSA tests, but the results were somewhat surprising.

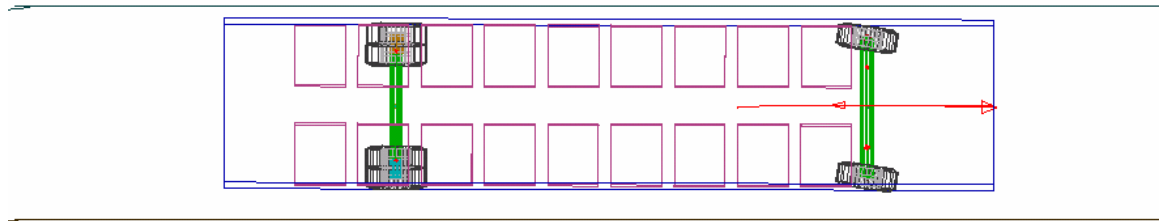


Figure 32. Bus traveling through ISO 3888 Double Lane Change course

Figure 33 shows the normal forces on the left and right front tires during the maneuver. At first glance it is obvious that the weight shift is much more drastic in this test than in the NHTSA tests. After the bus enters the first turn, and turns the wheel back to the right, the normal force on the front right tire drops below 5,000 N, a figure much lower than that experienced during an NHTSA test. Afterward, upon turning the opposite direction while exiting the course, the normal force on the front left tire actually drops to zero, and the bus experiences wheel lift-off, as can be seen in a front view of the simulation in **Figure 34**. Unlike the previous NHTSA test, the weight shift on the front wheels is greater than the weight shift on the back wheels, as seen in **Figure 35**.

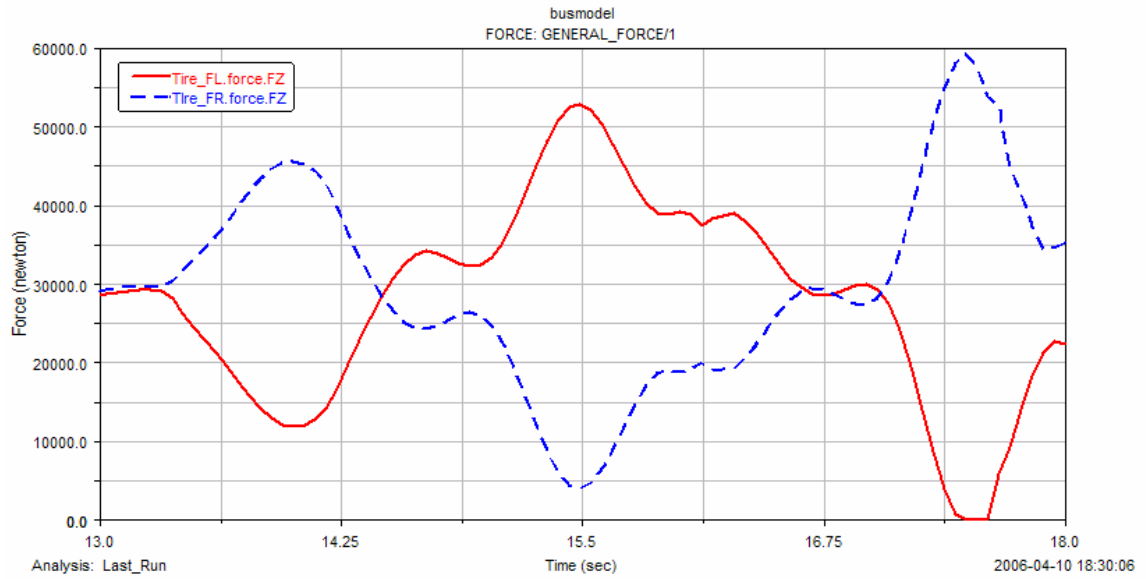


Figure 33. Normal forces on left and right front tires during the 65 mph ISO 3888 Double Lane Change test

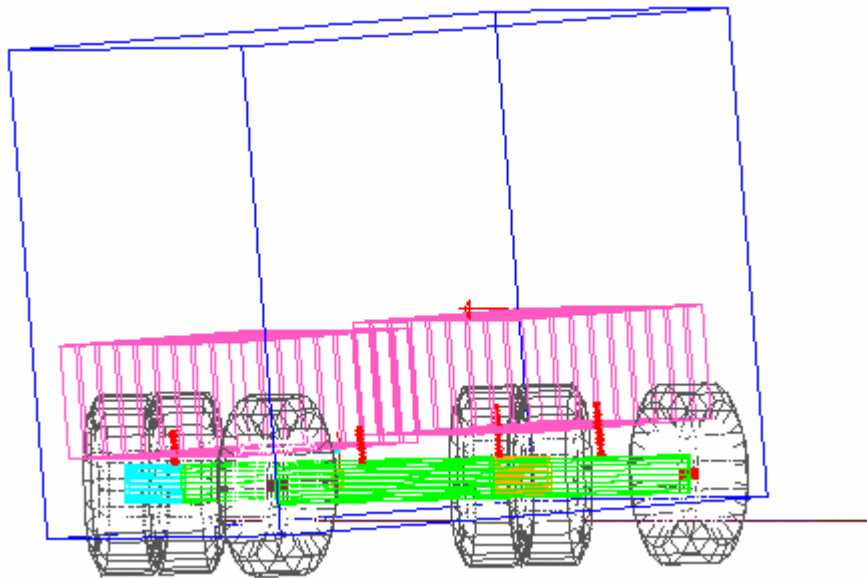


Figure 34. Front view of bus experiencing wheel lift-off during ISO 3888 Double Lane Change test

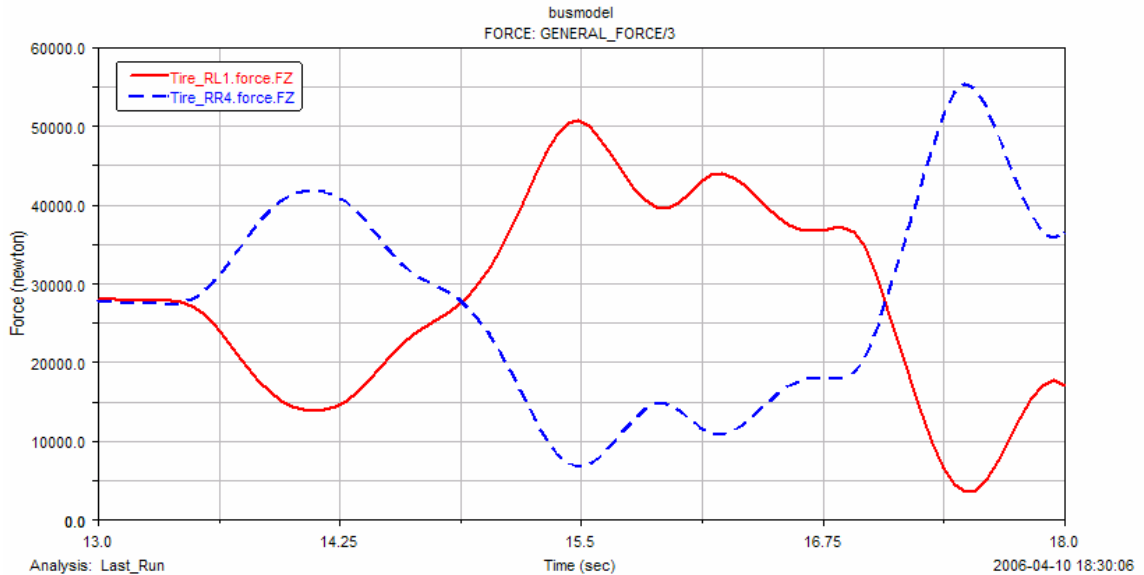


Figure 35. Normal forces on left and right rear outside tires during the 65 mph ISO 3888 Double Lane Change test

This increase in weight transfer can be attributed to greater traction between the tires and the road. Traction was a major concern in the NHTSA fishhook maneuver, and thus the vehicle never came close to rollover, even at high speeds. Logically, it seems as though a greater steering input would lead to a greater lateral acceleration and therefore more lateral weight transfer. The greater steering input, however, may have been directly related to the lack of traction and sent the bus into a skid. By decreasing the steering input to the point that there is still enough tire-road friction to keep from completely sliding, the bus was able to transfer weight from side to side enough for a wheel to lift. Without the excess sliding, there was still enough lateral acceleration to create this effect even though the steering input wasn't as great as in the NHTSA test.

Figure 36 depicts the yaw rate for the double lane change maneuver. The curve is much smoother than those shown for the fishhook tests, and shows no sudden

decrease in yaw rate except for a minor skid during the second turn, around 15.3 seconds. Other than this slight skid, the only changes in yaw rate shown are those that are associated with the vehicle changing direction based on course geometry. The yaw rate plot below shows a much more stable vehicle than in the previous NHTSA tests.

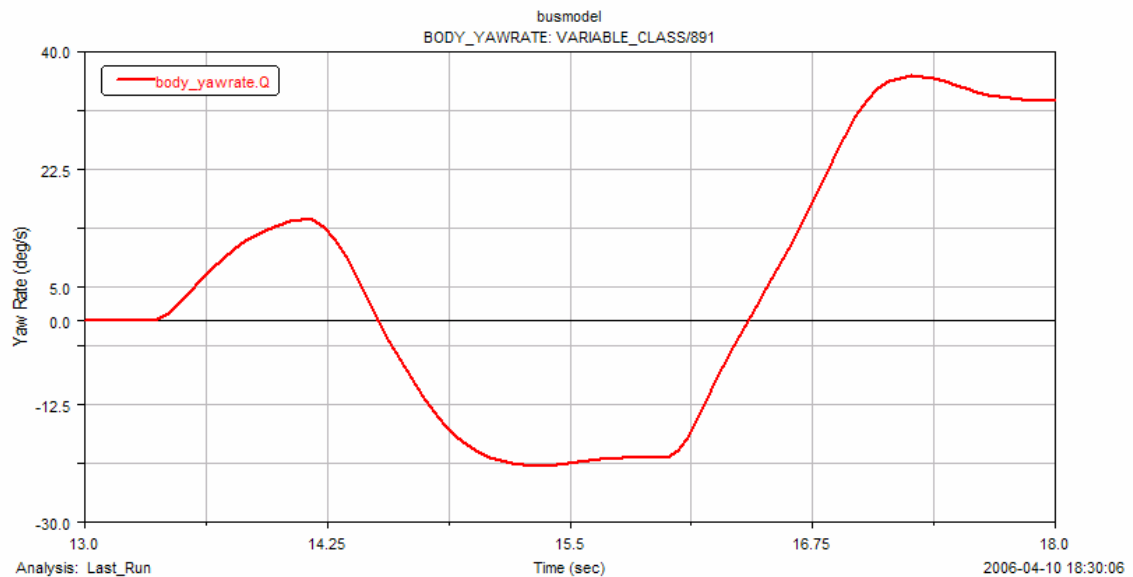


Figure 36. Yaw rate of bus during the 65 mph ISO 3888 Double Lane Change test

Like the 35 mph Fishhook 1a test, the test in which the bus was most stable out of the other fishhook tests, the lateral acceleration plot is very similar to the yaw rate plot above. Interestingly, the lateral acceleration, as seen in **Figure 37** below does not grow much beyond 0.45 g's at its greatest magnitude, as was seen in previous NHTSA tests. The same lateral acceleration accompanies lift-off in this scenario. The answer may again be correlated to the lack of slip experienced in this test. The greater traction allowed the same lateral force to create a moment great enough to lift the front-left wheel. Again, associated with lateral weight transfer is

roll angle. Since the bus in this case did experience lift-off, it would be assumed that the roll angle of the bus is actually greater than the 3.0 degrees seen in the NHTSA tests. The roll angle, shown in **Figure 38**, does in fact reflect this, as it reaches a maximum of about 4.6 degrees in magnitude.

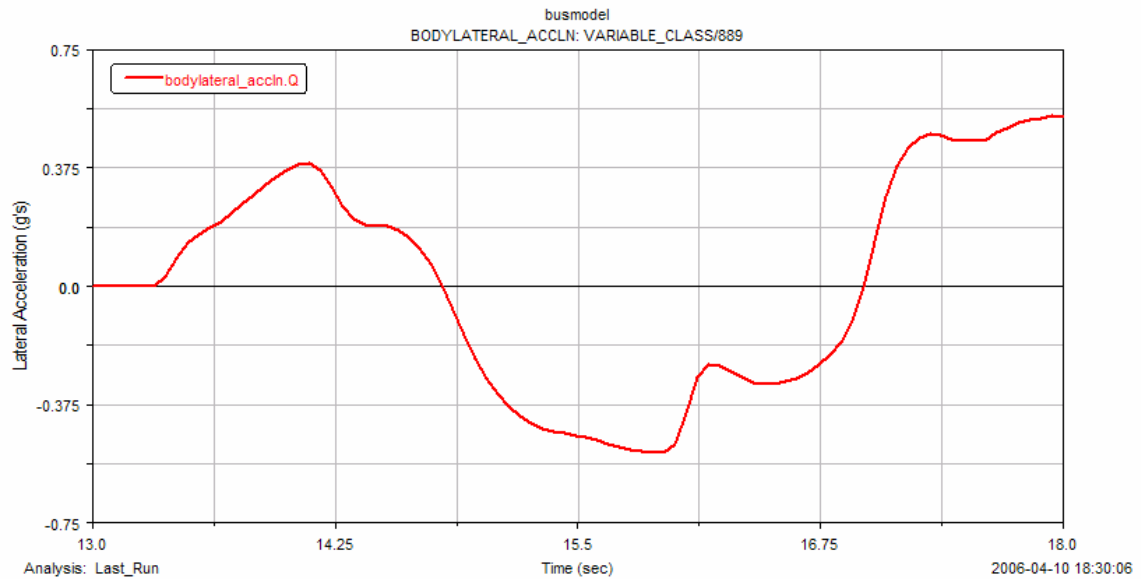


Figure 37. Lateral acceleration of bus during 65 mph ISO 3888 Double Lane Change test

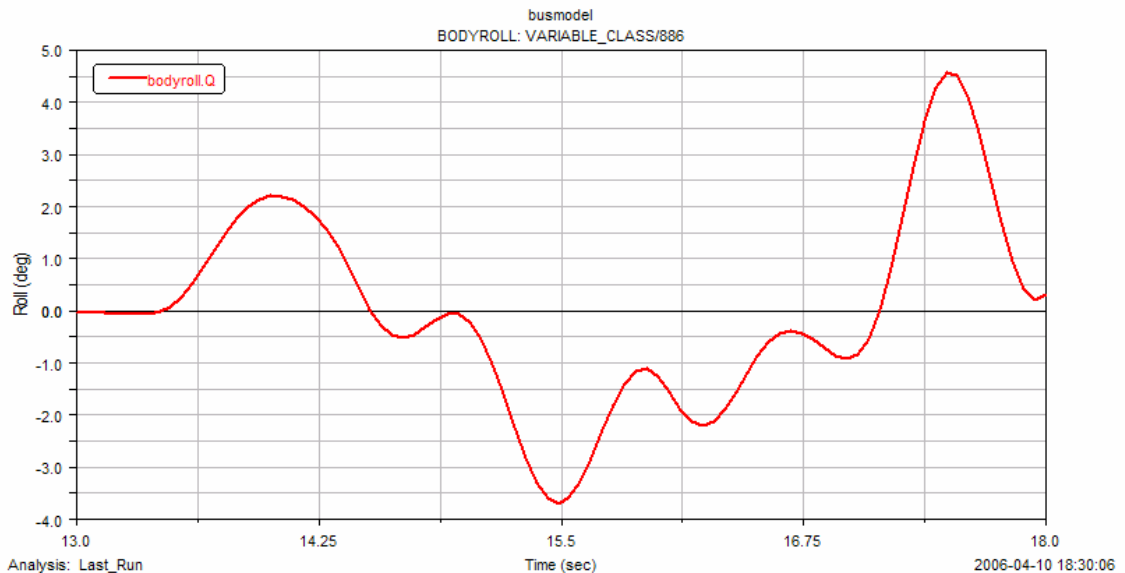


Figure 38. Roll angle of the bus during the 65 mph ISO 3888 Double Lane Change test

CHAPTER 3: CONCLUSION

Determining the rollover threshold of a vehicle is much more complicated than simply dividing a vehicle's track width by twice its CG height. Detailed vehicle dynamics models offer a better opportunity for exploring the physics involved in rollover. One such model was used in this work to investigate the potential of standard static and dynamic rollover tests for a transit bus. Results from all the rollover tests are summarized in **Figure 39**.

	<u>Lateral Acceleration to Cause Rollover (g's)</u>
Static Stability Factor	1.0
Tilt Table Test	0.72
NHTSA Fishhook 1a Test	No Wheel Lift-Off
ISO 3888 Lane Change Test	0.48 (one wheel lift-off)

Figure 39. Summary of rollover tests

The Static Stability Factor, or Static Rollover Threshold, of this bus is approximately 1.0, meaning that under the assumption that the bus is a rigid body, it would tip when its lateral acceleration were equal to 1.0 times the acceleration of gravity. In a simulated Tilt Table test, it was found that only 0.72 g's of lateral acceleration would cause the bus to tip. The difference between these two tests is that during the Tilt Table test, the bus was no longer assumed to be a rigid body, and its suspension and tire effects were included. The accuracy of this test may be somewhat in question, however, since the bus tipped at a rather large angle. In practice, it is recommended that the angular displacement of the tilt platform be kept as close to zero as possible, to maintain maximum accuracy.

The Tilt Table test and Static Stability Factor of the bus reveal two very different values for the lateral acceleration that would cause the bus to roll. Although these tests are only simulated, the situation mirrors real life in terms of the need to dynamically test the bus. With such varying data, the only way to get a better idea as to what lateral acceleration would actually tip the bus would be to put it through an actual driven maneuver. After performing the NHTSA Fishhook 1a test, it also became evident that it is necessary to tailor the test to the vehicle in question. At reasonable speeds, the bus did not show signs of rollover. As the speed was increased to speeds at which the maneuver in question became unrealistic, the bus still did not roll because the test in question created a situation in which the bus was much more prone to sliding than rolling.

The ISO 3888 Double Lane Change test gave most likely the most realistic reading. The steering input was much more reasonable in terms of real world scenarios than the NHTSA test, and the test actually produced viable rollover data. Although only one wheel lifted during the test, it is probably safe to say that given a little more lateral acceleration at that point, the rear wheels would have lifted also, given the nature of the normal force graph (**Figure 45**). Although the ISO test does not give an exact answer, it shows that the actual rollover threshold is most likely between 0.48 g's and 0.72 g's, as determined by the Tilt Table test.

Furthermore, these results show the need to create a rollover test specifically tuned to the needs of a bus. Tests such as those created by NHTSA and ISO appeal to the needs of passenger cars and trucks, where there is more of a public demand. Passenger vehicles and buses, however, are driven under different situations and

conditions. As the varying results above show, these tests do not give decisive results for a low-floor transit bus. Research into what types of maneuvers and situations have caused rollover in buses in the past would allow for the development of tests that are more suited to the needs of a bus.

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Appendix A

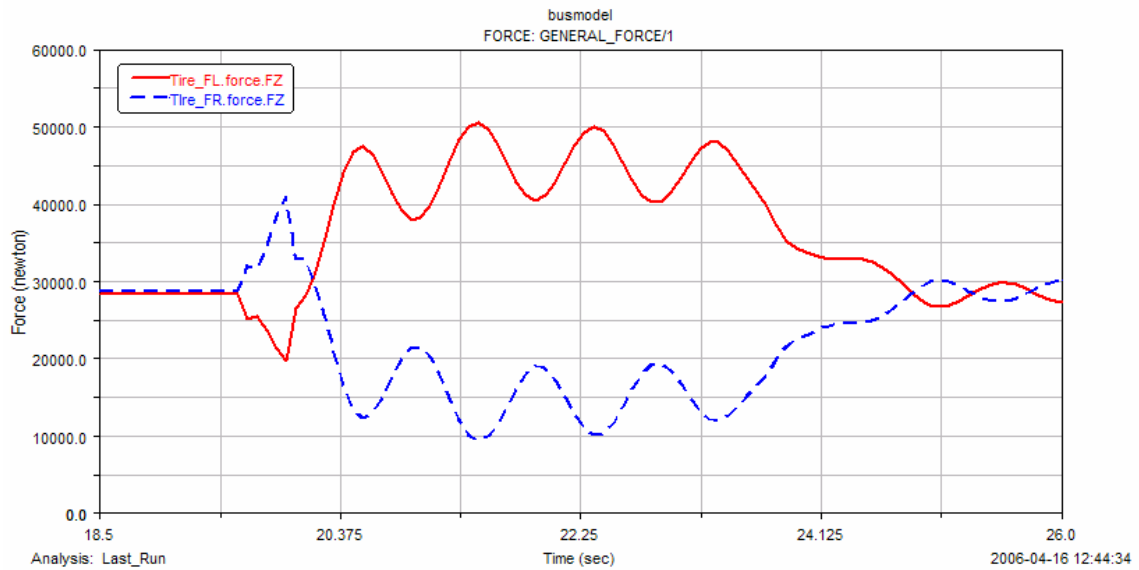


Figure A1. Normal forces on left and right front tires during the 45 mph Fishhook 1a test

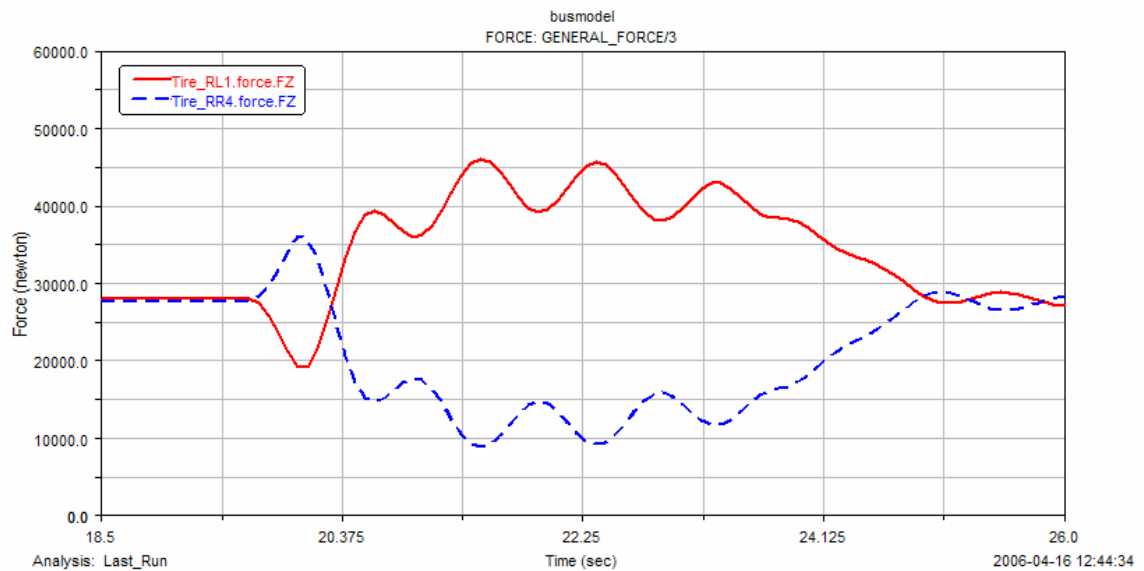


Figure A2. Normal forces on left and right rear outside tires during the 45 mph Fishhook 1a test

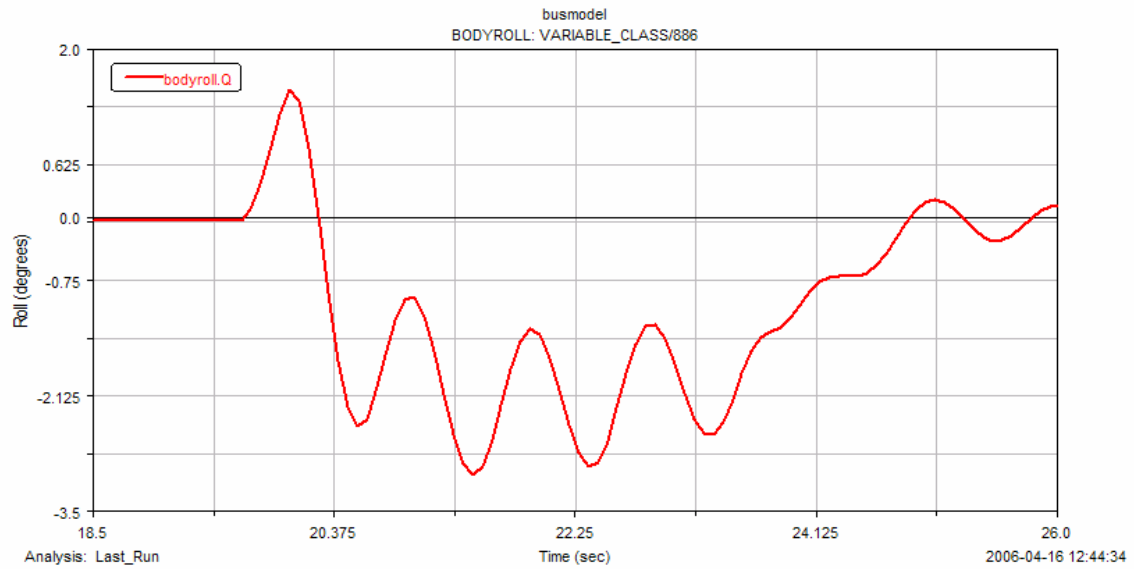


Figure A3. Roll angle of the bus during the 45 mph Fishhook 1a test

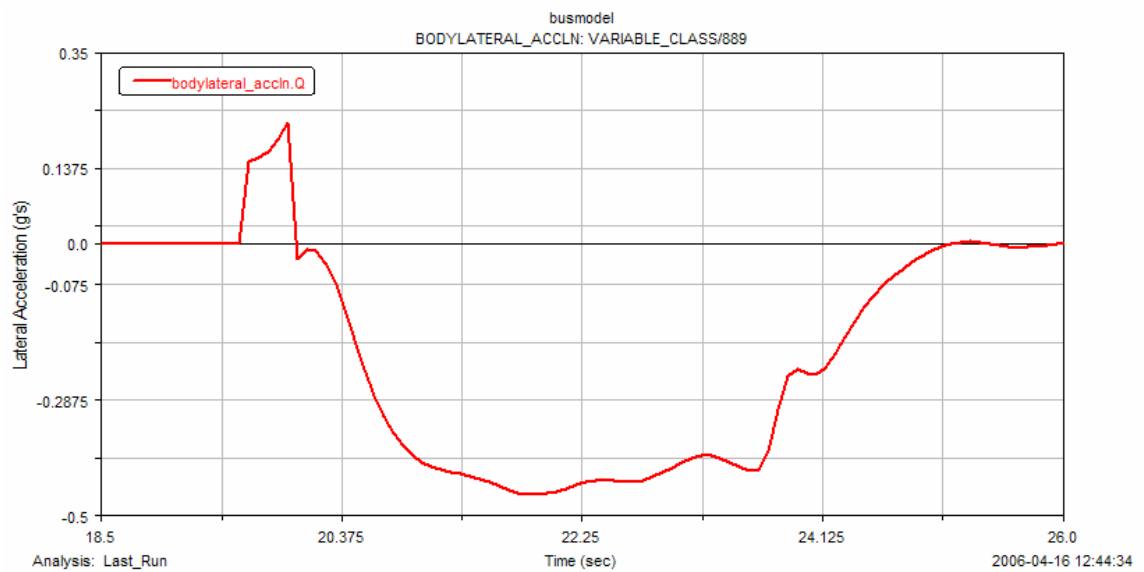


Figure A4. Lateral acceleration of bus during 45 mph Fishhook 1a test

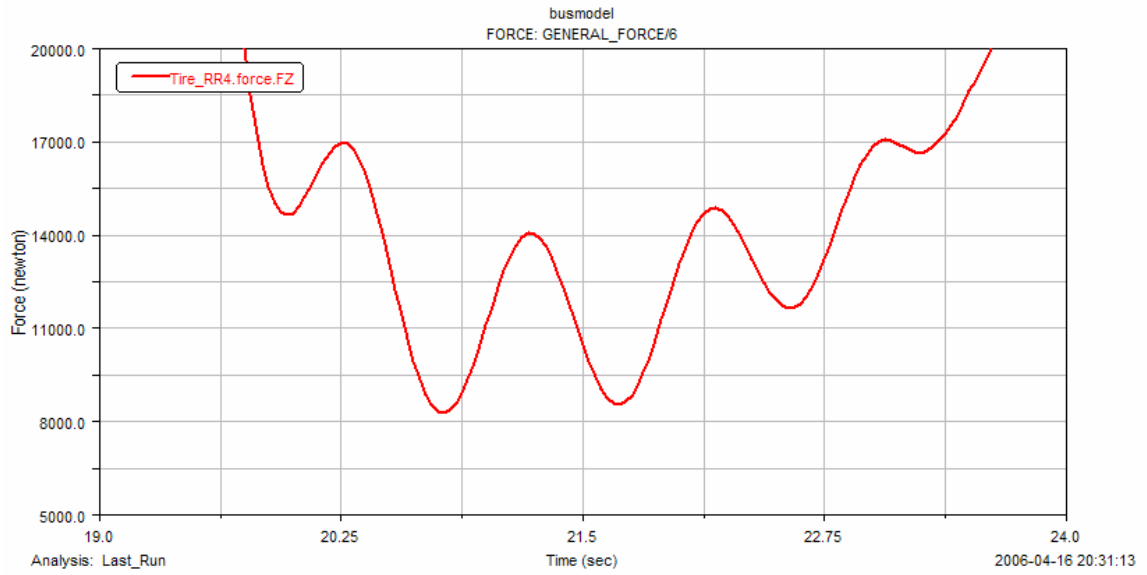


Figure A5. Normal forces on right rear outside tires during the 55 mph Fishhook 1a test

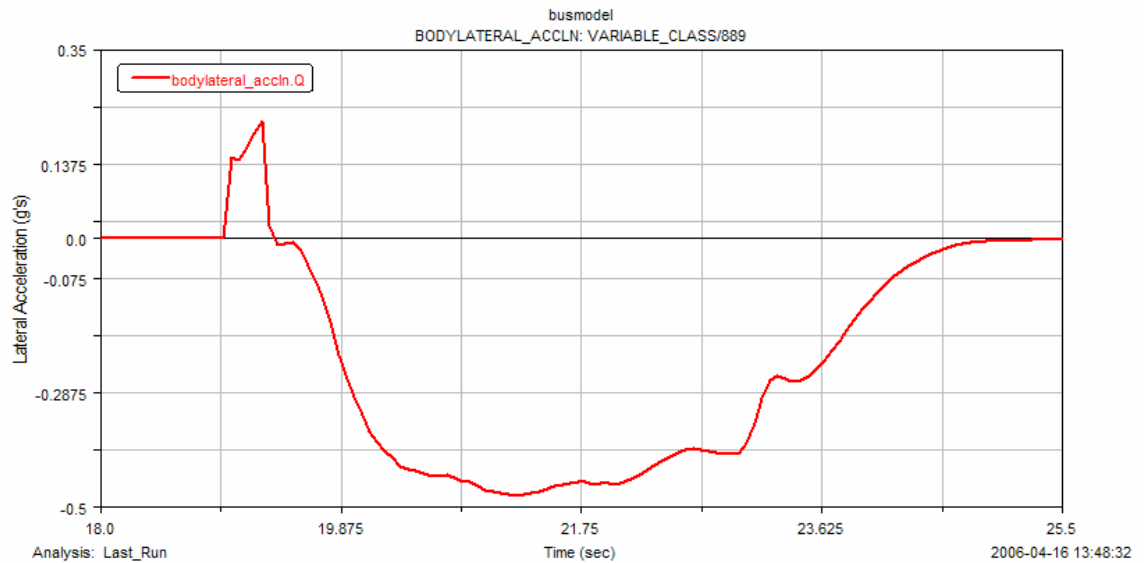


Figure A6. Lateral acceleration of bus during 55 mph Fishhook 1a test

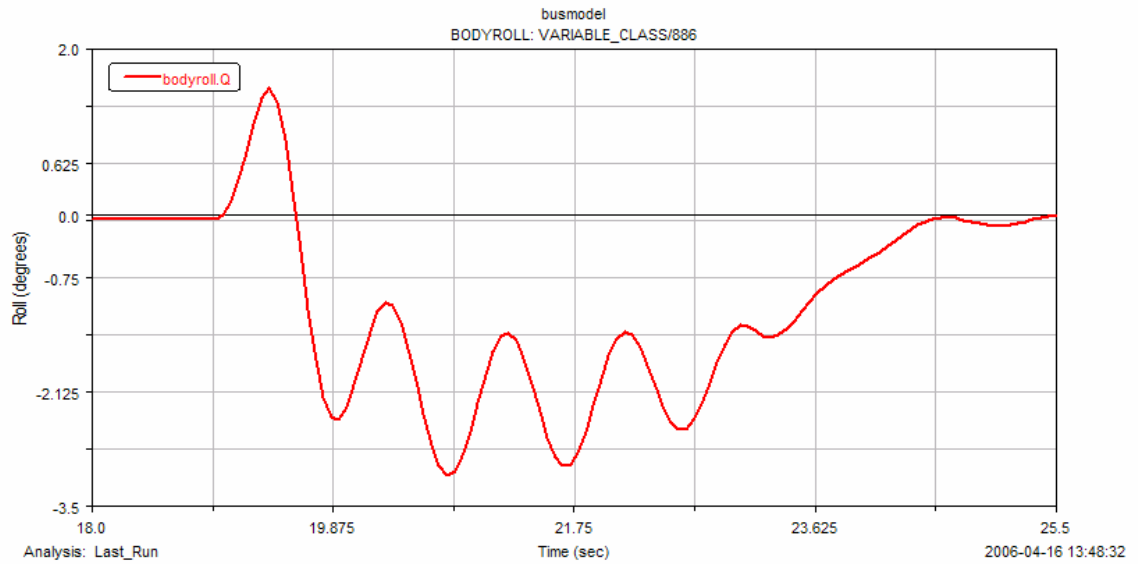


Figure A7. Roll angle of the bus during the 55 mph Fishhook 1a test

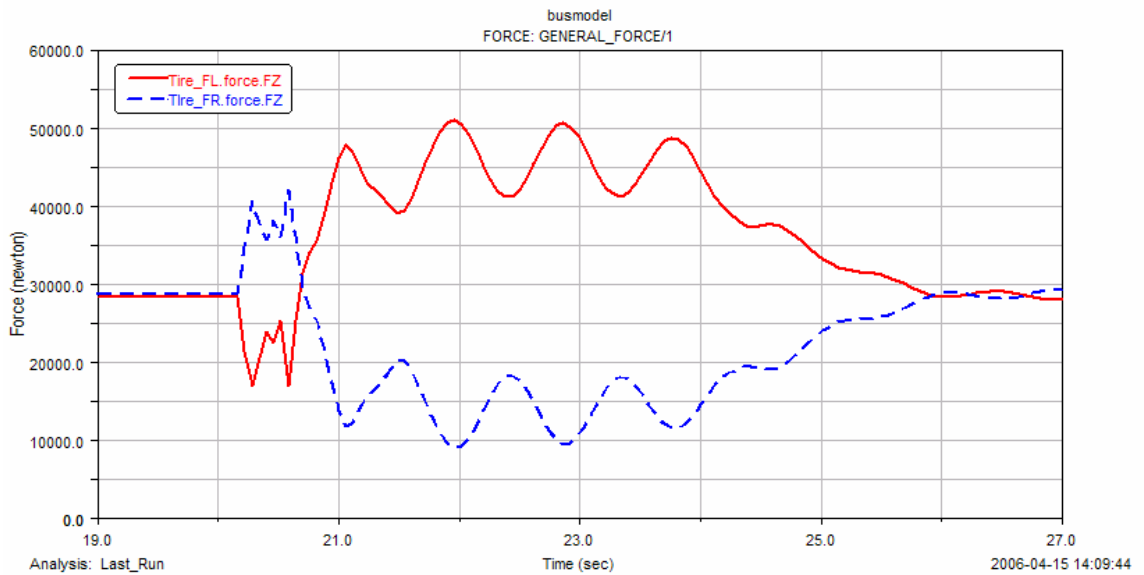


Figure A8. Normal forces on left and right front tires during the 65 mph Fishhook 1a test

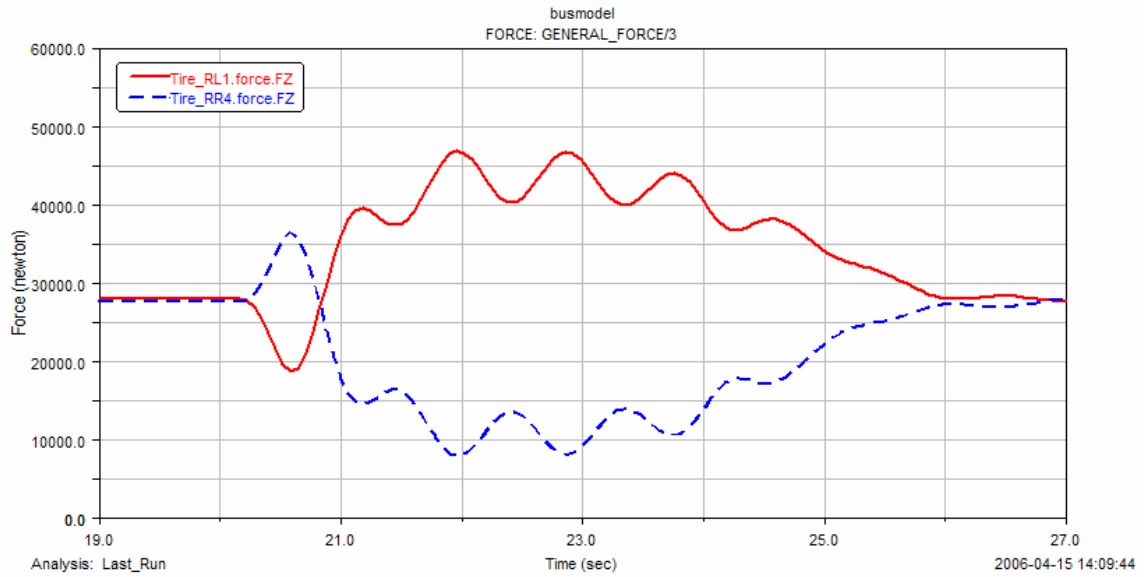


Figure A9. Normal forces on the left and right rear outside tires during the 65 mph Fishhook 1a test

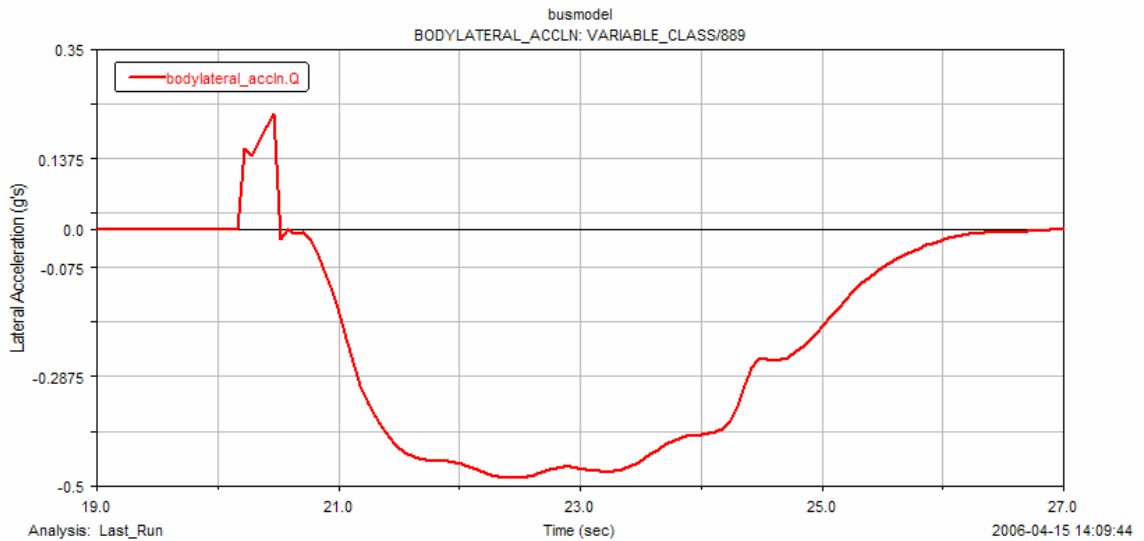


Figure A10. Lateral acceleration of bus during 65 mph Fishhook 1a test

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