NTRCI

NATIONAL TRANSPORTATION RESEARCH CENTER, INCORPORATED
UNIVERSITY TRANPORTATION CENTER

U02: Heavy Truck Rollover Characterization (Phase-A) Final Report

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EXECUTIVE SUMMARY

BACKGROUND

This Heavy Truck Rollover Characterization Program is a major research effort conducted by the National Transportation Research Center, Inc. (NTRCI) in partnership with Oak Ridge National Laboratory (ORNL), Michelin Americas Research Company (MARC), Western Michigan University (WMU), and Battelle Memorial Institute (Battelle). This research is one of the major projects conducted by the NTRCI in its role as a University Transportation Center (UTC) for the Research and Innovative Technology Administration (RITA), an agency within the Department of Transportation (DOT). This research also involved, via subcontract, Volvo Trucks North America, Inc., Clemson University and the Transportation Research Center, Inc. (TRC). This work, entitled Heavy Truck Rollover Characterization – Phase-A, addresses truck rollover issues dealing with a tractor and flatbed-trailer, and follows similar work that was conducted by NTRCI on a tractor-van-trailer in 2004-2005 [1].

BRIEF OVERVIEW

The overall objectives of this research were to:

- Contribute to the understanding of the dynamics of heavy truck rollover,
- Contribute to the development of advanced models of heavy truck vehicle dynamics that reflect project experiences, and
- Develop recommendations for improvement of the roll stability of heavy vehicles in preparation for realizing and testing such concepts in future phases of heavy truck rollover characterization research.

This project involved four major types of activities;

1) Tractor and flatbed-trailer characterization,
2) Simulation modeling,
3) On-track testing, and
4) Data analyses.

The tractor and flatbed-trailer characterization were necessary in order to generate kinematic and compliance data for later use in the development of simulation models of the vehicle.

Simulation modeling was conducted in order to better understand the dynamic behavior of the test vehicle, and if successful, would allow for the conduct of various "what if" design studies. Initial solid-body models, finite element models, and kinematic models were also developed and will prove to be important as this project moves into the design recommendation phase.

On-track testing was conducted to:

- Better understand the behavior of the test vehicle in a quasi-real-world environment,
- Allow the research team to study and experience the performance of the test vehicle while varying several control parameters, and
- Generate data for baseline comparison purposes with the simulation models.

Data analyses were conducted to assess the potential benefits of New Generation Single Wide-Based Tires (NGSWBTs), Electronic Stability Control (ESC) on the tractor, and variations in the Center-of-Gravity (CG) of the payload.

For the current Phase-A research a tractor-flatbed-trailer was studied. This configuration was considerably more torsionally compliant than the tractor-van-trailer. When the test-track experiences with the tractor-van-trailer are compared with those of the tractor-flatbed-trailer, the tractor-van-trailer demonstrated a more systematic and linear approach to axle lift as the vehicle speed was increased. In contrast, the torsionally compliant tractor-flatbed-trailer combination went from no wheel lift to a dramatic wheel lift (that is from zero to 24 inches of wheel lift) within just a few miles-per-hour. As a result, much greater test design detail was required for the tractor-flatbed-trailer than the tractor-van-trailer.

The study team would like to emphasize, however, that vehicle evaluations were performed on a closed test-track with a professional test driver. For the purpose of the study, payloads, speeds and steering inputs were all designed to place the vehicle in limit conditions, and should not be construed as maneuvers or configurations which represent the most typical usage.

The testing conducted in this phase (Phase-A) of the HTRC project is an extension of the previous work done in Phases 1 and 2. In the previous work, a box-trailer was tested with dual tire and NGSWBT tire configurations as well as with standard and wide axles for the box-trailer. While dispersion in the testing results limited the statistical assessment of the data collected in Phases 1 and 2, the data indicated that some combinations of axles and tires, such as wider axles and NGSWBT tires, offered the potential to raise the rollover threshold of the vehicle. The testing in Phase-A was conducted using an aluminum flatbed-trailer and was limited to standard axle widths (78 inches) for both the tractor and flatbed-trailer. For Phase-A, the data was sufficient to provide statistically valid conclusions indicating that the performance of the dual tires and NGSWBTs were equivalent, and that modifications in payload did change the vehicle's roll threshold.

Regarding the use of the tractor ESC; in all cases for which the tractor ESC was engaged, it provided a rapid reduction in lateral acceleration and vehicle speed. Because of safety concerns and resource limitations, determination of the speed margin gained in terms of wheel lift threshold through the use of the tractor ESC was not possible for all load configurations. However, the wheel lift duration was shown to be reduced for all test cases when the tractor ESC was engaged.

Behavior of the vehicle as the payload CG height changed resulted in the expected behavior. That is, higher CG payloads elicit wheel lift at slower speeds.

The team also attempted to utilize a simulation model, set up with faithfully collected physical characteristics for the tractor-flatbed-trailer, to emulate the performance experienced during the on-track testing. For this effort, TruckSim® developed by Mechanical Simulation Corp was used. If such performance could have been emulated, the research team would have a significant capability to address the impact of “what if” design changes in software rather than testing them in the more expensive on-track testing mode. Unfortunately, the behavior of the on-track vehicle for the various maneuvers could not be well emulated in the TruckSim® modeling environment. The reason for the differences is not known. This suggests that: 1) the modeling of vehicle stability is complex, especially for multiple degree of freedom articulated vehicles, 2) that perhaps the vehicle needs to be characterized and/or modeled in more depth than is currently capable, and/or 3) the modeling assumptions and associated equations of motion used in this analysis cannot effectively treat a tractor and flatbed-trailer with the low torsional stiffness properties observed in this case. Future modeling efforts may have to consider other vehicle dynamics models.

Using the combined tests of all of the phases, it appears that some tire and axle combinations hold the potential for improving class 8 tractor-trailer rollover threshold limits. However, there are many other parameters that may be reflected in the testing results, such as the torsional compliance of the flatbed-trailer, the torsional compliance of the tractor, the trailer length, and the load arrangement to name a few. These limit the definitive assessment of axle or tire effects on roll stability. Phase-B of this project will involve the testing of a tractor-tanker-trailer which has a torsionally stiff chassis. The testing will include differing tire combinations. One of the objectives of of the Phase-B testing is to investigate the impact of chassis torsional stiffness on vehicle roll stability. Further, it may be possible to assess if there is any interaction between chassis stiffness and tire selection that increases or decreases the global vehicle stability limits.

RESEARCH TEAM

ORNL was the overall lead of the Phase-A research and provided project oversight, and management, and expertise to support vehicle instrumentation; on-track testing; data collection, management and quality assurance of the collected data; and data analyses. MARC led the research involving the Kinematics and Compliance (K&C) Testing, and developed a procedure to conduct torsional stiffness testing of the test tractor and flatbed-trailer, which they subsequently carried out. MARC was also intimately involved with the design of the test-track testing, its execution, physical characterization of the tractor and trailer, and reflecting all of these efforts within their TruckSim® model. WMU also provided their expertise in the design of the test-track testing, creating solid-body models, and the development of their TruckSim® model of the tractor and flatbed-trailer. WMU also developed several unique roll stability design concepts that they studied via their modeling tools. WMU's analyses suggested that such concepts could provide a reduced rollover propensity for a tractor-flatbed-trailer combination. Battelle conducted an initial investigation into run-off-the-road events, and provided invaluable insights into truck rollover stemming from their experience in the cargo tank rollover area [2].

The research team for this effort built on the relationships that were formed during the Phase 1 and Phase 2 efforts conducted in the 2004-2005 timeframe. Each organization contributed actively to the successful execution of this project, and was able to address associated and complementary research needs and interests. This team also supplied significant and valuable resources that leveraged the NTRCI-UTC/DOT funding and was a great example of a strong and successful public-private partnership.

CONTEXT OF THE PHASE-A RESEARCH

The work conducted by this team focused on efforts to generate data and information on heavy truck rollover not currently available in the industry. It is part of a longer-term research program that will study the rollover propensity of a class 8 tractor-tanker-trailer (with and without tractor and trailer ESC; NGSWBTs vs. traditional dual tires; and different heights of the CG) in Phase-B of this program. The lessons learned and results from this research will be merged with the lessons learned and results of the tractor-van-trailer (Phases 1 and 2), and tractor-flatbed-trailer efforts (Phase-A); with industry experience and expertise; and in conjunction with tractor and trailer original equipment manufacturers (OEMs); to provide a basis for recommending design enhancements that would support enhanced roll stability and safety in class 8 combination vehicles (dubbed the "SafeTruck"). It is the vision of this research team that SafeTruck can be built, evaluated, tested and demonstrated to the industry and government bodies.

This phase of the project (Phase-A) was conducted to understand the roll stability and roll characteristics of a 2007 Volvo model, VT64T830, class 8 heavy-duty tractor with a fully-loaded 48 ft aluminum Utility flatbed-trailer with Intraax air suspension.

TESTING OVERVIEW

Testing was conducted at the TRC in East Liberty, Ohio. TRC was also contracted to provide the data acquisition system; to collect data during the test maneuvers and for executing the on-track testing. There were four test maneuvers selected by the project team for on-track testing, and a detailed test plan (see Chapter 18) was developed as a roadmap for test execution. The test maneuvers selected for rollover testing were:

- Constant radius,
- Ramp steer,
- Step steer, and
- Lane change (highway evasive maneuver).

The ramp steer maneuver was subsequently eliminated because of the successful execution of the high-speed constant radius maneuver. Three different elements of the tractor-flatbed-trailer configuration were varied during the testing;

1) Tires (standard dual tires versus NGSWBTs),
2) Flatbed-trailer payload CG height (two different payload CG heights), and
3) Tractor ESC system (on and off).

It should also be noted that testing was conducted at low speeds and lift threshold limit speeds.

The first three attempts to collect truck rollover data at TRC were unsuccessful. The first attempt revealed that when wheel lift was achieved, it was experienced at the drive axles of the tractor. Because the drive axle lifted first, the outriggers mounted in the center of the flatbed-trailer were felt to be inadequate to prevent the tractor from rolling over and the testing was therefore halted.

For the second attempt the outriggers were moved forward on the flatbed-trailer in order to minimize the potential for the tractor to rollover. However, due to the flexibility inherent in the flatbed-trailer, the outriggers did not adequately protect the rear of the flatbed-trailer from rollover and the testing was halted.

For the third attempt a second set of outriggers was added to the rear of the flatbed-trailer so that the tractor-flatbed-trailer was protected for both drive axle and trailer axle lift. However, during the third attempt at on-track testing, the tractor-flatbed-trailer speed reached relatively high levels without achieving wheel lift and TRC halted further testing.
After considerable discussion it was concluded that in order to achieve a more controllable wheel lift, the CG of the ballast had to be raised, allowing for wheel lift to occur at slower speeds. In order to accommodate this new ballast configuration, a rack system was designed to raise the ballast blocks above the deck to allow the tests to be conducted within a speed and articulation angle envelope that was acceptable to TRC. The fourth attempt to perform rollover testing was successful.

DATA ACQUIRED

During the test maneuvers, the TRC-owned eDAQ data acquisition system was used to collect 33 channels of data at a sampling rate of 102.4 Hz. This resulted in over 168 megabytes of data in the SIF file format (i.e., the format used by eDAQ). The SIF files were converted to over 310 megabytes of data in the more common “comma separated variable” ASCII format for distribution to the team members.

SUMMARY OF RESULTS

Constant Radius Test

To support simulation model validation, steady-state constant radius testing was completed at a low speed to quantify the understeer characteristics of the tractor-flatbed-trailer combination. For the low-speed testing, the data indicated that at low lateral acceleration levels, the vehicle needed approximately three degrees of steering input at the steering wheel above that which would be dictated by the Ackermann geometry in order to maintain the measured radius of curvature. At higher lateral acceleration levels, the steering input approached the Ackermann requirement. This indicates that the vehicle was slightly understeering, but at a very slowly decreasing rate with respect to its lateral acceleration. All of the tested payload and tire combinations had very similar understeer curves. Due to the high-CG payload condition, the steady-state testing was limited to accelerations below 0.2 g. Changing tires or payload heights did not change the understeer response significantly.

For the high-speed constant radius maneuver, the Duncan Multiple Range test that was used to analyze the data showed that the reduction of the payload height by 1 ft was statistically significant, but the selection of tires was not (similar conclusions were derived from the step steer and lane-change maneuvers analysis, as discussed below). The ability to discern a statically significant change in vehicle lift response of a 1 ft payload height change indicated a reasonably well controlled test method. The tests were repeated with the tractor ESC system engaged with the expected result that wheel lift was prevented for all of the constant radius runs at higher speeds.

The high-speed constant radius test data also showed that for the high-CG payload case, the drive axles lifted at a slightly lower lateral acceleration level than the flatbed-trailer axles. This is in contrast to the low-CG payload case wherein the flatbed-trailer axles lifted with the drive axles. Because of the flatbed-trailer's flexibility, which severely limits its ability to effectively transmit torque from one end to the other, it was deduced that the front and rear portions of the flatbed-trailer have very similar roll limits.

The drive axle lift sensitivity was 0.06 g/ft of height change in the payload; 0.12 g/ft in system CG height change (includes the masses of the axle, outrigger, load rack, etc.). The flatbed-trailer axle lift sensitivity was 0.04 g/ft of height change in the payload; 0.067 g /ft in system CG height change. Given that the mass of the unloaded tractor is greater than the unloaded flatbed-trailer, the amount of payload which can be carried over the flatbed-trailer axles is greater than the payload at the drive axles.

Step Steer Test

The step steer test was designed to evaluate the transient behavior of the vehicle using a repeatable steering input. Testing was initially performed without the tractor ESC engaged to evaluate the vehicle's stability, and then with the tractor ESC engaged to assess the benefits of such a system.

Several key vehicle metrics were selected as potential lift indicators including vehicle velocity, lateral acceleration, roll rate, yaw rate, roll angle, and vehicle side slip angle. The results of the statistical analysis showed that payload height and lateral acceleration were well-correlated to the lift event, while the remaining parameters were not. The analysis also showed that it was possible to determine with 95% confidence that the low-CG payload vehicle response was statistically different from the response of the high-CG payload vehicle for both drive and flatbed-trailer axle sets. It was not possible to distinguish between the behaviors of the dual tires and the NGSWBTs for either the drive or flatbed-trailer axle sets. The indication is that the dual tires and the NGSWBTs have similar performance in roll stability for a torsionally compliant flatbed-trailer.

For the cases where the tractor ESC system was engaged with a high-CG payload configuration, the analysis of the data showed that, although the axle lift thresholds for speed or lateral acceleration were similar to those observed with the tractor ESC off condition, the duration of the observed wheel lift decreased. This would be expected since the engagement of the tractor ESC system should tend to bring the vehicle back into a stable operating situation sooner. The data, however, could not be used to statistically quantify the lift duration reduction because the driver's response after a wheel lift occurred, varied significantly. For the low-CG payload case, the tractor ESC system was effective in preventing wheel lift under the same conditions and speeds that wheel lift was observed for this maneuver when the tractor ESC was not engaged.

Lane Change Maneuver Test

For the lane change maneuver, roll stiffness (***RS***), computed as the ratio of the maximum roll angle divided by the maximum lateral acceleration was used as the analysis variable (the variable ***RS*** was computed for the tractor, and the front and rear of the flatbed-trailer). For each lane-change maneuver (a total of 70 runs), two ***RS*** values were computed: one for the first half of the maneuver (left turn) and one for the second half (right turn). Since in the majority of the cases (more than 85%) the distributions of ***RS*** had very dissimilar variances for the different treatments analyzed, a Smith-Satterthwaite Test was used to test the null hypothesis that the means of each pair of compared distributions of ***RS*** (e.g., dual tire ***RS*** vs. NGSWBT ***RS***, for high-CG payload and high-speed) were the same. Concentrating on the first part of the maneuver, the results showed that the null hypothesis could only be rejected with a low level of confidence (less than 95%) when the dual tires were compared against the NGSWBTs for the cases of high-CG payload/high-speed/tractor ESC-off, high-CG payload/low-speed/tractor ESC-off, and high-CG payload/high-speed/tractor ESC-on. This indicates no statistically significant differences between the two types of tires regarding roll stability with this torsionally compliant flatbed-trailer. The tests also seem to indicate that the engagement of the tractor ESC system made the ***RS*** distributions of the dual tires and the NGSWBTs "more equal," since the null hypothesis could only be rejected at lower confidence levels than when the system was not engaged. This difference was more pronounced in the second half of the maneuver, when the tractor ESC system had some time to act (i.e., the first half of a lane change maneuver consisted of an abrupt change in the direction of travel that cannot be predicted by the tractor ESC system). Notice, however, that because only five runs were conducted with the tractor ESC on (three with dual tires and two with NGSWBTs), those tests are not as powerful as those conducted with the tractor ESC off (12 runs; seven with dual tires and five with NGSWBTs).

Regarding the effect of payload CG height change; the results show that for the high-speed case, the null hypothesis could be rejected at a very high level of confidence (close to 99.99%+, except for the tractor ***RS*** comparison in the second half of the maneuver where the null hypothesis could be rejected with 96.9% confidence). This strongly indicates that the 1 ft decrease in the height of the CG payload increases the roll stability of the vehicle substantially (average decreases in ***RS*** of 16% for the tractor and 13%-17%-to-20%-30% for the flatbed-trailer, for the first and second half of the maneuver, respectively, were observed). For the low-speed runs the null hypothesis could only be rejected at the 99% confidence level when considering the ***RS*** distributions for the rear of the flatbed-trailer, indicating that the 1 ft payload CG height decrease showed statistically significant gains in roll stability for the flatbed-trailer. For the tractor and the front of the flatbed-trailer, the null hypotheses could be rejected with 98% and 90% confidence level, respectively.

In summary, the results of the field tests show that, in terms of roll stability, the reduction of the payload height by 1 ft was statistically significant in affecting the roll stability for all of the maneuvers, but the selection of tires was not. Although there were very few runs conducted with the tractor ESC system engaged, the results of the tests seem to indicate that there is a positive contribution of this system in terms of roll stability. However, the testing also indicated that the tractor ESC system would not prevent wheel lift under some situations (i.e., for some speed, steering and load combinations) but appears to reduce the duration of the wheel lift. The results suggest that the effect of the tractor ESC needs to be further and specifically investigated, for the lane change maneuver. Additionally, to address the issue of wheel lift duration, a different vehicle test strategy and set of measurements will be required.

MODELING RESULTS

For the modeling analysis, TruckSim® (a commercial application) was utilized as the platform to develop a functional model of the vehicle using the kinematics and compliance data as well as manufacturers' design/build data sheets that were gathered for the tractor and flatbed-trailer in this phase of the project. Particular attention was paid to inputs such as axle motions and suspension stiffness, as well as chassis rigidity.

Constant Radius Test

Both the low- and high-speed constant radius tests were evaluated in the model to determine if the model could reproduce the performance of the vehicle as observed during the field tests. The low-speed model was executed using a constant radius turn of 75 m (the same as the field test), and a speed increment of 1 km/h (0.62 mph) every 3 sec (slightly slower than that observed in the field tests). The model was found to be sensitive to the torsional stiffness parameters for the tractor and flatbed-trailer. When the measured vehicle torsional stiffness was used, the model did not correctly predict the understeer response of the vehicle. Artificially stiffening the chassis in the model helped to reproduce the correct slope in the steering vs. lateral acceleration plot, but it over-predicted the amount of understeer by about 5 deg at the steering wheel.

The high-speed model was evaluated using a slow constant steering rate of 5 deg/s at the steering wheel, with the truck traveling at 61 km/h (37.9 mph) (this speed was selected because it was the lift speed that was realized in the field testing). Using the measured torsional stiffness values for the vehicle, the simulation model significantly under-predicted the wheel lift threshold for both the tractor and the flatbed-trailer in both the high-CG and low-CG payload configurations. Increasing the torsional stiffness of the model by an order of magnitude accurately emulated the same drive axle lift threshold for the high-CG payload condition, but not for the low-CG payload condition. The flatbed-trailer axle lift threshold was not correctly predicted for either load condition using the stiffened chassis. Further investigation is needed into the model representation of torsional stiffness, and the vehicle system sensitivity to this parameter.

Step Steer Test

As in the previous case, the modeling efforts for the step steer were also affected by torsional stiffness questions which, in general, were the same as those in the high-speed constant radius testing. The simulation model was fairly close at predicting the lift threshold of the tractor for the artificially stiffened chassis case, as well as the flatbed-trailer lift point for the "as-measured" chassis case. However, the remaining lift points were not correctly estimated, and neither chassis model correctly predicted the lift sequence of flatbed-trailer axle lift before drive axle lift.

In summary, the torsional stiffness of the flatbed-trailer used in the TruckSim® model has a significant impact on the overall modeled vehicle behavior. Understanding how the chassis of a flexible vehicle behaves is important for understanding the overall vehicle's response to various inputs. Furthermore, modeling such torsional flexibility takes considerable care, is not trivial, and is a significant area for future research.

Solid Modeling

The flatbed-trailer used in this study was scanned by WMU using an ATOS II white light scanning system. This system produces a point cloud from which very precise computer models can be generated. The scan produced a highly precise, material specific model which can be used in a library for the development of a future truck concept. With the precise representation obtained, the WMU team was able to produce a finite element model for structural analysis testing. The structural model matched the experimentally determined torsional characteristics of the actual flatbed-trailer used in testing.

WMU also developed an ADAMS kinematic model of the flatbed-trailer suspension which allows selected parameters to be varied to assess their influence on the overall tractor-flatbed-trailer model. The solid models, finite element models, and kinematic models will prove to be important as this project moves into the design recommendation phase. These models provide the foundation for full flex body modeling which will enable the assessment of road surface encounters and their influence throughout the model.

Advanced Design Concepts

Advanced design concepts were explored that extend beyond the current-day truck design. As part of this study several tractor-trailer design concepts were suggested and initially investigated. These included: a moveable 5^{th} wheel for the tractor, a deployable third trailer axle, and an active suspension concept; all of which were addressed through modeling efforts. Although these are still considered to be in the initial stages of a feasibility study, all have shown, at least conceptually, some potential for improvement of the roll stability of a combination vehicle. These concepts, as well as others that might present themselves in future phases, will be further studied for their applicability.

Vehicle Handling Characteristics

The as-tested tractor-flatbed-trailer was modeled and evaluated using a simulation program as well as studied from a parametric perspective. As part of this study the importance of looking at the class 8 tractor-flatbed-trailer as a system was stressed. To assist in evaluating parameters in the tractor-flatbed-trailer design that might ultimately lead to improvements in rollover threshold, a unique software program was developed by WMU. The program predicts quite well some of the handling characteristics of the tractor and flatbed-trailer, and has design attributes that allows one to study, conceptually, the influence of a selected attribute on stability. Design parameters such as axle wall thickness, suspension bushing characteristics, etc. can be altered to benchmark their influence on the dynamics of the tractor-flatbed-trailer. This program allows variations in load placement, frame structural properties, and details of the suspension. Future work will potentially include the development of a complete and accurate tractor model and further advance the study of the combination vehicle as a system.

FUTURE PROGRAM EFFORTS

Completion of the Phase-B efforts in 2009 will have provided the research team with vehicle stability experience across three different trailer platforms (box-trailer, flatbed-trailer and tanker-trailer); may provide insights into the relationship between torsional stiffness and the benefits of NGSWBTs for roll stability; and given insights into the behavior of tractors and trailers equipped with ESC and advanced suspension systems in severe maneuver conditions. Such a rich experience base will allow the HTRC to be highly qualified in putting together a larger team involving tractor and trailer OEMs and major first tier suppliers to design, develop, evaluate, test and demonstrate an integrated tractor-trailer concept that reflects significant and enhanced roll stability for class 8 combination vehicles.

CHAPTER 1. INTRODUCTION/BACKGROUND

INTRODUCTION

Heavy truck rollover crashes are not frequent occurrences (approximately 3% of all crashes for combination trucks). Although this percentage is low, fatalities associated with heavy truck rollovers are disproportionately high. Truck rollover is a factor in about 52 percent of all fatal crashes of combination trucks. In addition, it is not unusual for the economic losses of the payload and other associated insurance costs to be a significant cost to the transporter.

Understanding the interactions of vehicle payload, tires, suspensions, vehicle types, vehicle stiffness (tractor and trailer, individually and in combination), and roadway surfaces/tire interface on truck rollover events could contribute significantly to improving heavy truck safety. Such understanding can be applied to support the design and evaluation of new technologies such as wider axles, wider single tires, adaptive suspension systems, rollover warning systems, etc. It could also contribute to improving roadway design to minimize the potential for truck rollover stemming from vehicle-highway interactions, and could contribute to more effective regulation aimed at reducing truck rollovers. Additionally, such understanding is essential for the development of next generation heavy vehicle dynamics models that would be valuable tools for evaluating new product innovation, and for the assessment of the interactions associated with complex vehicle dynamics.

Understanding the dynamics and characteristics of heavy truck rollovers is complex, and cannot be achieved thoroughly without the conduct of physical vehicle testing, the collection of vehicle dynamics data, development of information on truck rollover performance, and the use of specialized lab equipment such as kinematics and compliance testers and torsional stiffness testers. In addition to being complex, limited resources for truck rollover research necessitates that it be conducted in a partnership that involves federal agencies, national laboratories, universities and private industry; and be conducted with a long-term perspective.

BACKGROUND

Prior heavy truck rollover characterization research was conducted by NTRCI in partnership with ORNL, Dana Corporation, MARC, Clemson University, WMU Center for Advanced Vehicle Design and Simulation) and Battelle. Phases 1 and 2 of the Heavy Truck Rollover Characterization Research Program were conducted in 2004-2005 for the NTRCI under a Grant from the Federal Highway Administration (FHWA). This work focused on characterizing a tractor-van-trailer, and engaged in initial efforts to generate data and information on truck rollover that was not currently available in the industry. In particular, the research of Phases 1 and 2 focused attention on the contribution of NGSWBTs and wider-slider suspensions toward reducing the rollover propensity for a tractor-van-trailer combination.

From 2007-2008, research was conducted by ORNL, MARC, WMU/CAViDS and Battelle (partners in the HTRC) for the NTRCI University Transportation Center (NTRCI-UTC) under a Grant from the Research and Innovative Technology Administration (RITA). This work (Phase-

A of the Heavy Truck Rollover Characterization Research Program) focused on characterizing a tractor-flatbed-trailer, and generated further data and insights on heavy truck rollover. In particular, the research of Phase-A focused attention on the contribution of tires, tractor ESC, and payload CG height on roll stability.

Overview of Phase 1

Phase 1 involved a 1999 Peterbilt 379 class 8 tractor and 2004 Wabash dry freight van trailer with dual tires on the tractor drive axles and on the box-trailer axles (both equipped with rear air suspensions). The tractor and box-trailer were instrumented with a number of sensors to capture their dynamics as they engaged in various testing maneuvers (an evasive maneuver, swept sine, constant radius, and a run-off-the-road maneuver). Candidate maneuvers considered for this project were derived by examining various vehicle dynamics sources including ISO standards and SAE papers. Phase 1 testing generated controlled baseline data that was usable for comparison with Phase 2 results and all future testing in this research program.

For all of the tests, the box-trailer was loaded with ballast for a gross vehicle weight rating of 79,000 lbs., and the speeds were gradually increased so that wheel lift was experienced both visually and via instrumentation.

Overview of Phase 2

Testing in Phase 2 involved the same maneuvers as conducted in Phase 1, but included the use of NGSWBTs on the tractor, on the box-trailer, and both. Phase 2 also involved testing with and without a wider-slider suspension on the trailer.

Results of Phase 1 and 2 Research

A significant amount of data was collected on all maneuvers performed (1.2 Gigabytes of data from 45 data channels sampled at 0.01 sec), and information was also captured via videotaping (one camera inside the cabin and three others outside; plus one off-board camera). Once wheel lift was experienced, six repetitions were executed. This number of repetitions were the maximum number that could be afforded by the available funding, and provided sufficient confidence (albeit not statistical significance) to demonstrate the trends and patterns in the tractor-van-trailer roll phenomenon.

Data analysis results indicated that for the tractor-van-trailer combination, whenever NGSWBTs were used on the tractor, box-trailer or both, that at the moment of wheel lift, the roll stability index (RSI – the maximum angular displacement of the box-trailer per maximum lateral acceleration) was always less than that which was derived for the tractor-van-trailer configuration using all standard dual tires. A lower RSI indicates a more stable vehicle system. Use of the wider-slider suspension on the box-trailer further reduced the RSI.

Overview of Phase-A

Phase-A was conducted to understand the roll stability and roll characteristics of a 2007 Volvo model, VT64T830, class 8 heavy duty tractor (see figure 1) with a fully-loaded 48 ft aluminum Utility flatbed-trailer with Intraax air suspension (see figure 2).

Results of Phase-A Research

Phase-A testing proved to be considerably more difficult than testing in Phases 1 and 2. One of the primary differences was the high degree of torsional flexibility inherent within the flatbed-trailer design. Unlike the tractor-van-trailer combination, testing of the tractor-flatbed-trailer indicated no statistically significant differences between the standard dual tires and the NGSWBTs regarding roll stability with this torsionally flexible flatbed-trailer. This included testing with the tractor ESC on and off, and varying the payload CG height. The dynamic behavior of the box-trailer vs. flatbed-trail in on-track testing was also different. In the former, the box-trailer rear-axle wheel-lift (first wheel lift in a maneuver) was experienced almost exclusively, while for the latter (flatbed-trailer), the first wheel lift was almost always the tractor's rear drive-axle.

Significant modeling efforts were also engaged in by MARC and WMU using the commercially available TruckSim® model. Although the tractor-flatbed-trailer was faithfully characterized (including data generated from K&C and torsional stiffness testing), the TruckSim® model could not emulate the tractor-flatbed-trailer behavior exhibited during on-track testing. One possible reason for the modeling issues could be that the torsional stiffness of the flatbed-trailer mocked-up in the TruckSim® model appears to have a significant impact on the overall modeled vehicle behavior. The effect of this parameter is suspected as a potential reason for TruckSim® not accurately predicting the wheel lift points, but it is not known for certain that this is the reason for the modeling errors. Understanding how the chassis of a flexible vehicle behaves is important for understanding the overall vehicle's response to various inputs. Furthermore, modeling this flexibility takes considerable care, is not trivial, and is a significant area for future research.

Figure 1. Photograph. 2007 Volvo Model, VT64T830, Class 8 Heavy Duty Tractor.

Figure 2. Photograph. 48 Ft Aluminum Utility Flatbed-Trailer With Intraax Air Suspension.

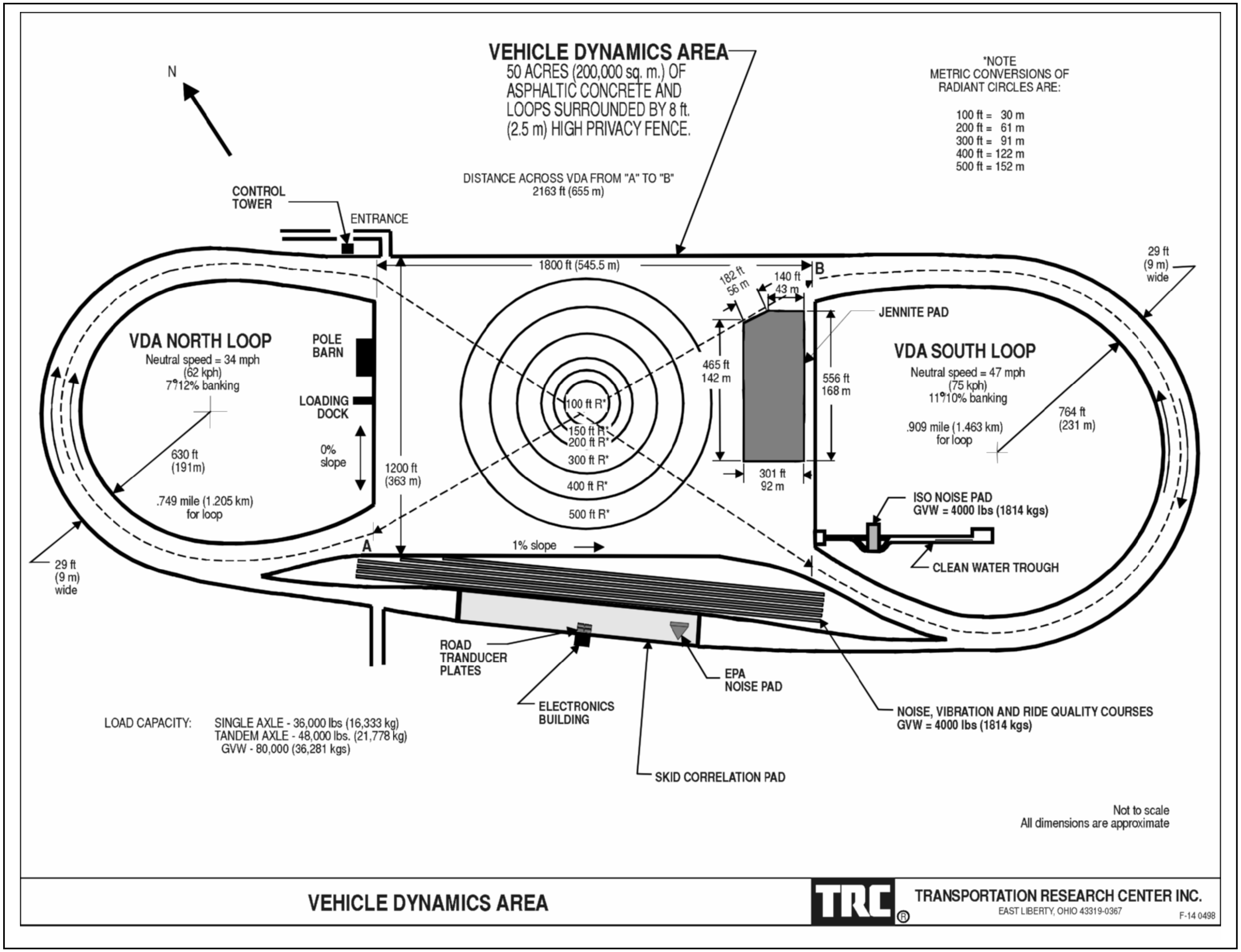


Figure 3. Diagram. TRC, Inc.

Solid body models of the flatbed-trailer's suspension were also created by WMU. These models allowed precise finite element models to be created for input into WMU's kinematic models. These models will prove to be important as this project moves into the design recommendation phase (Phase-C). These models provide the foundation for full flex body modeling which will enable the assessment of road surface encounters and their influence throughout the model.

Lastly, advanced design concepts were explored by WMU that extend beyond the current-day tractor-trailer design. As part of this study several tractor-trailer design concepts were suggested and initially investigated. These include:

- A moveable 5^{th} wheel for the tractor,
- A deployable third trailer axle, and
- An active suspension concept.

Each of these three was addressed through WMU's modeling efforts. Although they are still considered to be in the initial stages of a feasibility study, all have shown, at least conceptually, some potential for improvement of the roll stability of a combination vehicle. These concepts, as well as others that might present themselves in Phase-B, will be further studied for their applicability.

CHAPTER 2. PROJECT OVERVIEW

TRUCK ROLLOVER TEST PLAN

The purpose of the Truck Rollover Test Plan was to provide guidance to the HTRC research team for conducting a heavy truck rollover test at a chosen test-track and to describe the related truck configuration, data acquisition configuration, equipment, and test events. The test plan provided a day-by-day script of the activities to be performed at the test-track. It includes information on:

- The tire configuration for each test,
- The tire change schedule,
- The maneuver to be performed,
- Tractor ESC on or off,
- Instructions to the driver for executing each maneuver,
- The speeds for which the maneuver is to be executed,
- The number of repetitions,
- The direction of travel for the maneuvers (clockwise or counter-clockwise; left or right),
- A description of the vehicle configuration,
- Payload location,
- Payload CG height,
- Frequency for sensor data collection,
- A list of identified contingencies, and
- A list of recommendations to deal with any emerging contingencies.

A copy of the final Phase-B Truck Rollover Test Plan is provided in 0.

SELECTION OF TEST-TRACK FACILITIES

Testing for Phases 1 and 2 were conducted at TRC in East Liberty, Ohio. For those tests, a company, Michigan Scientific of Milford, Michigan was utilized to instrument the tractor-van-trailer, including a data acquisition system and a pair of wheel-end-torque sensors. Dana Corp., of Kalamazoo, Michigan, a HTRC partner for Phases 1 and 2, donated the use of their pair of wheel-end torque sensors, and paid for the efforts of Michigan Scientific. TRC's functions for the testing in Phases 1 and 2 were to:

- Provide access to the TRC facility/test-tracks to the HTRC test team,
- Provide the driver for executing the maneuvers, and
- Add the TRC-owned ballast into the box-trailer.

For those tests, Michigan Scientific also provided on-site engineering support responsible for assuring that all of the instrumentation was functioning as needed.

For the Phase-A testing, the HTRC considered conducting the testing at other test-track facilities besides TRC. Because Dana Corp. and Michigan Scientific were not involved in the Phase-A activities, the task of instrumentation, including the configuration of the data acquisition system (i.e., the eDAQ system) would be required from the test-track staff. The HTRC selected TRC for the conduct of the Phase-A testing. TRC's functions for the testing in Phase-A were to:

- Provide access to the TRC facility/test-tracks to the HTRC test team,
- Provide the instrumentation and instrument the tractor and flatbed-trailer as specified by the HTRC,
- Provide an eDAQ system, configure it for the data to be collected, and install it in the test tractor,
- Provide the driver for executing the maneuvers,
- Mount and dismount the tires as needed, and
- Add the Michelin-owned ballast onto the flatbed-trailer.

TRC was also contracted to build two custom load racks, as specified by the HTRC for use in the testing, when it was determined that the Michelin payload distribution was not going to result in the desired maneuver performance.

INSTRUMENTATION OF THE TEST TRACTOR-FLATBED-TRAILER

Instrumentation of the test tractor-flatbed-trailer was the responsibility of TRC. The HTRC test team provided oversight to the instrumentation effort and handled instrumentation-issues at times when TRC technicians were not readily available (e.g., on weekends). The tractor-flatbed-trailer was instrumented to collect data on 33 channels of data. A list of data channels for which data was collected in the Phase-A testing is provided in table 1. A working-grade graphic of the test tractor-flatbed-trailer showing placement of some of the location sensitive sensors is provided in .

Table 1. List of Phase-A Data Channels

Channel Name	Channel Units	Description
TimeStamp	sec	Time
speed_mph	mph	Speed from GPS
lon	deg	Longitude position GPS
lat	deg	Latitude position GPS
TRKVELOCITY	mph	Tractor velocity, forward
TRKSIDEVX	mph	Tractor lateral velocity, side
LRINTTRKSG	us*	Strain gauge – axle 2 left
RRINTTRKSG	us*	Strain gauge – axle 2 right
LRDRVTRKSG	us*	Strain gauge – axle 3 left,
RRDRVTRKSG	us*	Strain gauge – axle 3 right
TRKSTRANGLE	deg	Tractor steer angle – angle of front wheels
TRKSTRWHANGL	deg	Tractor steering wheel angle
TRKLHRIDEHGT	in	Tractor ride height - left
TRKRHRIDEHGT	in	Tractor ride height - right
TRKXAXIS	g	Acceleration, longitude (forward)
TRKYAXIS	g	Acceleration, lateral (side)
TRKZAXIS	g	Acceleration, vertical
TRKROLLRATE	deg/sec	Tractor roll rate
TRKPITCHRATE	deg/sec	Tractor pitch rate
TRKYAWRATE	deg/sec	Tractor yaw rate
TRLANGLE	deg	Flatbed-trailer angle, angle tractor and flatbed-trailer
TRLLFRIDEHGT	in	Flatbed-trailer left, front ride height
TRLRFRIDEHGT	in	Flatbed-trailer right, front ride height
LRTRLHGT	in	Flatbed-trailer left, rear ride height
RRTRLHGT	in	Flatbed-trailer right, rear ride height
TRLVX	mph	** Flatbed-trailer lateral velocity, side
TRLSIDEVX	mph	** Flatbed-trailer velocity, forward
TRLYAXIS	g	Acceleration, flatbed-trailer
TRLYAWRATE	deg/sec	Yaw rate, flatbed-trailer
TRLLFSTRAIN	us*	Strain gauge – axle 4 left
TRLRFSTRAIN	us*	Strain gauge – axle 4 right
TRLLRSTRAIN	us*	Strain gauge – axle 5 left
TRKXAxis2	g	Tractor acceleration (duplicate signal to sync eDAQ units)
TRLRRSTRAIN	us*	Strain gauge – axle 5 right

* Strain gage native units
**Inverted channel names; data collected according to channel description

ON-TRACK TESTING OVERVIEW

On-track testing was conducted at the TRC in East Liberty, Ohio. TRC was also contracted to provide the data acquisition system; to collect data during the test maneuvers; and, for the testing conducted in May, 2008, TRC designed and constructed new load racks per the specifications provided by the Phase-A project team. The four test maneuvers were selected by the project team, and a detailed test plan was developed as a roadmap for test execution. The test maneuvers selected for rollover testing were:

1) Constant radius,
2) Ramp steer,
3) Step steer, and
4) Lane change (highway evasive maneuver).

The ramp steer maneuver was subsequently eliminated because of the successful execution of the high-speed constant radius maneuver. Three different elements of the heavy duty tractor-trailer configuration were varied during the testing;

1) Tires (standard duals versus New Generation Single Wide-Based Tires (NGSWBTs)),
2) Trailer payload CG height (low and high position), and
3) Tractor ESC system (on and off).

The parameters that were varied during the on-track testing were:

- Tires (standard dual tires and NGSWBTs),
- ESC (engaged and not-engaged),
- Payload CG height (high-payload and low-payload), and
- Speed (high-speed and low-speed).

OVERVIEW OF KINEMATICS AND COMPLIANCE (K&C) TESTING

K&C testing is an essential element for development of functional vehicle dynamic simulation models such as TruckSim®. Functional simulation models use the descriptions of relative motion between each wheel plane relative to the chassis as a function of suspension and steering displacement (kinematics) and tire forces (compliance). MARC conducted K&C testing on both the Volvo tractor (see figure 5) and the Utility flatbed-trailer (see figure 6). Data from these tests were utilized within MARC and WMU TruckSim® models, and within other WMU vehicle dynamics models.

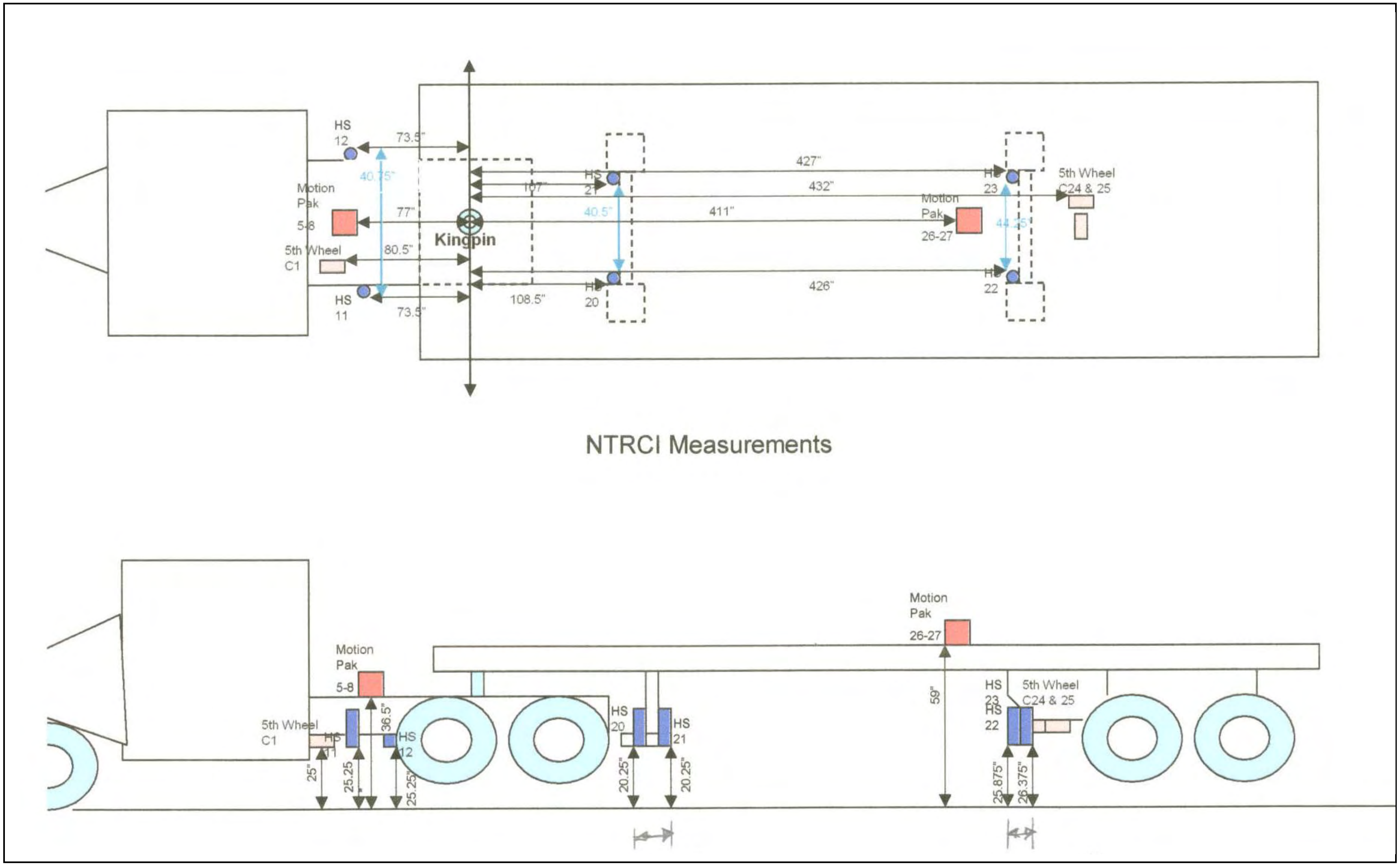


Figure 4. Diagram. Location of Selected Sensors on the Test Tractor-Flatbed-Trailer.

Figure 5. Photograph. Volvo VT64T830 Tractor on the K&C Test Rig at MARC.

OVERVIEW OF TORSIONAL STIFFNESS TESTING

At the initiation of this project, no known resource existed for the measurement of the torsional rigidity of trailers. In order to acquire the torsional stiffness data for the flatbed-trailer for the Phase-A modeling work, a simple and cost effective procedure was designed by MARC. The procedure uses the MARC K&C rig with the roll moment input through the K&C roll beam. The flatbed trailer is fixed to a tractor at the fifth wheel. The torsional stiffness device uses load cells under the rear axles to measure the chassis torque at the rear of the flatbed-trailer. A known input torque was applied to the 5th wheel. Inclination of the trailer at the 5th wheel and the rear axles was measured. Difference in torques and angles defines the torsional rigidity. Figure 7 and figure 8 provide graphics that illustrate MARC's torsional stiffness rig.

WMU also collected torsional stiffness data for the flatbed-trailer via a field procedure that was developed by them. The procedure involved securing one end of the flatbed-trailer, and measuring the angular displacement caused by adding a shim-set to one side of the flatbed-trailer. The procedure is described in more detail in 0 of this report.

Figure 6. Photograph. Utility Trailer on the K&C Test Rig at MARC.

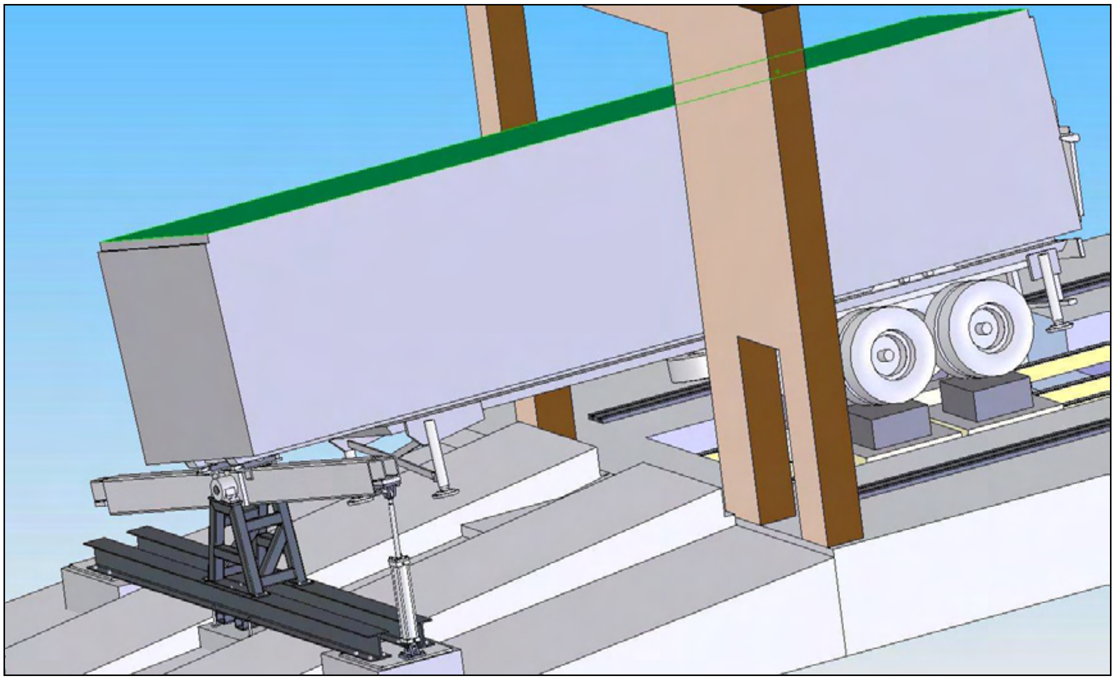


Figure 7. Illustration. MARC's Torsional Stiffness Testing Apparatus (side-view).

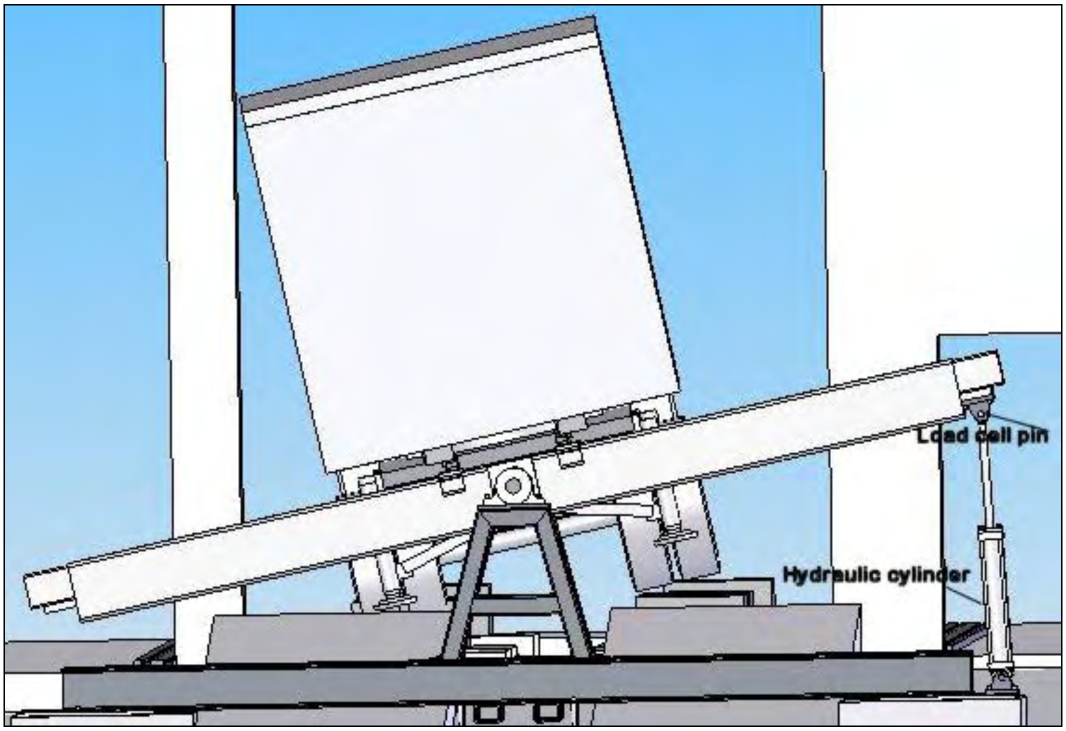


Figure 8. Illustration. MARC's Torsional Stiffness Testing Apparatus (front-view).

OVERVIEW OF THE DATA ANALYSES

ORNL and MARC both engaged in analyzing the data generated during the on-track testing.

ORNL focused on the statistical analysis of the lane change maneuver (a transient maneuver) and looked at the impact of the tires, the tractor ESC (on and off), and the change of the payload CG height on roll stability. ORNL's statistical analysis involved the calculation of the maximum roll stiffness (maximum angular displacement per maximum lateral acceleration) at various locations along the tractor-flatbed-trailer. A Smith-Satterthwaite Test was also employed. Details and results of ORNL's analysis are presented in 0.

MARC conducted statistical analysis for the constant radius (a steady-state maneuver), and the step steer (a transient maneuver); however, a different statistical analysis approach was taken. MARC utilized the Duncan Multiple Range test to study the constant radius maneuver, and logistic regression methods employing a stepwise procedure for studying the step steer maneuver. Details and results of MARC's analysis are presented in 0.

Although the approaches taken by ORNL and MARC were different, the primary conclusions were similar. Those conclusions were that the dual tires and NGSWBTS had similar impacts on roll stability both for the tractor ESC being on and off, and for the high- and low-payload CG heights. It was also seen that a 1 ft change in the payload-CG height had a significant impact on the roll stability, and with the tractor ESC engaged, the duration of a wheel lift event was shorter than without the ESC engaged. Lastly, with the tractor ESC engaged, the severity of a roll incident could be reduced; however, there are certain extreme conditions with the tractor ESC engaged that would not prevent wheel lift, but as stated previously, its duration was reduced from the similar case without the tractor ESC engaged.

OVERVIEW OF MODELING EFFORTS

Modeling efforts were engaged in by both MARC and WMU. The models that were used/developed were either functional models (TruckSim®) or component models (ADAMS, a Finite Element Analysis model, and a kinematics model built in MATLAB/Simulink/SimMechanics).

TruckSim® models were set up by both MARC and WMU independently of each other. Inputs for the TruckSim® models were gathered from MARC's K&C testing, the torsional stiffness testing conducted by MARC and by WMU and from manufactures' design/build data sheets. Both versions of the TruckSim® model faithfully represented the tractor-flatbed-trailer utilized within the on-track testing. However, neither TruckSim® model could effectively replicate the performance of the tractor-flatbed-trailer on the test-track. The high degree of torsional flexibility of the tractor-flatbed-trailer combination, and the complexity of effectively modeling torsionally compliant vehicles could be factors that contribute to the lack of agreement between the TruckSim® models and the test-track results. Further study is needed to better understand this difference. Details of the TruckSim® modeling efforts are presented in 0.

WMU's component models were developed as part of a future modeling analysis suite for evaluating advanced roll stability design concepts and technologies. In Phase-A, the flatbed-trailer was scanned by WMU using an ATOS II white light scanning system. This system produces a point cloud from which very precise computer models can be generated. A highly precise, material-specific model was developed from these scans. The model can be used in a library for development of a future truck concept. With the precise representation obtained, the team was able to produce a finite element analysis (FEA) model for structural analysis testing. The structural model matched the experimentally determined torsional characteristics of the actual flatbed-trailer used in testing.

WMU also developed an ADAMS kinematic model of the flatbed-trailer suspension which allows selected parameters to be varied to assess their influence on the overall truck model. The solid models, FEA models, and kinematic models will prove to be important as this project moves into the design recommendation phase. These models provide the foundation for full flex body modeling which will enable the assessment of road surface encounters and their influence throughout the model. Details of the WMU modeling efforts are presented in 0.

PROJECT SCHEDULE

The Heavy Truck Rollover Characterization Phase-A efforts were originally planned to be conducted from April 1, 2007 through March 31, 2008. For this initial schedule, on-track testing was scheduled to have been conducted in October, 2007. As noted in 0, the October, 2007 testing attempt, as well as testing attempts in November, 2007 and January, 2008 were unsuccessful. Following the January, 2008 testing attempt, the HTRC felt that it was not possible to maintain a completion date of March 31, 2008 for the Phase-A efforts. This decision was driven primarily by the inability to conduct testing at TRC until the weather was more cooperative (several days of test-track time were compromised due to weather in the January, 2008 on-track testing attempts). The HTRC and NTRCI selected test-track testing to be conducted in May, 2008, and agreed on the ending date for Phase-A efforts to be September 30, 2008. Phase-B efforts are planned for initiation on November 1, 2008, but in order to catch-up on the overall Heavy Truck Rollover Characterization Program schedule, the time span for Phase-B efforts will be only nine months; concluding on July 31, 2009. The scheduling of on-track testing for Phase-B efforts needs to consider the time of year in which testing is planned, and to consider this as part of decision making with regard to which test-track the HTRC selects to conduct the Phase-B testing. (Note: the run-off-road work is planned for the next year as a separate effort from Phase B, although also funded by NTRCI.)

CHAPTER 3. PARTNER ROLES

The Heavy Truck Rollover Characterization Phase-A research was conducted by the HTRC which is composed of organizational participants from national laboratories, academia, non-profit organizations and private industry. This section provides a brief overview of the roles that each of the HTRC partners had in the Phase-A research

THE OAK RIDGE NATIONAL LABORATORY (ORNL)

The Oak Ridge National Laboratory (ORNL), which is the largest Department of Energy (DOE) science laboratory, was the primary technical lead and project manager for the Phase-A effort. ORNL conducted a similar role in the Phase 1 and 2 Truck Rollover Characterization efforts for NTRCI.

ORNL also had a significant role in the Phase-A instrumentation and testing which was based on the significant prior test-track and field operational test experience gained in conducting projects for various agencies of DOT, DOE and for the NTRCI.

ORNL had a major role in the Phase-A data analysis of field-test and test-track data. Similar work has been done for various agencies of DOT, DOE and the NTRCI.

MICHELIN AMERICAS RESEARCH COMPANY (MARC)

Michelin Americas Research Company (MARC) was the primary industry technical lead for the Phase-A research. MARC was one of two primary industry participants in the Phase 1 and 2 Heavy Truck Rollover Characterization efforts. MARC's roles in Phase-A included the conduct of K&C and torsional stiffness testing for the tractor and flatbed-trailer,, simulation modeling, support for on-track testing, and data analyses.

WESTERN MICHIGAN UNIVERSITY/CENTER FOR ADVANCED VEHICLE DESIGN AND SIMULATION (WMU/CAVIDS)

Western Michigan University/Center for Advanced Vehicle Design and Simulation (WMU/CAViDS) provided experience in vehicle dynamics and computer modeling and was the lead in truck and trailer product design as a pre-cursor activity for Phases C and D.

WMU/CAViDS assisted in developing the Phase-A testing regimen; developed a solid model of the flatbed-trailer; and developed some suggested alternative component directions for the tractor and flatbed-trailer design.

BATTELLE MEMORIAL INSTITUTE (BATTELLE)

Battelle Memorial Institute (Battelle) was the lead for investigating the run-off-the-road maneuver that will be accomplished as another NTRCI-funded research project. Efforts included a review of the run-off-the-road literature, assessment of test-track candidates for conducting the associated experiments, and a recommendation for how to proceed with the evaluation of advanced designs and technologies for future efforts.

VOLVO TRUCKS NORTH AMERICA (VOLVO)

Volvo provided technical advice on the execution of the various on-track tests, and was the builder of the leased VT 830 tractor used in testing. Initially, Volvo prepared a preliminary high-roof tractor for Phase-A testing. The HTRC subsequently selected a newer mid-roof sleeper unit (Volvo VT64T830) with the “factory” installed “VEST” (Volvo’s proprietary roll stability system) as the test tractor. Volvo also engaged in a number of tractor-related maintenance efforts in preparation for, during, and subsequent to the testing. This included work that was associated with the tractor’s electrical system and its engine control module. Volvo’s primary roles and responsibilities involved the assurance of minimal down-time for the tractor and arranging for prompt logistics for the tractor as required.

CHAPTER 4. SUMMARY OF TRUCK ROLLOVER LITERATURE REVIEW

MARC conducted a literature review of heavy truck rollovers early in Phase-A. The review focused on untripped events and compiled information related to objective field testing, kinematics testing and modeling. It was reported that there are many constraints on the design of a tractor and trailer. For the trailer, design constraints result from docking, flooring in the trailer to accommodate drive-on/off loaders, and clearance of relatively tall tires. Tractor constraints include the size and weight of the engine, trailer attachment requirements, chassis flexibility in managing ground elevation changes, manufacturing constraints and driver needs. Other constraints by the US government include size, weight and height.

For a tractor-trailer, the literature reports that there is a 23 percent increase in the likelihood of a tractor-trailer rollover for each 10 percent increase in payload weight, and a 49 percent increase in the likelihood for each 10% increase in speed.

The stability of a tractor-trailer is limited by its relatively high CG, vehicle compliances, connection tolerances, and suspension clearances. The typical effects of the suspension compliances and component clearances on basic vehicle stability are shown in figure 9. This stacking of the compliances and clearances helps to demonstrate why the rollover threshold is, in general, lower for many tractor-trailers. These vehicles are limited by both functional design constraints and realistic component operational limits. The result is that the typical-tractor-trailer rollover threshold is typically in the 0.3-to-0.4 g range. Considering that the US guidelines for highway curve design result in lateral accelerations as high as 0.17 g at the advised speed limits [2], and that drivers maneuver their vehicles at well over 0.2 g fairly regularly [2], the rollover margin of the vehicle for the given road conditions is smaller than that for other vehicles which use the public road system.

The literature review discusses the physics of rollovers and includes information on roll moments, roll axes, centripetal acceleration, axle rollover thresholds, coupled vehicles, rigidity effects, and lateral and torsional compliance. Various testing and evaluation approaches are also discussed including tilt-table testing, test-track/field testing and modeling (including discussions on multi-body component modeling and kinematics-based modeling). Along with the discussion on testing and evaluation methods is a review of common metrics for the evaluation of rollover. These metrics include lateral acceleration, yaw rate gain, roll rate, and vehicle side slip.

The literature review also discusses various approaches to achieve enhanced roll stability including active suspensions, ESC systems and yaw and roll control. A copy of the literature review is provided in 0 (Appendix A).

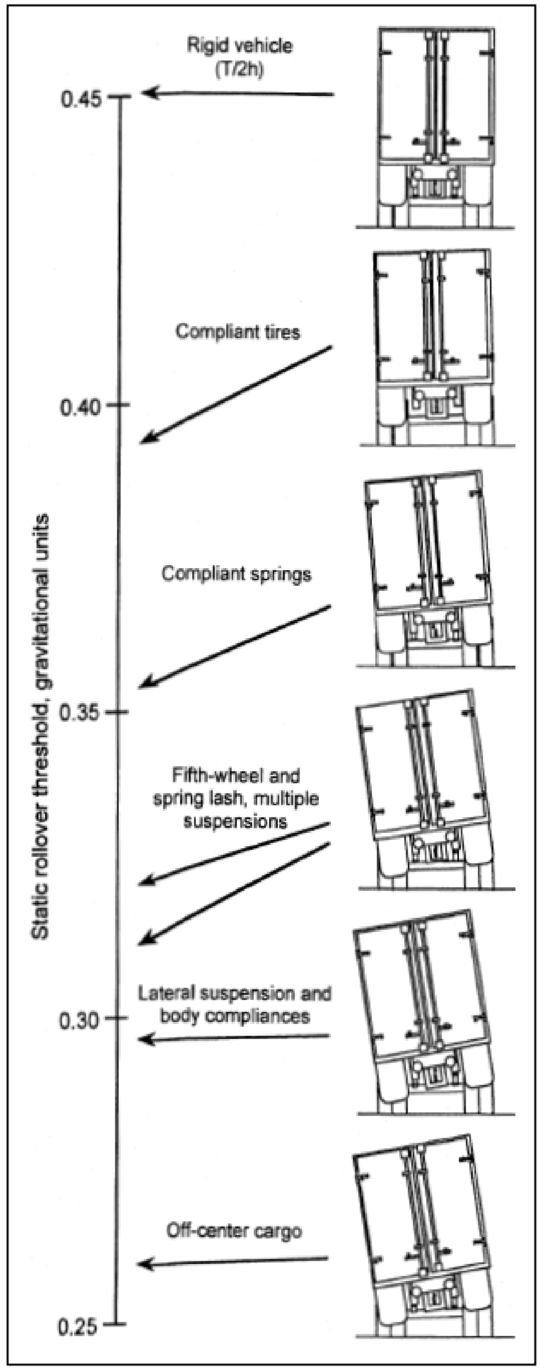


Figure 9. Illustration. Stability Limiting Sources [3].

CHAPTER 5. PLANNING FOR PAVEMENT EDGE DROP-OFF EXPERIMENTS

Motor vehicle rollovers fall in two broad categories; tripped and untripped. A rollover is said to be tripped if some fixed object suddenly strikes the tires and upsets the vehicle. This can happen by striking a curb or guardrail or by tumbling down an embankment. A rollover would also be termed tripped if a lateral force on a tire is suddenly released; as when a tire is scrubbing a vertical pavement edge but quickly climbs the edge. Untripped rollovers, on the other hand, occur on essentially level pavement with the overturning moment being simply the cornering force on the tires, opposed by the centrifugal force at the CG.

The test-track experiments conducted in this project duplicated scenarios for untripped rollovers. The test tractor-flatbed-trailer was in a curve on level pavement. The Office of Safety Research of the Turner-Fairbank Highway Research Center has recommended that NTRCI consider conducting an investigation of heavy vehicle roll stability following a roadway departure over a pavement edge drop-off. This section of the Phase-A Final Report describes the need for heavy vehicle pavement edge drop-off research, and outlines a safe, but meaningful test of this type of roll stability incident. An important part of this activity was to identify suitable test-tracks, because not all facilities can accommodate this unusual experiment. A fuller version of this discussion was delivered in a separate report to the NTRCI [5].

CIRCUMSTANCES SURROUNDING ROLLOVERS

Most experimental heavy vehicle rollover research involves untripped rollovers. There are many good reasons for this, one of which is that the outcomes are simpler to predict. This means they are safer to conduct and the models are easier to verify. Since untripped rollovers depend on fewer outside forces, they better reflect the vehicle's basic stability, and therefore an improvement in the vehicle's stability will benefit both tripped and untripped rollovers. Untripped rollovers, however, are not the most common kind of heavy vehicle rollover. A recent report on cargo tank rollovers for the FMCSA [2] indicated that only 14 percent of cargo tank rollovers are untripped. This is shown in figure 10 which shows cargo tank crash statistics and the types of crashes in which rollovers occur. Note that more than half of the rollovers were associated with a run-off-the-road incident. Figure 11 indicates the distributions of locations where cargo tank rollovers occur. One-third of rollovers of cargo tank trucks carrying a hazardous material occur on an undivided highway, away from an intersection. Information from a different crash database indicates that the conventional "too fast in a curve," phenomenon is not the sole reason for cargo tank rollovers.

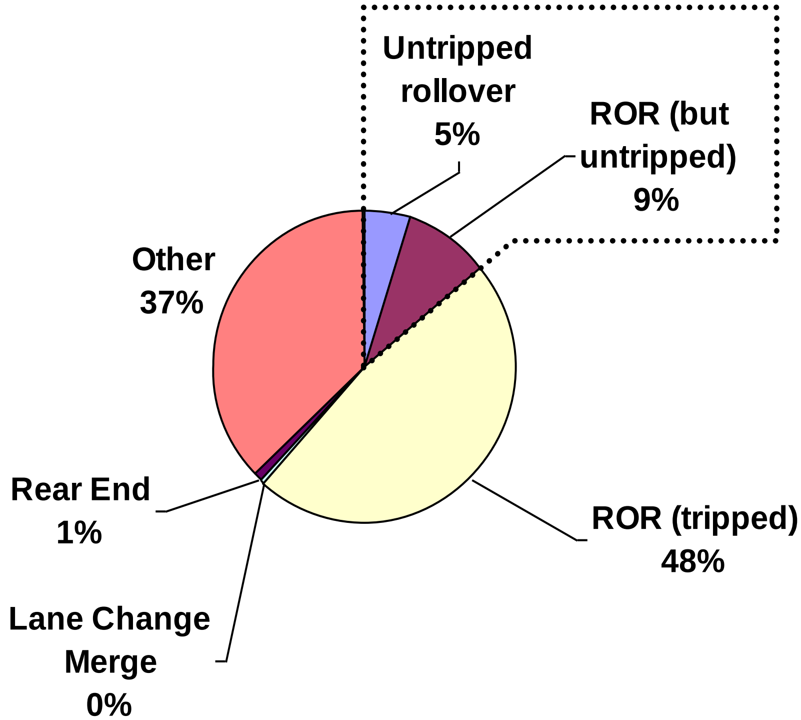


Figure 10. Pie Chart. Cargo Tank Crash Statistics (Adapted from Figure 2-8 Of Pape et al. [2], based on data in the General Estimates System).

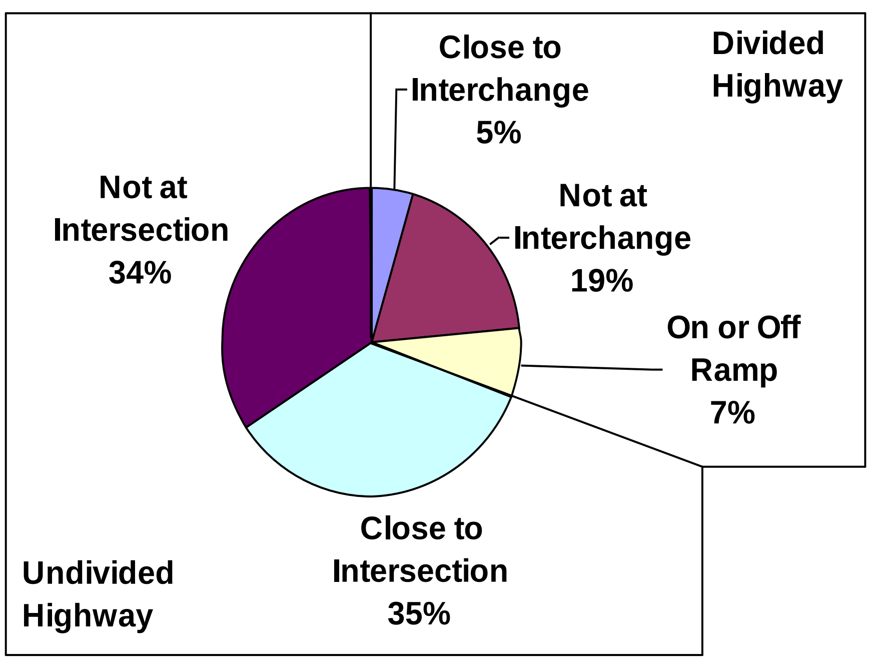


Figure 11. Pie Chart. Distributions of Locations Where Cargo Tank Rollovers Occur (Adapted from Table 2-38 of Pape et al. [2], based on data in the Motor Carrier Management Information System).

A vehicle may drive off the road inadvertently because the driver was distracted, or it may leave the road because of an avoidance maneuver, among other reasons [6.] Several scenarios may lead to a rollover following a roadway departure. A severe drop-off, such as that shown in figure 12 may lead directly to the rollover. Such drop-offs are not standard in finished construction, though drop-offs of several inches may occur temporarily in construction zones where traffic is maintained. When a vehicle strays far from the traveled way, it may come to an embankment. A steep embankment can trip even the most stable vehicle, but top-heavy commercial vehicles with high CGs are more subject to rollover even at more modest slopes. A rollover can occur at a very low speed when the tires on one side of a truck drive off the pavement into soft soil (in an untripped rollover, the truck rolls toward the outside of a curve, where it is pulled by centrifugal force. In contrast, combination vehicles have rolled to the inside of a turn at highway intersections when the trailer tires leave the pavement and fall into soft soil). The most common reason for a truck rolling over following a roadway departure may not be the departure itself, but the recovery maneuver. Research with light vehicles has documented several ways this can happen. If a heavy vehicle suddenly mounts the drop-off and is on the pavement with a high curb angle (where the vehicle is pointed across the pavement, but is still traveling more-or-less along the pavement), it can roll over. This is why carriers train their drivers in such a way that if they find themselves off the road, they should slow to a specified low speed before attempting to return to the pavement. Drivers with any doubt that a successful recovery is possible should stop and call for help so they can be towed back to the pavement.

Figure 12. Photograph. A Vertical Pavement Edge with an Unusually High Drop-Off (from Wagner and Moler [7]).

Pavement drop-offs were a topic of experimental research in the late 1970’s and the early 1980’s. The experimental studies were carried out with vehicles that were popular at the time, and with light vehicles; only two studies included a vehicle as large as a pickup.

In a joint report from the FHWA and the AAA Foundation for Highway Safety, Hallmark et al. [8] thoroughly examined the problem of pavement edge drop-offs. The report summarizes previous light-vehicle research on run-off-the-road recoveries, and it surveys state practices and roadway conditions concerning drop-offs. The report describes the “Safety Edge” which is a beveled edge to the pavement, not more than 30 deg to the horizontal (see figure 13 and figure 14). Several states have adopted the Safety Edge in their construction specifications. An ongoing three-year study [10] is examining the benefits of the Safety Edge by comparing the run-off-the-road-related crash rates on selected rural roads that have been resurfaced with and without the Safety Edge.

DESCRIPTION OF RECOMMENDED EXPERIMENTS

This section describes a series of maneuvers that will continue the prior research with light vehicles and meet the needs of the FHWA. The set of maneuvers was discussed with representatives from several test-tracks. The plan that will be described is one that they, in principle, agreed that they could conduct. The maneuvers have been developed in enough detail so that the test-track representatives can understand what the NTRCI team would like to do, and to begin a discussion within the NTRCI team about potential future efforts. It is not a detailed test plan, and many decisions remain to be made. The description focuses on tripped rollover maneuvers; a series of conventional tests will be essential to quantifying the inherent stability of the vehicle in these experiments.

Figure 13. Photograph. Safety Edge (From FHWA [9]).

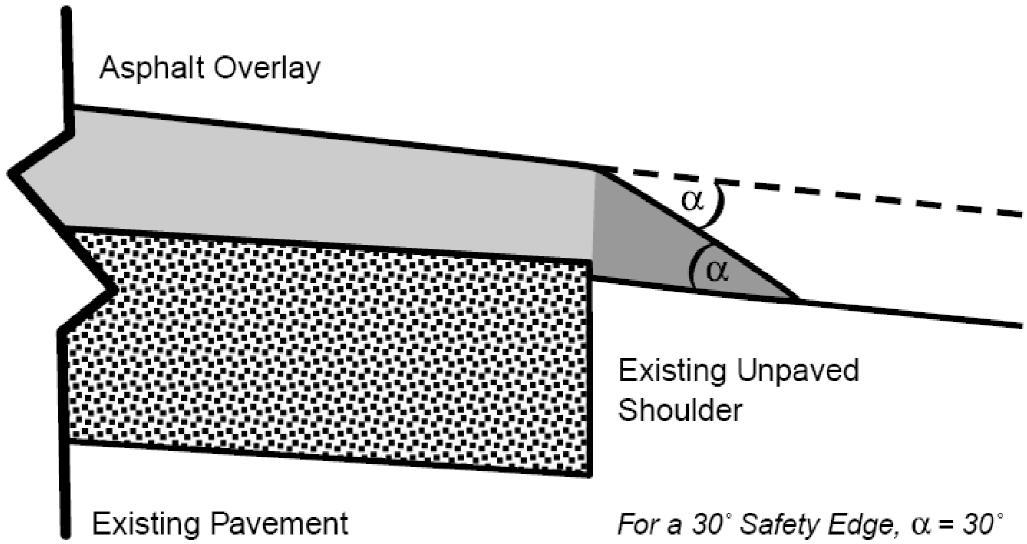


Figure 14. Illustration. Schematic Drawing of the Safety Edge (from Hallmark et al. [8]).

The maneuver to be conducted in the proposed research is illustrated conceptually in figure 15. Three moments in time are shown in the figure, and portions of the vehicle are viewed from three directions in respective portions of the figure. The maneuver begins with the truck running up to speed in a straight line. The driver takes the truck along the edge of the pavement, and then the right tires descend a shallow ramp, as shown at position 1 in figure 15. This establishes a stable, straight path with the left tires on the pavement and the right tires at a slightly lower level, on the shoulder. This is position 2 in figure 15. Cones or pavement markings will indicate to the driver how far the right tires are to be from the pavement edge. The driver will then steer at a pre-determined angle toward the pavement edge. The front right steer tire will climb the pavement edge (position 3 in figure 15), with the rest of the vehicle soon following. The driver will then slow the vehicle and steer back parallel with the pavement. In the first runs, the speed and path will allow plenty of width and remaining length of the pavement for the driver to safely bring the vehicle to a stop. Then the driver will be instructed to repeat the maneuver but attempting to steer so that the vehicle would remain within its original travel lane following the recovery. After the ability to accomplish this safely has been established, parameters will be varied (such as achieving a more shallow scrub angle) to make the recovery maneuver more difficult.

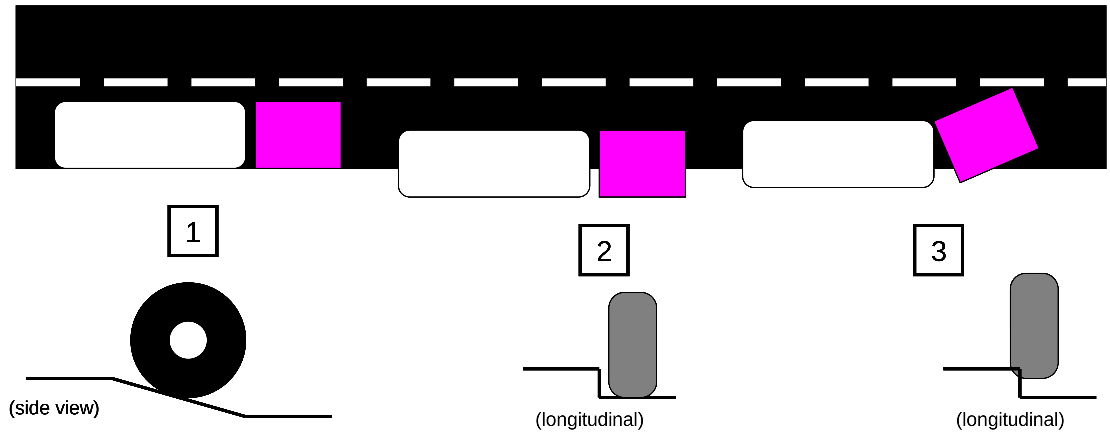


Figure 15. Illustration. Time Sequence for a Suggested Tripped Rollover Test Maneuver Showing Vehicle and Tire Locations.

Controlled inputs to the experiment will be the speed, the geometry of the drop-off, and the approach angle; as well as possible variations in the vehicle itself. Measured outputs will be the set of vehicle dynamics data, similar to what has been recorded in the Phases 1, 2 and A of the Heavy Truck Rollover Characterization Research Program. The most important outcome of such an experiment will be whether the recovery was “successful;” that is, whether the vehicle remained upright (as opposed to experiencing the touch down of the outriggers) and is able to return to its original travel lane (within 12-to-14 ft of the drop-off).

Table 2 lists the parameters that might be varied during the experiment. As with all experiments, budget trade-offs and compromises will be necessary in the selection of the most important parameters and their values. The parameters are listed in the table roughly in a decreasing order of importance, with the understanding that other researchers from other perspectives may ascribe different relative importance. The first three, speed and the two parameters related to edge geometry, are essential for characterizing behavior in drop-off recovery. Vehicle-dependent parameters such as tire selection and suspension geometry coordinate well with ongoing NTRCI rollover research. The respective merits of the parameters are discussed within the table.

Table 2. Parameters in a Pavement Edge Drop-Off Experiment

	Variable	Range	Comments
1	Speed	Up to 70 mph	Ideally, the test speeds would range up to 70 mph, to span the range of possible highway speeds. Some distance (roughly a mile) would be required for a heavy truck to reach this speed. Research with passenger cars found that a high (4.5 in) edge is potentially unsafe even at 35 mph, so only the mildest edges will be attempted at higher speeds. The selection of the maximum test speed should be made after simulation of the recovery maneuvers, and the test driver should be given the authority to halt testing at any time that a maximum safe speed appears to have been reached.
2	Edge Height	1, 3, and 4 in	A heavy truck should not have difficulty mounting a 1 in step. The present limit for a vertical edge for passenger cars is 4 in so there is no need to plan for a higher edge. An intermediate height is a desirable experiment if resources permit.
3	Edge Slope	Vertical, 35 deg to the horizontal	The vertical edge is the most difficult to mount. Due to factors such as erosion and temporary construction conditions, a vertical edge is plausible on highways. The "Safety Edge" of 35 deg should be part of the experiments because it is the current recommendation.
4	Approach Angle	Scrubbing, oblique	Scrubbing is a most dangerous phenomenon; a few low-speed experiments should be attempted to document the results and to develop training materials. Expert opinion, and to a lesser extent, simulations, will establish a minimum safe oblique angle to begin the experiments
5	Approach Direction	Dropping off, recovering	This step involves a recovery where a vehicle attempts to climb a drop-off, following a departure. This is the primary interest expressed by the FHWA. The behavior of the vehicle following a sudden drop-off would be of interest to vehicle designers.
6	Stability Control	None, full roll and yaw stability	A pavement drop-off and subsequent remount are essentially a double lane-change maneuver across a pavement discontinuity. Modern electronic stability aids have demonstrated a benefit in double lane-change maneuvers on level pavement (both dry and slippery). This test will determine the extent to which they can benefit drop-off recovery maneuvers as well..
7	Tire	Dual tires, NGSWBTs	The tire style is not an inherent part of the drop-off experiment, but it fits well with the other NTRCI experiments; and the drop-off would provide a valuable challenge to the tires.

	Variable	Range	Comments
8	Vehicle Configuration	Single vs. combination unit	The effect of a semi-trailer or a pull trailer on the vehicle behavior is difficult to predict.
9	Vehicle	Suspension, track width, other	The NTRCI and HTRC are considering innovative suspension designs that may be suitable for experimentation, either with standard maneuvers or with these drop-off tests. The experiments should be coordinated with ongoing work by tank manufacturers to improve the stability of their products. If the design of the vehicle becomes an important parameter in the experiments, tilt-table measurements of the various vehicles would be beneficial.
10	Payload CG Height	Low, high	A substantial change in the vehicle configuration, such as the payload CG height would be useful to simulations, after the experiments. If the experiments are with a cargo tank vehicle, changing the payload height is much more difficult than with dry freight using concrete ballast. Limits of safe maneuvering that are established with a cargo tank vehicle, which have an inherently high-CG, would be conservative for application to other, more stable vehicles. Unless changes in the height are associated with fundamental changes in the vehicle design, this parameter is a low priority.
11	Shoulder	Hard (concrete), soft (gravel)	When an overlay is put on a highway, the original pavement may provide a hard surface adjacent to the drop-off. Probably a more common condition is a gravel or soil surface immediately beyond the drop-off, as would be the result of erosion. The surface condition beyond the drop-off is probably a bigger handling difference to the handling before re-mounting than after remounting. If the experiment focuses on the post-mounting recovery maneuver, the surface condition beyond the drop-off is likely of little consequence.

Two test-tracks, the Nevada Automotive Test Center, and the Smithers Winter Test Center, have communicated to members of the HTRC that they can support the kind of testing discussed here. Other test centers agreed, in principle, to the nature of this testing, but do not have the room to permit the truck to accelerate to 70 mph with adequate recovery space. They can be contacted again if the first series of experiments are to be conducted at lower speeds.

The edge heights and geometries will be limited mostly to those that are considered to be within the current guidelines and best practices. There is no reason to expect that experiments with a heavy vehicle would lead to a relaxing of the standards that must apply to light vehicles as well. A small number of cases may be run with a geometry that is outside of the guidelines (for example, a vertical 4 in drop) to avoid the need to extrapolate a trend or to provide comparison with prior data, but concern for safety will limit those cases.

If only one pavement edge treatment is tried, it should be the "Safety Edge," a drop of less than 4 in with a slope of 30-to-35 deg to the horizontal; which is the current FHWA recommendation [7]. As such, a maximum speed or some minimum set of conditions for a heavy vehicle's re-mounting can be established for this benchmark. Because the edge geometry that is safe for a heavy vehicle may be shallower than that for a light vehicle, shorter and more gently sloping edges should be subsequently tried.

Experiments with tank trucks always raise the question of sloshing of a partial load or of a full load with intentional outage. Significant sloshing may diminish the safety of a maneuver and complicates the analysis, so partial loads are not listed among the possible experimental parameters. Moreover, most carriers avoid partial loads when possible. The tank for the experiments will most likely be filled with water. Because water is denser than petroleum distillates, a tanker designed for this cargo (i.e. a DOT 406 or oval cross-section tank) would reach its legal gross weight with a partial load of water. It may be preferable to use a tank intended for acid (e.g. a DOT 412 or circular cross section). Such a tank could be filled with water and have a gross weight of only slightly less than it would have in normal service. An alternative is to put foam or bladders in a tank of oval cross section to achieve a representative mass distribution without a moving liquid.

Trainers of professional tank truck drivers should be consulted in developing the test plan. The first maneuvers to be attempted for mounting the pavement edge should be at speeds and angles of approach that are accepted by the carriers as being safe. Having verified that under controlled conditions that those instructions are indeed safe, then the experiments can then proceed to higher speeds, other angles of attack, and other post-remount steering patterns.

A successful recovery maneuver will depend strongly on the skill of the driver. A computer driven truck may be able to accelerate and begin the road departure, but the counter steering following the re-mounting of the edge would be quite difficult for a driving machine. The best approach will be to have an experienced test driver in the cab, take adequate safety precautions, and rely on the driver's judgments as part of a qualitative data collection effort.

CHAPTER 6. TEST-TRACK TESTING

OVERVIEW OF TEST-TRACK TESTING

The test-track testing of the tractor-flatbed-trailer was accomplished with two different tire configurations (one with NGSWBTs on the tractor-flatbed-trailer, the other with standard dual tires on the tractor-flatbed-trailer). Three different test maneuvers were performed with each tire configuration; two transient and one steady-state. The two transient tests were: 1) a step steer maneuver and 2) a highway evasive maneuver. The steady-state test was a constant radius turn maneuver.

The tires used for the testing were as follows:

- Steer Tires: 275/80R22.5 XZA3
- Standard Drive Axle Tires: 275/80R22.5 XDA Energy
- NGSWBT Drive Axle Tires: 445/50R22.5 X One XDA
- Standard Trailer Tires: 275/80R22.5 XTA Energy
- NGSWBT Trailer Tires: 445/50R22.5 X One XTA

Tire configuration 1**:** NGSWBTs were mounted on the tractor drive axles and flatbed-trailer. The steer axle, drive axle, and flatbed-trailer axle tires for tire configuration 1 were: XZA3, X One XDA, X One XTA, respectively.

Tire configuration 2**:** Conventional dual tires mounted on the tractor drive axles and trailer. The steer axle, drive axle, and flatbed-trailer axle tires for tire configuration 2 were: XZA3, XDA Energy, XTA Energy, respectively.

Testing Attempts

The first three attempts to collect truck rollover data at TRC were unsuccessful. The first attempt (October 25-26, 2007) revealed that when wheel lift was achieved, it was experienced at the drive axles of the tractor. Because the drive axle lifted first, the outriggers mounted in the center of the flatbed-trailer were felt to be inadequate to prevent the tractor from rolling over in the event that a more severe wheel lift were to occur; and the testing was therefore halted. The HTRC subsequently decided that if wheel lift was to be experienced first on the drive axle, then an alternative outrigger placement was necessary in order to assure that the test tractor-flatbed-trailer would not rollover during testing.

For the second attempt (November 16, 2007) the outriggers were moved forward on the flatbed-trailer in order to minimize the potential for the tractor to rollover. However, due to the flexibility inherent in the flatbed-trailer, the outriggers, now located at a more-forward location, and providing more protection for the tractor, did not adequately protect the rear of the flatbed-trailer from rollover. The testing resulted in a larger than desired flatbed-trailer axle lift which was deemed too severe for testing conditions.

For the third attempt (January 10-12, 2008), a second set of outriggers was added to the rear of the flatbed-trailer so that the tractor-flatbed-trailer was protected for both drive axle and flatbed-trailer axle lift. However, during the third attempt at testing, the tractor-flatbed-trailer speed reached relatively high levels without achieving wheel lift. TRC halted further testing because further increases in speed would have resulted in too much energy being in the system, and if a violent wheel lift had occurred, the integrity of the outriggers might have been compromised. After considerable discussion within the team and with subject matter experts outside of the team, it was concluded that in order to achieve a more controllable wheel lift, the CG of the payload had to be raised, allowing for wheel lift to occur at slower speeds. It was also decided that the payload should be grouped into two elements located over the drive axle and the flatbed-trailer axles. In order to accommodate this new payload system, a load rack system was designed to raise the payload ballast blocks above the deck. It should be noted that the load racks would allow the CG of the ballast to be changed which added an additional study variable, and were designed to allow the tests to be conducted within a speed and articulation angle envelope that was acceptable to TRC. The fourth attempt (May 15 – 21, 2008) to perform rollover testing was successful.

The low-speed maneuvers were conducted for the purposes of establishing a baseline assessment for the vehicle, and to generate data of relevance to model validation. The high-speed maneuvers were characterized as speeds at which wheel lift could be achieved for a particular maneuver. The highest speed attained during the May test campaign was less than 40 mph (64.4 km/h). The low-speed maneuvers were executed at several miles per hour slower than the high-speed maneuvers. For the step steer and lane change maneuvers, low-speed was 15 mph (24.1km/h); for the constant radius, the low-speed varied from 5-to-25 mph (8.1-to-40.2 km/h). The vehicle was able to perform all of the maneuvers at low-speed in a very stable manner.

Both the low-CG and high-CG payload conditions were tested using conventional 275/80 R22.5 dual tires on the drive and trailer axles. An additional test with the high-payload CG case only, was evaluated using 445/50 R22.5 NGSWBTs.

Constant Steer Angle Maneuver

This test was conducted to understand the steady-state behavior of the vehicle (understeer, yaw rate gain, lateral acceleration gain, etc.). Multiple low speeds (5 mph, 10 mph, 15 mph, etc.) were addressed (up to 25 mph). At a particular speed, the driver held this speed constant for 60 sec. At this stable condition, the cornering ability of a tractor-flatbed-trailer was quantified by observing the level of lateral acceleration that was sustained for a given steering input. For this maneuver, the driver held the steering wheel at a constant angle which caused the vehicle to travel in a circular motion.

The steady-state test was conducted as follows: while maintaining a constant turn radius, the vehicle's speed was steadily increased until wheel lift occurred. Once wheel lift was achieved, the test was repeated three additional times. This test was performed with the ESC off.

The driver performed this same test with the tractor ESC engaged. For this test, the driver accelerated as he did previously until the tractor ESC engaged.

Step Steer Maneuver

The step steer was selected and executed because it provides information in a repeatable manner on the transient responses of the vehicle to a change in steering input. This information contributes to the understanding of the total vehicle system response. The step steer maneuver used for this test was based on ISO 7401.

The step steer maneuver as executed here was a straight-line test with a single quick right turn input of 170 deg at the steering wheel at a rate of 170 deg/sec. A sufficient test area was required to operate the tractor-flatbed-trailer at a constant speed of 35 mph for approximately 30 sec. Using a bracketing method, the speed corresponding to wheel lift without the tractor ESC engaged was determined. The test was repeated six times at this speed. The tractor ESC was then engaged and the test was repeated until a consistent tractor ESC activation pattern was observed. Two additional runs with the tractor ESC engaged were made at a speed above the wheel lift speed. The vehicle's gross weight for this, and all other tests was very near 80,000 lb

Lane Change Maneuver

This test was conducted to determine the transient response of a vehicle when subjected to a sudden lane change. The resulting data was used to study the transient roll control characteristics of vehicle attitude and rollover. The lane change course was set up with pylons as shown in figure 16. It was set up on a level asphalt area of sufficient size to accommodate safe entry and exit of the course. With the tractor ESC off, the driver entered Gate 1 from the left at constant speed. After exiting Gate 1, the driver made a quick turn to the left and then back to the right in order to line up the vehicle to enter Gate 2. He was asked to maintain a constant speed during the execution of the maneuver. The driver initially made a total of six low-speed runs, negotiating the course at 15 mph. After the six low-speed runs, the speed was increased in increments of 2 mph until the rollover threshold was met (i.e., one or more wheels left the ground during the maneuver). Once the rollover threshold speed had been determined, the test was repeated six times.

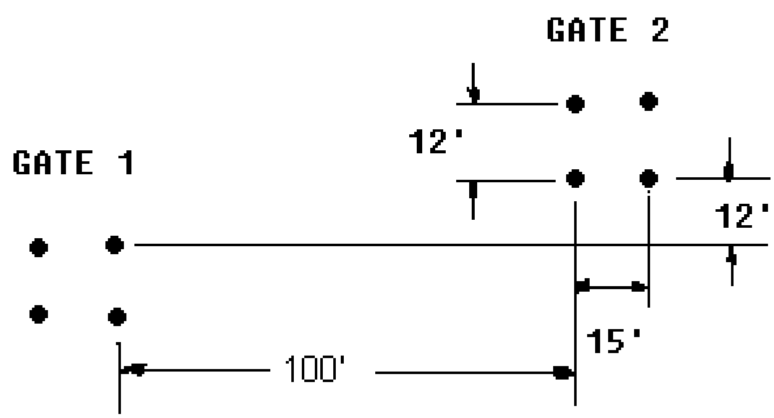


Figure 16. Diagram. Lane Change Maneuver Pylon Layout.

After six runs were completed at the rollover threshold speed, the driver repeated the runs with the tractor ESC engaged. This continued until a consistent tractor ESC activation pattern was observed. At least two additional runs were made with the ESC engaged at a speed above the wheel lift speed. The vehicle's gross weight was very near 80,000 lb.

CHAPTER 7. K&C TESTING

TESTING OVERVIEW

MARC performed K&C testing on a 2007 Volvo VT64T830 6X4 truck (VIN 4V4LC9KJX7N481567) and on a 2008 Utility Flatbed Trailer (VIN 1UYFS24808A291803). The testing was performed on the MARC K&C Test Rig which allows complete tractor K&C testing on a over-the-road heavy truck. This test rig is located at the MARC Research and Development Center in Greenville, South Carolina.

Both the truck and the flatbed-trailer were tested in a condition as close to the "as-tested at TRC" configuration as possible.

The tractor was tested with a full load of fuel. It was fitted with 275/80R22.5 Michelin XZA3 tires on the front axle and 445/50R22.5 Michelin X One XDA tires on the middle and the rear axles.

A weight rack with weights having a combined weight of 9,524 Kg was fitted to the 5th wheel. This was accomplished by engagement of a kingpin mounted on the bottom of the weight rack.

Fixtures on the bottom of the weight rack prevent pitching and rotation of the weight rack while mounted to the 5th wheel.

The tractor was then placed onto the test rig.

Each road wheel sits on a scale platform which measures the vertical load as well as allowing low friction movement in the X-Y plane.

A wheel plane transducer, used to track to motion of the wheel, was fitted to each wheel position and adjusted to eliminate runout and to ensure concentricity. Similarly two body transducers were fitted to the chassis of the tractor; one for the front of the tractor and the other for the rear.

Figure 17. Photograph. Loaded Weight Rack before Fitting to the Tractor's 5th wheel.

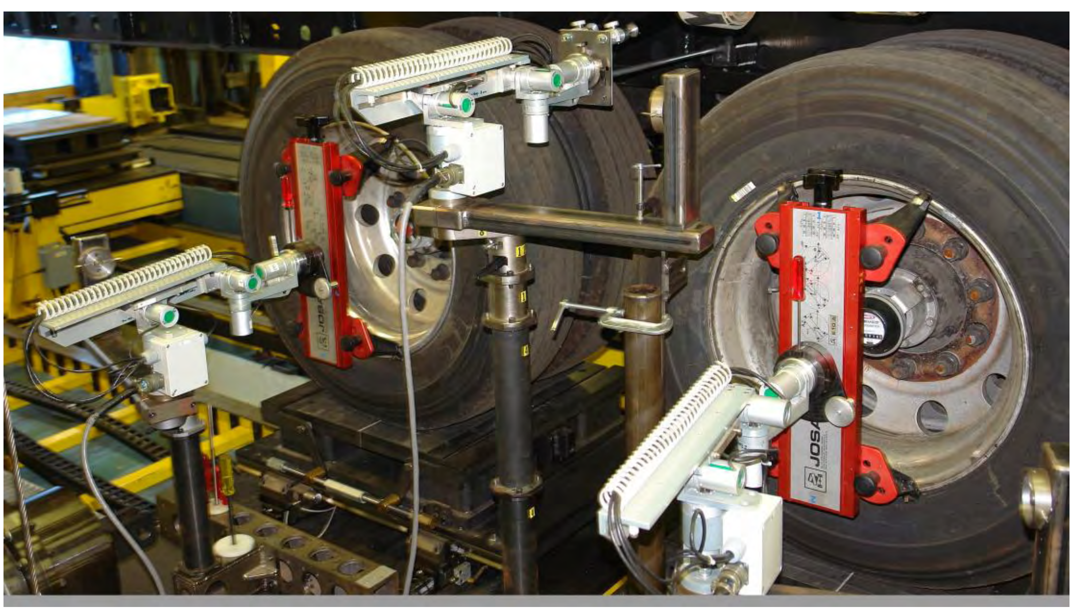


Figure 18. Photograph. Wheel Plane Transducers and Body Transducer.

A hydraulic pump was plumbed in the power steering system so that the steering system could be energized during testing without running the engine.

An instrumented steering wheel was fitted to the tractor and measured steering wheel angle and steering wheel torque.

TRACTOR TESTING

The following tests were conducted on the Volvo VT64T830 tractor:

- Vertical deflection tests for the front, middle and rear axles,
- Longitudinal compliance tests in the braking direction for the drive, middle and rear axles,
- Longitudinal compliance tests in the traction direction for the drive, middle and rear axles,
- Lateral compliance tests on each wheel position individually as well as each axle with the lateral forces applied "on-center" and applied 30 mm behind,
- Roll kinematics testing,
- Steering system property assessment,
- Front axle steering geometry assessment,
- Chassis torsional stiffness assessment, and
- Shock absorber characteristics assessment.

Figure 19. Photograph. The Volvo VT64T830 Tractor on the MARC K&C Rig.

Test specific information is as follows:

Front Axle Vertical Deflection Test

Under the tractor, a square-threaded linear activator was connected to the cross frame connector in a location in front of the motor. This connection is in the center of the tractor in the lateral direction. This test uses the wheel position sensors connected to each of the front axle wheels and the front body position transducer. Data from the front axle wheel scales are recorded as well.

The test rig then pulls the tractor down until the wheel load reached 0.5 g higher than the pre-test value. At this point the test rig reverses direction and raises the vehicle up until the wheel load becomes 0.5 g lower than the pre-test value. Again the rig changes direction and begins to pull the truck down. This motion is repeated for three complete cycles. Only data from the last two cycles are recorded and used in further processing.

Middle and Rear Axle Vertical Deflection Test

The middle and rear axles are tested together. The square-threaded linear activator was positioned between the middle and the rear axles, and connected to a beam. The length of this beam is in the lateral direction beneath the tractor, parallel to the axles. The beam is sufficiently long to extend outward past the frame of the tractor. Links are used to connect the ends of this beam to the weight rack so the pull-down/lift-up forces are applied to the tractor through the 5^{th} wheel.

This test uses the wheel position sensors connected to each wheel of the middle and the rear axle, as well as the rear body position transducer. Data from the middle and rear axle wheel scales are also recorded.

The vehicle's brakes are applied during testing and the air suspension leveling system has been disconnected and locked in the full-laden curb condition. The air bags were charged with the air pressure found to be present at the governed ride height before testing began. During testing, the air system was isolated from the air supply.

The test rig pulled the vehicle down until the wheel loads reached 0.5 g higher than the pre-test value. At this point the test rig reverses direction and raises the vehicle up until the wheel loads become 0.5 g lower than the pre-test value. Again the rig changes direction and begins to pull the truck down. This motion is repeated for three complete cycles. Only data from the last two cycles are recorded and used in further processing.

Compliance Tests

All compliance tests were performed by pulling the ground out from beneath the tires, i.e., the low-friction platforms located between the tire and the scales are pulled out. The limit for the pull force for these tests is 0.4 g of the pre-test static weight. Pulling beyond this limit often results in sliding between the tire and the safety-walk covered platform.

Longitudinal Compliance Tests

Front Axle Braking Compliance Test: A pneumatic cylinder at the rear of the tractor is connected to a balance beam which is in turn connected to each of the front wheel platforms with cables. Load cells are placed in each of the cables so that the force that the ground is being pulled out from under the tire can be measured. The tractor chassis is connected rigidly to a beam at the front bumper and the weight rack is connected to a restraint apparatus in the rear. The tractor chassis is held so that the individual wheel loads are the same as in the unrestrained condition. The longitudinal restraint in the rear of the tractor is positioned such that the line of action of the restraining force passes horizontally through the pitch axis pin of the 5^{th} wheel, eliminating pitching of the weight rack and the corresponding weight change on the rear axles. This test uses the wheel position sensors connected to each of the front axle wheels and the front body position transducer. Data from the front axle wheel scales are recorded as well. The vehicles brakes are engaged during this test.

The pneumatic cylinder then pulls the wheel platform until the load of the in-line load cells reaches 0.4 g higher than the pre-test value. At this point the pneumatic cylinder reduces the load to zero. This motion is repeated for three complete cycles. Only data from the last two cycles are recorded and used in further processing.

Plots were produced of the vertical wheel load change, toe change and steer, wheel center longitudinal displacement, locked wheel rotation and torque reaction at the handwheel as a function of the longitudinal load.

Middle Axle Braking Compliance Test: The same procedure as that used for the front axle compliance test (described above) is used here with the exception that the rear body and the middle axle wheel sensors are used. The in-line load cells and cables are connected to the platforms of the middle axle. The air suspension system was isolated from the air supply during testing in the same manner as in the vertical testing.

Plots were produced of the vertical wheel load change, toe change and steer, wheel center longitudinal displacement and locked wheel rotation as a function of the longitudinal load

Rear Axle Braking Compliance Test: The same procedure as that used for the front axle compliance test (described above) is used here with the exception that the rear body and the rear axle wheel sensors are used. The in-line load cells and cables are connected to the platforms of the rear axle. The air suspension system was isolated from the air supply during testing in the same manner as in the vertical testing.

Plots were produced of the vertical wheel load change, toe change and steer, wheel center longitudinal displacement and locked wheel rotation as a function of the longitudinal load.

Middle Axle Traction Compliance Test: This test is similar to the middle axle braking compliance test (described above) except that the pneumatic cylinder was moved to the front of the tractor and attempts to pull the platforms forward from beneath the tires. The tractor’s brakes are not engaged during this test. A driveline lock was installed onto the engine’s flywheel to prevent engine crankshaft rotation.

The air suspension system was isolated from the air supply during testing in the same manner as in the vertical testing.

Plots were produced of the vertical wheel load change, toe change and steer, wheel center longitudinal displacement and locked wheel rotation as a function of the longitudinal load.

Rear Axle Traction Compliance Test: This test is similar to the middle axle braking compliance braking test (described above) except that the rear body and the rear axle wheel sensors are used. The in-line load cells and cables are connected to the platforms of the rear axle. The air suspension system is isolated from the air supply during testing in the same manner as in the vertical testing.

Plots were produced of the vertical wheel load change, toe change and steer, wheel center longitudinal displacement and locked wheel rotation as a function of the longitudinal load.

Middle Axle Overrun Compliance Test: This test is similar to the middle axle traction compliance test (described above) except that the pneumatic cylinder was moved to the rear of the tractor and attempts to pull the platforms rearward from beneath the tires. The tractor's brakes were not engaged during this test. A driveline lock was installed onto the engine's flywheel to prevent engine crankshaft rotation. The air suspension system was isolated from the air supply during testing in the same manner as in the vertical testing.

Plots were produced of the vertical wheel load change, toe change and steer, wheel center longitudinal displacement and locked wheel rotation as a function of the longitudinal load.

Rear Axle Overrun Compliance Test: This test was similar to the middle axle overrun compliance test (described above) except that the rear body and the rear axle wheel sensors are used. The in-line load cells and cables were connected to the platforms of the rear axle. The air suspension system was isolated from the air supply during testing in the same manner as in the vertical testing.

Plots were produced of the vertical wheel load change, toe change and steer, wheel center longitudinal displacement and locked wheel rotation as a function of the longitudinal load.

Lateral Compliance Tests

For all lateral compliance tests the wheel platforms (ground) are pulled in the lateral direction by a pneumatic cylinder. Load cells are placed between the cylinders and the platforms to measure the applied lateral forces. The movement of the wheel is measured by the wheel position sensor, and the body movement is measured by its body position sensor. The vehicle is restrained at the correct ride height by a beam connected to the front bumper area in the front, and by the weight rack in the rear.

This test was performed on an axle as well as on each wheel position individually. All axle testing was performed in-phase; no out-of-phase lateral compliance tests were performed.
In order to generate aligning moment data, all lateral tests were performed twice. The first test has the lateral force applied directly beneath the centerline of the axle. The test was then

repeated with the point of application moved rearward, behind the centerline of the axle by 30 mm.

Front Axle Lateral Compliance Tests

A test suite of lateral compliance tests was performed on the front axle as described in the paragraph above (Lateral Compliance Tests).

Plots were produced that included the wheel load change, toe change, steering rack corrected toe change, applied moment toe change, rack corrected applied moment toe change, steering rack displacement, camber change, track change, wheel center lateral displacement, and the torque reaction at the handwheel.

Middle Axle Lateral Compliance Tests

A test suite of lateral compliance tests was performed on the middle axle as described in the section entitled "Lateral Compliance Tests."

Plots were produced that included the wheel load change, toe change, applied moment toe change, camber change, track change and wheel center lateral displacement.

Rear Axle Lateral Compliance Tests

A test suite of lateral compliance tests was performed on the rear axle as described in the section entitled "Lateral Compliance Tests."

Plots were produced that included wheel load change, toe change, applied moment toe change, camber change, track change and wheel center lateral displacement.

Vehicle Roll Characteristics Test

Two roll beams were fastened to the tractor. One roll beam was bolted in place of the front bumper while the rear roll beam was bolted to the weight rack. The tractor was otherwise unrestrained. The beams are cable driven and impart a pure moment into the chassis. The tractor was free to roll about its true roll axis. All wheel loads and displacement transducers were used for this test as well as both body transducers. The two roll beams roll in-phase.

The following plots were generated from this test:

- Front wheel loads vs. suspension deflection,
- Front toe change vs. roll angle,
- Front toe change vs. roll angle (rack corrected),
- Front toe change (contact patch yaw corrected),
- Front camber vs. roll angle (ground relative),
- Handwheel torque vs. roll angle,
- Rear wheel loads vs. suspension deflection,
- Rear toe change vs. roll angle,
- Rear toe change (contact patch yaw corrected),
- Rear camber vs. roll angle (ground relative),
- Total roll moment vs. roll angle,

- Front and rear roll moment vs. roll angle,
- Front and rear wheel loads vs. applied moment,
- Static roll weight transfer coefficient vs. applied moment,
- Load biased roll steer vs. roll angle,
- Roll stiffness distribution vs. roll angle,
- Front and rear loads vs. roll angle, and
- Static roll weight transfer coefficient vs. roll angle

Steering System Characteristics Tests

A number of steering tests were performed on the tractor. From these tests the following plots were generated:

- Steering ratio,
- Ackermann,
- Ackermann deviation,
- Steer torque lock-to-lock,
- "Center" steer torque (wheels on air plates),
- Steering flexibility (both wheels),
- Steering flexibility (left wheel only),
- Steering flexibility (right wheel only),
- Steering column flexibility (both wheels),
- Steering column flexibility (left wheel only), and
- Steering column flexibility (right wheel only).

Steering Geometry Characteristics Test

A steering geometry test was performed on the steering system. From these tests the following plots were generated:

- King-pin inclination,
- Caster angle,
- Mechanical trail (ground plane),
- Mechanical offset (ground plane),
- Spindle trail (wheel center), and
- Spindle offset (wheel center).

FLATBED TRAILER TESTING

The Utility flatbed-trailer was backed into the test rig and a K&C suite of tests was performed on it. For the vertical tests, and for all of the compliance testing, the tractor was used to support the kingpin. For the roll testing and the torsional testing, the tractor was replaced with a testing beam at the kingpin so that the flatbed-trailer had the necessary degree of freedom in the roll direction.

The software used to operate the test rig and to analyze the data was developed to process two- and three-axle vehicles. It assumes that the vehicle has a front axle, maybe a middle axle and a rear axle. For all of the flatbed-trailer testing and reporting discussed here the middle axle name will refer to the leading flatbed-trailer axle, while the rear axle name will refer to the trailing flatbed-trailer axle.

Figure 20. Photograph. Flatbed-Trailer on the MARC K&C Test Rig.

Middle and Rear Axle Vertical Deflection Test

This test was performed and reported in the same manner as was done for the tractor.

Compliance Tests

These tests were performed and reported in the same manner as was done for the tractor. Traction and overrun tests were not performed on non-driven axles.

Vehicle Roll Characteristics Test

A roll test was performed in a similar manner as was done for the tractor except that there were no front axle results.

CHAPTER 8. TORSIONAL STIFFNESS TESTING

Torsional stiffness testing was conducted by both MARC and WMU. MARC's efforts were accomplished on their K&C rig. WMU developed a field-based procedure for acquiring this data. Both the MARC and WMU testing will be described.

MARC TORSIONAL STIFFNESS TEST PROCEDURE DEVELOPMENT

Two torsional tests were developed by MARC.

The tractor test method was designed to test the torsional properties of the tractor chassis. This test used the tractor K&C rig to apply twisting moments to the tractor and to measure the loads and resulting frame twist angle needed to calculate its torsional stiffness. During the torsional stiffness test, the front and rear roll beams were driven out-of-phase resulting in a twisting of the tractor frame along the X-axis. The front body displacement transducer was connected to the tractor frame immediately rearward of the front suspension, and the rear body displacement transducer was connected to the frame immediately forward of the middle axle suspension. These transducers were used only for the roll angle which is valid only for the Y-Z plane at the position the sensor is mounted. Load cells were installed in the cables which activate the roll beams so that a torque can be calculated. All wheel position scales were recorded along with the Y-location of the wheel centers to calculate the reactions at the wheels.

Summing the torques at the front half of the tractor with those from the rear half of the tractor yielded the torque thru the center section of the frame, between the two roll angle planes. The gradient of the plot of the torque across the center section of the tractor frame versus the angle of twist between the measurement planes, describes the torsional stiffness.

The flatbed-trailer torsional stiffness measurement was conducted by placing the flatbed-trailer onto the K&C rig. The flatbed-trailer was backed onto the rig so that the flatbed-trailer’s wheels were on scale platforms, as is typical. A beam was designed and constructed so that the flatbed-trailer’s kingpin could be engaged with a 5^{th} wheel, and so the beam could impart to the flatbed-trailer a torque about the X-axis. A surveyor’s precision level was placed on the flatbed-trailer’s deck such that the plane of the level’s sweep was determined by the angle of the deck at a point exactly between the two flatbed-trailer axles (see figure 21). Targets were placed on the deck of the flatbed-trailer such that a surveyor’s precision level could be used to measure the targets deviation in the Z-direction from the level’s optical plane during the twist of the deck (see figure 22).

Figure 21. Photograph. Precision Level and Targets on the Deck of Flatbed-Trailer.

A hydraulic cylinder, acting through a precision load pin, was used to rotate the beam and to generate a known torque input to the flatbed-trailer. The flatbed-trailer was twisted a small amount, and the load pin force and the height of all of the targets were recorded.

Figure 22. Photograph. The Flatbed-Trailer on the Torsional Beam with the Hydraulic Cylinder.

The gradient of the plot of the torque through the trailer versus the angle the trailer was twisted is the torsional stiffness of the trailer.

WMU FLATBED-TRAILER TORSIONAL STIFFNESS TESTING

Objective of the WMU Flatbed-Trailer Torsional Stiffness Efforts

Torsional stiffness of a flatbed-trailer was experimentally determined to assist in developing a field measurement technique that could be utilized to determine this performance characteristic. As part of the Phase-A efforts, a technique was needed that would help in establishing this parameter. The technique helped establish overall torsional stiffness, torsional stiffness to each axle, segmented torsional stiffness and the ratio between the structural torsional stiffness and the combined tire and suspension torsional stiffness.

Introduction to WMU's Torsional Stiffness Efforts

Torsional stiffness of a flatbed-trailer of a tractor-trailer combination, as well as other compliance parameters which link the flatbed-trailer to the tractor have been determined to have the potential for an influence on their rollover threshold. The flatbed-trailer, which links to the tractor through the kingpin, obtains part of its torsional rigidity from the tractor suspension. Stiffness characteristics of the tractor suspension, flatbed-trailer structure, flatbed-trailer suspension, as well as load placement and load distribution affect this threshold.

Improvement of the rollover threshold has the possibility of reducing tractor-trailer rollovers and their resulting fatalities. The flatbed-trailer type was assumed to be one which has the lowest structural torsional stiffness, while the tanker-trailer type has the possibility of having the highest torsional stiffness characteristic.

WMU Torsional Stiffness Procedure

A 48 ft flatbed-trailer was used in this study. The flatbed-trailer was equipped with an air suspension with the air bags supporting the suspension through the trailing arms at a slightly greater than a 1-to-1 mechanical advantage (the air bags were aft of the axle centerline on trailing arms that were pivoted forward of the axle).

An 8 in section I-beam with a half-inch web and flange thickness was used as the member through which the torque could be applied. Two 12,500 lb interface load cells were attached to the I-beam with a spacing such that the transverse positioning of the beam would directly impart the applied couple to the longitudinal I-beams of the flatbed-trailer. In field-testing of the loaded flatbed-trailer, the 12,500 lb load cells were replaced with 50,000 lb load cells. All other procedures remained the same for all three tests. The load cells were laterally separated by 43 in, while the longitudinal trailer I-beams had a center-to-center nominal distance of 44 in. The I-beam was positioned just forward of the kingpin on the 5th wheel plate. The I-beam was supported by two heavy duty screw jacks. The location of the screw jacks is shown in figure 23. For the purposes of this test, deflection of the I-beams and stands were considered minimum.

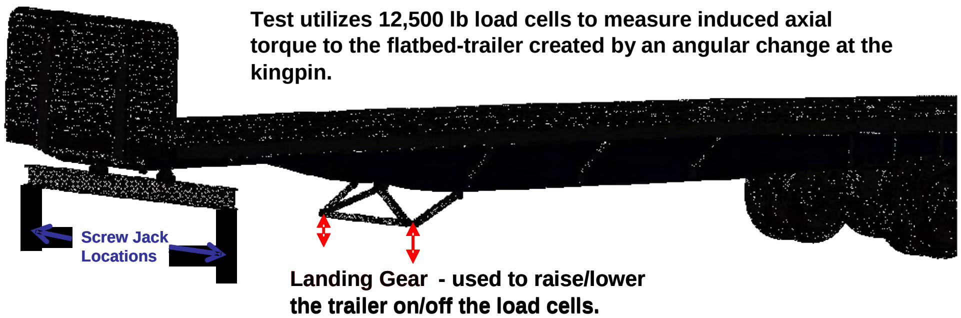


Figure 23. Illustration. Location of Testing Components.

To apply the torsional load, it was determined that using a series of shims positioned between the load cells and the 5th wheel plate would provide the most repeatable measurement (rather than angular measurement). With the exception of the first "verification run," the shims used were a nominal thickness of a 0.245 in. In all of the testing, the flatbed-trailer was lowered onto the load cells using the flatbed-trailer landing gear to minimize the amount of external support equipment required (in the event that this method could be field-useable) as shown in figure 24. Shims were moved from one load cell to the opposite side load cell to impart the torque through the differential height of the shim pack. A maximum of approximately 2 deg of angular twist was imparted to the flatbed-trailer through this method.

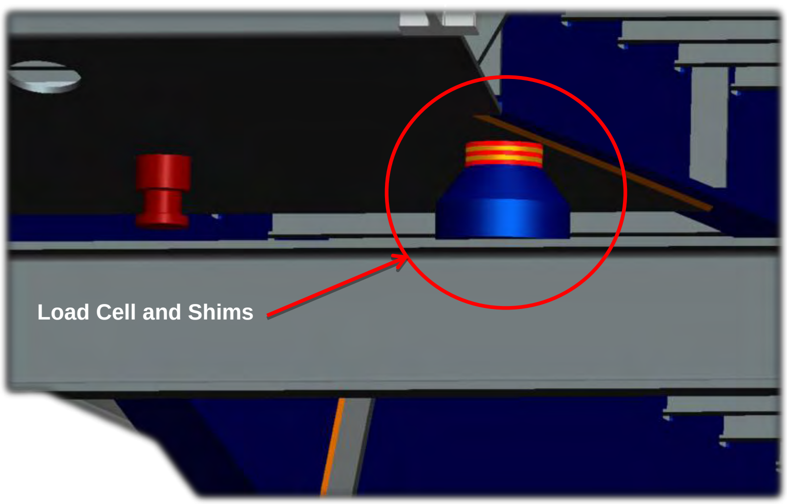


Figure 24. Illustration. Location of Load Cells and Shims.

In an effort to identify specific areas with the lowest torsional stiffness, the flatbed-trailer was divided into discrete sections to allow angular measurement change for each section as the shim packs were changed. Angular measurements were taken at seven locations along the length of the flatbed-trailer, and one additional measurement was taken directly from the surface of the 5th wheel plate. It was determined that the most valuable measurements might be accrued if the flatbed-trailer were divided into sections that provide discrete transitions. These locations are defined and shown in table 3 and figure 25, respectively.

Table 3. Position Identification

Reference Dimensions	
P-0	On top of the flatbed-trailer bed at the front-most point.
P-1	On top of the flatbed-trailer bed, 55 in aft of the 0.0 reference, 1st transition, start of the transition to the larger I-beam section height.
P-2	On top of the flatbed-trailer bed, 172 in aft of the 0.0 reference, 2nd transition, start of the large I-beam constant section height.
P-3	On top of the flatbed-trailer bed, 308 in aft of the 0.0 reference, start of the transition to the slider area, end of the large section height.
P-4	On top of the flatbed-trailer bed, 361 in aft of the 0.0 reference, start of the slider area.
P-5	On top of the flatbed-trailer bed, 415 in aft of the 0.0 reference, center of the front air bags.
P-6	On top of the flatbed-trailer bed, 464 in aft of the 0.0 reference, location of the 2nd axle air bags.
P-7	On top of the flatbed-trailer bed, 559 in aft of the 0.0 reference, end of the trailer bed.

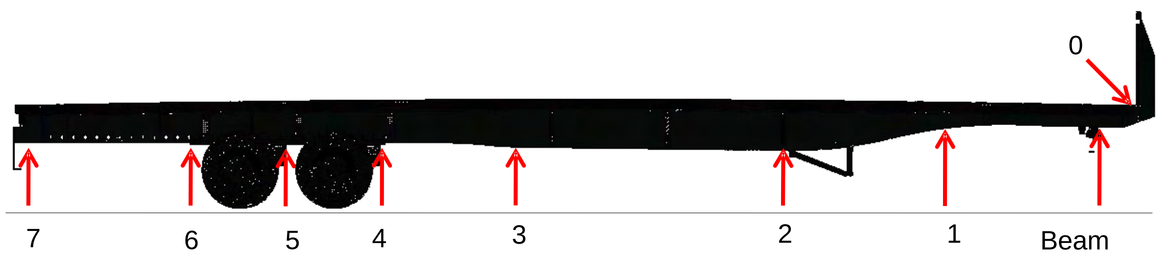


Figure 25. Illustration. Reference Positions for P-1 Through P-7.

In addition to those positions identified in table 3, the aforementioned 5th wheel plate was measured which ideally should respond directly to shim changes. The angular measurements were performed on top of the flatbed-trailer bed with a digital level that had the capacity for measurement to 0.1 deg. An angular measurement was taken at each of the above points for each load case. Once again, this technique was focused on a simple "field-useable" technique. Identifying locations for consistent measurement on the surface proved relatively simple, but introduced some error in the segmented measurements due to the inconsistent nature of the surface. The overall twist introduced by the shim technique provided reliable overall measurements.

Improvements could be made by taking measurements either directly on the frame rails at the discrete locations, and through the use of a more precise electronic angle measurement system. The problem with frame rail measurement is that compromises have to be made in the suspension area as to where a measurement can be made. For this first analysis, simplicity was felt to be the prudent course of action. The torsionally compliant areas of the frame structure were identified. A relatively constant stiffness was found along the length of the trailer axis with this technique.

The next challenge faced was how and where to position the "reaction supports;" those supports against which the torque would react. Functionally, four techniques were employed as outlined in the procedure below.

<u>WMU's Final Field Procedure for Torsional Stiffness Measurement and Summary Results</u>

1. Place the forward axle on jack stands with the rear axle drooped (hanging). There should be no air in the air bags thus forcing the reaction to be directly through the travel stops in the air bags. This then places the reaction points directly into the slider at the position of the front air bags. A dial indicator was positioned across the air bag to assure that compliance in the bump stop did not alter the measurements (deflection of these stops was less than 0.005 in with the torque applied. As a result, this deflection was considered negligible (this may not always be the case with all trailers however)).

   Through this test, an average torsional stiffness between the loading beam and the front air bags of 1,700 ft-lb/deg was obtained.

2. Place the rear axle on jack stands with the front axle drooped (hanging) as shown in figure 26. There should be no air in the air bags thus forcing the reaction to be directly through the travel stops in the air bags. This then places the reaction points directly into the slider at the position of the rear air bags. A dial indicator was positioned across the air bag to assure that compliance in the bump stop did not alter the measurements (deflection of these stops was less than 0.005 in with the torque applied. As a result this deflection was considered negligible as in step 1 (this may not always be the case with all trailers however)).

   Through this test, an average torsional stiffness between the loading beam and the rear air bags of 1,525 ft-lb/deg was obtained. The decreased value in comparison with that obtained in step 1 indicates the effect of the increased length by moving from the front axle to the 2nd axle.

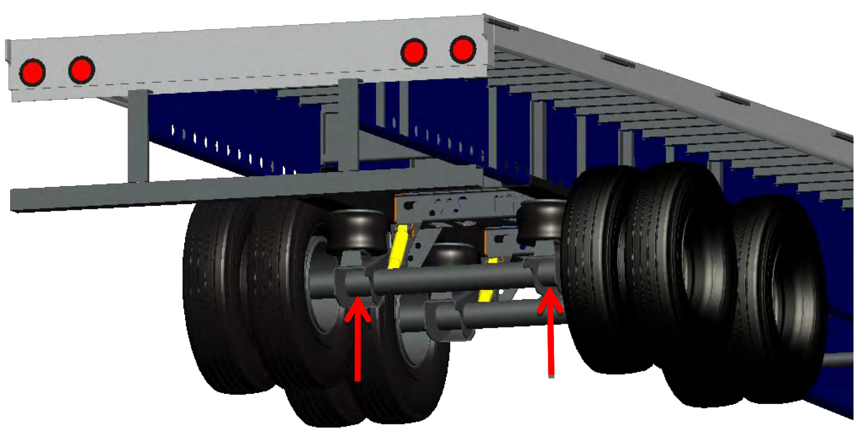


Figure 26. Illustration. Location of Jack Stands for Frame and Axle Contributions.

3. Place heavy duty jack stands under the rear-most section of the frame as shown in figure 27. This effectively eliminates the suspension from coming into play, and should therefore produce the most accurate evaluation of the frame structure. This produced results indicating a nominal stiffness of approximately 1,500 ft-lbs/deg along the flatbed-trailer length, with only a change of approximately 25 ft-lbs/deg. Although in this case there is an increased length, there is also a decrease in path length to the supports because

all suspension links were eliminated.

Through this test, an average torsional stiffness between the loading beam and the rear-most section of the frame, with minimal suspension influence, resulted in a value of 1,500 ft-lb/deg.

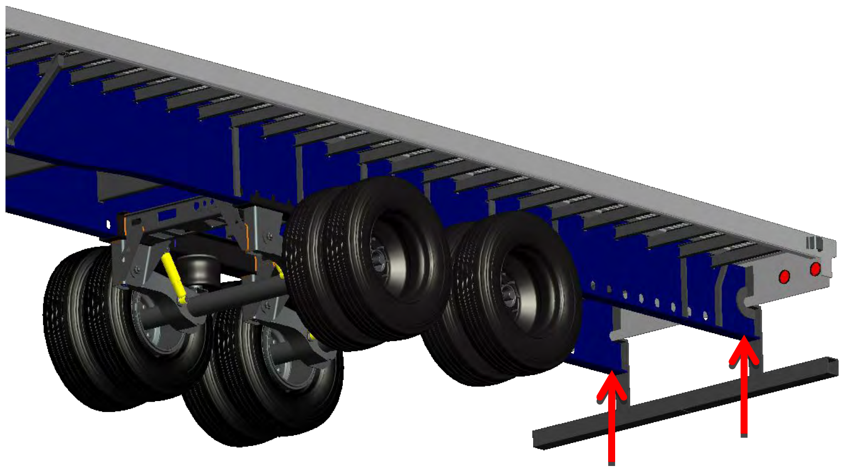


Figure 27. Illustration. Jack Stand Location for Frame-Only Contributions.

4. Place the flatbed-trailer on the floor and air the suspension; therefore bringing the flatbed-trailer suspension characteristics and the tires into play. Equation 1 presents the stiffness relationship for the overall system when the structure, suspension and tires are included.

   Through this test, an average torsional stiffness between the loading beam, acting through the frame, suspension and tires, produced a combined stiffness ($k_{\phi\ overalll}$) of 1,510 ft-lb/deg.

$$\frac{1}{k_{\phi\ overalll}} = \frac{1}{k_{\phi\ structure}} + \frac{1}{k_{\phi\ susp}} + \frac{1}{k_{\phi\ tires}} \qquad \textbf{1}$$

Figure 28. Equation 1.

   For practical purposes it will be assumed that the tires and the suspension can be lumped together reducing the equation 1 to that of equation 2 and equation 3.

$$\frac{1}{k_{\phi\ comb}} = \frac{1}{k_{\phi\ structure}} + \frac{1}{k_{\phi\ susp/tires}} \qquad \textbf{2}$$

Figure 29. Equation 2.

$$\frac{1}{k_{\phi\, susp/tires}} = \frac{1}{k_{\phi\, comb}} - \frac{1}{k_{\phi\, structure}} \qquad \mathbf{3}$$

Figure 30. Equation 3.

Rearranging:

$$k_{\phi\, susp/tires} = \frac{k_{\phi\, comb} \times k_{\phi\, structure}}{k_{\phi\, structure} - k_{\phi\, comb}} \qquad \mathbf{4}$$

Figure 31. Equation 4.

This "theoretically" allows the roll stiffness of the suspension and tires to be determined, at least in a "cursory" fashion, from the previous values. This analysis provided a suspension roll stiffness of approximately 11,000 ft-lb/deg for the suspension system.

5. The stiffness per unit length analysis showed some consistency especially in the areas P-1 to P-3. This technique could be used to determine representative localized stiffness for attention by builders. The values indicated in this technique could be a useful "field" tool.

6. In the final test, the torsional load was incremented downward to evaluate the hysteresis in the system. The hysteresis that was found could be from numerous sources including any combination of the structure and/or suspension. This should be further analyzed in the future.

CHAPTER 9. DATA COLLECTION

The data collection effort for this project was extensive and generated a rich database of diverse information on tractor and flatbed-trailer performance under different maneuvers for different configurations of tires, payload height, and tractor ESC settings. The information that was gathered included not only the variables used in the analyses presented in this report, but also many other important variables that will allow future investigation of other aspects of tractor-trailer performance not covered in this study.

As explained elsewhere in this report (see 0 and 0), two tire configurations were tested for three different maneuvers that were conducted with two different payload heights and with and without the tractor ESC system engaged. Thirty-three different variables, measured through a variety of sensors, were recorded every 0.01 sec and generated 300 megabytes of data. Information was also acquired by videotaping.

The next sub-section presents a summary of the information gathered during these tests, including a detailed description of the data acquisition channels. Subsequently, issues related to the data that was collected and used in the analyses presented in this report are addressed. The final sub-section discusses, in some detail, the steps that were taken to prepare the raw data for the analyses presented in this study.

SUMMARY OF DATA COLLECTED

One of the main objectives of the tests that are part of this study was to understand how different elements (e.g., dual tires and NGSWBTs, payload configurations, etc.) affect the overall vehicle roll stability. Additional goals included acquiring actual rollover data to calibrate and expand simulation models, and determining predictive indicators of roll that can be used to alert drivers and to prevent rollover accidents.

To achieve these objectives and goals, three different maneuvers were designed (see 0) and large amounts of data were collected during the execution of each of them. The tractor and flatbed-trailer used in this study were instrumented with an extensive array of sensors that measured those variables judged as the most relevant in performing the required rollover evaluations and analyses. Table 1, included in 0, presents a detailed list of all of the sensors, deployed for the on-track testing, with a description of each one of the variables measured. The 33 variables were sampled at 102 Hz (close to 0.01-sec intervals), which provided a very high resolution of the information collected.

Notice that in the list of channels presented in Table 1 there is one that is repeated; i.e., TRKXAXIS and TRKXAxis2 both gather vehicle longitudinal acceleration from the same sensor. Since two data acquisition units were used to acquire the information (i.e., unit 1 collected information from roughly the first half of the channels, and unit 2 from the remaining half of the channels), a channel common to both units was used to verify that those units were synchronized; and if not, to synchronize them.

As described elsewhere in this report (see 0 and 0), a combination of two different configurations of tractor-trailer tires, two payload heights, two settings for the tractor ESC system, and two different vehicle speeds were used to test the tractor-flatbed-trailer under three different maneuvers. However, not all of the tests were performed on all of the possible combinations. For example, while combinations that included a high-CG payload were tested with the vehicle equipped with both dual tires and NGSWBTs, the low-CG payload combinations were only tested with dual tires. Also, for obvious reasons, the tractor ESC on condition was tested only for combinations that involved the vehicle traveling at high speed.

Table 4 presents a summary of the tests that were conducted for each combination of tires, CG level, tractor ESC setting, and vehicle speed. Each entry in the table indicates for the corresponding combination (column headers), the number of runs that were conducted for that specific maneuver (row header), followed by the size of the data files (in parenthesis) containing the information gathered, in megabytes. Consider, for example, the first cell of the body of table 4. Four runs were conducted for the clockwise (CW) constant radius maneuver, with the vehicle equipped with dual tires, traveling at high speed with a high-CG payload, and with the tractor ESC not engaged[1]. During these tests, the sensors collected a total of 13.24 megabytes of information resulting from the 0.01-sec sampling of the 34 variables described in table 1. For the same vehicle settings, no tests were conducted for the CCW constant radius maneuver; 13 tests were performed for the lane change (Highway Evasive) maneuver (13.36 megabytes); and 17

1 High speeds were around 30 and 40 mph, with and without the ESC system engaged. Low speed tests were conducted at around 30 mph.

tests were performed for the step steer maneuver (24.36 megabytes)[2]. The totals for these vehicle settings were 34 tests performed, and 50.96 megabytes of collected information; while for the entire battery of tests for all combinations studied, these figures were 184 tests and 298.44 megabytes, respectively. Three additional runs (3.20 megabytes of information) were collected and used to determine "zero" conditions for several sensors. Besides collecting information directly from the sensors mounted on the tractor and the flatbed-trailer, video images were also gathered for the tests, as well as GPS information (using an independent VBox system) that collected spatial information at higher resolution (5 Hz) than the latitude and longitude channels collected by the data acquisition system (1 Hz).

Table 4. Summary of Number of Runs Conducted During On-Track Testing for Various Testing Configurations

Maneuver	Dual Tires						NGSWBTs			Total
	High-CG			High-CG			High-CG			
	Tractor ESC Off		Tractor ESC On	Tractor ESC Off		Tractor ESC On	Tractor ESC Off		Tractor ESC On	
	High Speed	Low Speed	High Speed	High Speed	Low Speed	High Speed	High Speed	Low Speed	High Speed	
Constant Radius - CW	4 (13.24)*	2 (18.13)	2 (3.72)	4 (19.96)	1 (7.48)	3 (9.6)	4 (14.35)	1 (8.87)	2 (4.1)	23 (99.45)
Constant Radius - CCW					1 (9.32)			1 (10.87)		2 (20.19)
Lane Change	13 (13.36)	7 (8.42)	3 (2.98)	15 (11)	6 (6.91)	8 (6.37)	10 (10.8)	6 (7.01)	2 (2.29)	70 (69.14)
Step Steer	17 (24.36)	6 (8.19)	6 (6.33)	19 (17.36)	6 (8.81)	8 (6.78)	16 (22.88)	6 (9.32)	5 (5.63)	89 (109.66)
Total	34 (50.96)	15 (34.74)	11 (13.03)	38 (48.32)	14 (32.52)	19 (22.75)	30 (48.03)	14 (36.07)	9 (12.02)	184 (298.44)

*The body of the table gives Number of Files (Size in megabytes)

DATA ISSUES

The lessons learned during earlier phases of the project indicated the need for tools that would help identify, in real- or quasi-real-time, problems with the information gathered due to sensor malfunction or other type of issues affecting these sensors; or the data acquisition system itself. Based on this identified need, ORNL developed software (SensorChk) that:

- read the files generated by the data acquisition system (DAQ),
- translated the information from the native format of the DAQ to a comma-separated-values (csv) flat text file, and
- checked that the values read for each sensor were within its expected range of operation.

The software also generated statistics on, for example, the percentage of time that the sensor was operating at its lower or upper limits, the percentage of time there were errors attributed to the DAQ system itself, and other measurements that helped to automatically evaluate (based on user input parameters) whether or not each sensor was working normally. The system had a very simple interface that quickly indicated to the user the status of the system for the particular run

2 While each run of the lane change and step steer tests contains just one replication of the maneuver, one run of the constant radius tests contains several replications of the maneuver (i.e., several circles were completed in each run).

processed (see figure 32 and figure 33). During all of the tests at TRC, the sensors functioned without problems, except on one occasion in which one sensor malfunctioned. The problem was identified and reported by the SensorChk utility software, and the sensor was subsequently repaired and put back into service.

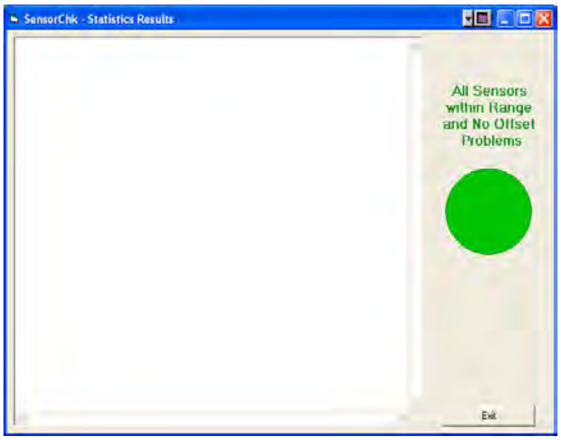


Figure 32. Screen Shot. Data Checking - No Sensor Problems.

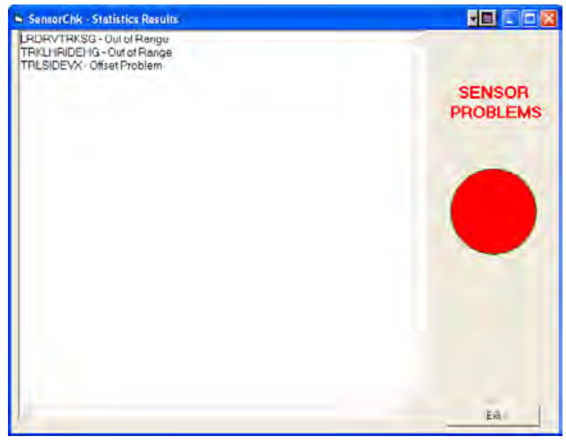


Figure 33. Screen Shot. Data Checking - Sensors with Problems.

Further analysis of the data in real-time was performed by software created by MARC. That software consisted of a set of MATLAB routines that read the csv files produced from the ORNL SensorChk application very quickly and plotted the data from multiple runs; checking for data quality and sensor response in a near-real-time manner. These tools were used to monitor driver and sensor performance and to improve the test execution.

The tool developed by MARC was useful in showing the driver what he was actually doing and facilitated better understanding of the objectives of the testing. Additionally, the tool could quickly show the response of the vehicle in the last maneuver so that good decisions could be made on needed changes for the next maneuver. As such, the tool could be used to evaluate the next proposed maneuver based on the past maneuver's responses.

The tool did the following tasks in an automated manner:

- read in multiple csv files where each file contained the results of a single pass.
- corrected height sensor errors.
- corrected the raw data for "zero" errors based on an ASCII file containing the sensor's values for each channel when the vehicle was at its nominal state.
- interpolated the GPS sensor data to fill in the under-sampled gaps in the latitude and longitude data.
- converted the GPS data to X- and Y-direction motions (X is motion along a given latitude, Y is motion along a given longitude) using a Universal Transverse Mercator grid system transformation (UTM).
- read an additional ASCII file containing the location of the sensors on the truck so that calculated channels could be generated. As an example, the roll angle of the tractor-flatbed-trailer was calculated from two height sensors on each side of the tractor-flatbed-trailer.

- determined the roll-corrected lateral accelerations.
- calculated the vehicle path radius for circular tests.
- filtered the data for improved data processing.
- plotted the data for quick field analysis in a near-real-time manner.

In combination, these two utilities helped to: assure the data quality of the information collected; and helped to identify small problems with the execution of the maneuvers which could be fixed before continuing with the tests. In summary, there were very few data issues which were not corrected immediately, thus assuring a high quality of the data collected. Nevertheless, there were some issues with the data that only surfaced during the data analysis task such as, for example, the zeroing of the steering wheel angle. Those issues were identified post-testing and were added to the list of lessons learned from Phase-A research.

DATA PREPARATION FOR ANALYSES

As mentioned above, the ORNL developed SensorChk utility read the data collected by the DAQ system and translated the information from its native SIF (Somat Information File) format to a flat csv text file. The csv data consisted of data columns with the channel name and unit at the top as shown in figure 34.

TimeStamp	speed_mph	lon	lat	TRKVELOCITY
	mph	Degrees	Degrees	MPH
2008-05-09T17:08:18.0000000	22.90626526	-83.54373169	40.30858994	25.01362991
2008-05-09T17:08:18.0097656	22.90626526	-83.54373169	40.30858994	24.95663834
2008-05-09T17:08:18.0195312	23.46549988	-83.54367065	40.30850601	24.97292137
2008-05-09T17:08:18.0292969	23.46549988	-83.54367065	40.30850601	24.92407227

Figure 34. Raw Data.

MARC developed a set of applications in MATLAB that read and evaluated these csv files. The MATLAB applications read each channel and recorded the name and unit information as well. Further, when loading the data, the application converted the time stamp (first column) into a decimal-second-time with the beginning of the run set as the “0” time point.
The data was organized into structured variables for easy manipulation and clarity of data organization. The data retained included the:

- data file name and location (i.e. directory information),
- number of data channels,
- name of each data channel,
- units for each data channel,
- data sample rate,
- number of data points,
- number of data cycles (number of points/sample rate), and
- raw data for each channel.

After the data was read, converted to the right format, and saved into the database, several procedures, described below, were applied to the collected information to prepare it for the data analysis and modeling tasks.

Height Sensor Correction

Through the use of two height sensors on opposite sides of the vehicle, the roll angle of the vehicle could be calculated using simple trigonometry. Unfortunately, the six height sensors used to evaluate vehicle roll angle were not well behaved, and reported a significant volume of invalid data as well as significant noise. To obtain usable information, applications had to be developed to filter and interpolate the signal to recoup the vehicle's true height change. Because a vehicle with 80,000 lb mass does not change height suddenly, the filtering could assume that the true vehicle motion was a low-frequency motion. Figure 35 provides an example of the as-measured data and the correction of the data.

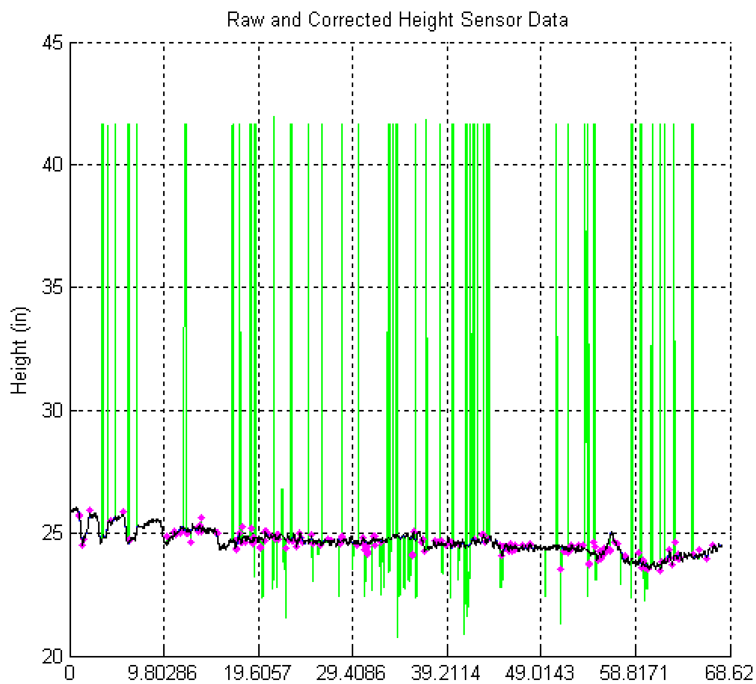


Figure 35.Graph. Height Sensor Correction (Height [in] vs. Time [sec]).

Here the green line is the measured signal. The pink dots are dropped data points, and the black line is the resulting vehicle height. Several techniques were used in the cleaning of the data to address differing problems with the raw data. This process was developed in the MARC software and was included in the field data evaluation capabilities.

The first data cleaning method involved a histogram-based trimming of the false high amplitude spikes in the data. This was accomplished by looking for bi-modal data and eliminating the upper mode. The technique involved placing a second-order fit through the histogram of the height data and determining where the minimum of the resulting fit occurred. If the minimum occurred inside the measured height data range, then a significant amount of false high data existed. Data above the minimum point in the fit were removed.

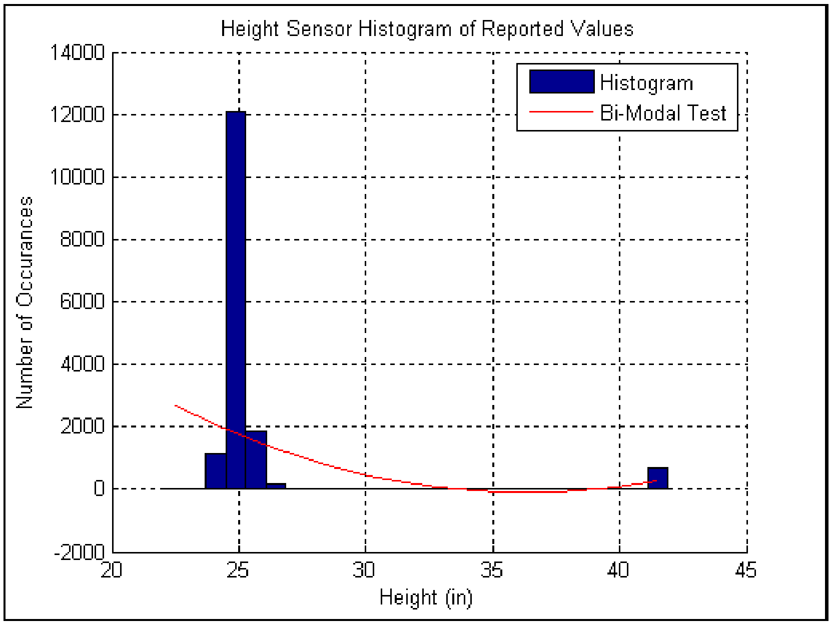


Figure 36. Graph. Height Sensor Invalid Bi-Modal Test.

For cases with fewer high amplitude spikes, the analysis did not produce a bi-modal response and no data was removed in this step.

The second cleaning method was a residual analysis where the remaining data was fitted using a fifth-order, 10 Hz low-pass Butterworth filter. An effective standard deviation of the data with respect to the fit was determined by evaluating the difference in the fit and the remaining raw data. The data points that were greater than 2.5 times the standard deviation were removed. This was done one point at a time with a new Butterworth fit being calculated each time a point was removed until no points remained that were greater than 2.5 times the standard deviation; these are the pink points in figure 35. The idea here is that the outliers represent higher frequency deviations (lower frequencies drag multiple points and thus the filter) and because the vehicle does not physically roll much faster than 5 deg/sec, the outliers are considered false points. This filter is, in effect, a modified robust regression technique.

The final step in the data cleaning was to linearly interpolate new values for the false data using a cubic spine fit that passed through each remaining data point.

<u>Data Zero</u>

When reading the data, the raw data was "zeroed" by subtracting the nominal readings for each sensor as determined during the static measurements. Table 5 is the zero file for the dual tire high-CG payload configuration case. Note this also serves as a channel listing because each channel has a zero.

Table 5. Data Zero File

```
% zero file

timestamp.val       = 0
speed_mph.val       = 0
lon.val             = 0
lat.val             = 0
trkvelocity.val     = 1.51735
trksidevx.val       = -0.275144
lrinttrksg.val      = -4.484
rrinttrksg.val      = -2.979
lrdrvtrksg.val      = -2.046
rrdrvtrksg.val      = -2.9042
trkstrangle.val     = 0.99926
trkstrwhangl.val    = 13.1236
trklhridehgt.val    = 24.8139
trkrhridehgt.val    = 25.3
trkxaxis.val        = 0.00671797
trkyaxis.val        = -0.02336933
trkzaxis.val        = 0.999857
trkrollrate.val     = -0.149129
trkpitchrate.val    = -0.163276
trkyawrate.val      = -0.311592
trlangle.val        = -2
trllfridehgt.val    = 17.8102
trlrfridehgt.val    = 17.9075
lrtrlhgt.val        = 19.3193
rrtrlhgt.val        = 19.3717
trlvx.val           = 0.00641069
trlsidevx.val       = 0.0227806
trlyaxis.val        = 0.0256914
trlyawrate.val      = -0.181762
trlyawrate.val      = 0.13668
trllfstrain.val     = -.3745
trlrfstrain.val     = .1107
trllrstrain.val     = -.3151
trkxaxis2.val       = -0.004154
trlrrstrain.val     = -.0700
```

GPS Data

The data was collected at 102 Hz except for the GPS data which was collected at 1 Hz. This was due to the limitations of the GPS system. However, the vehicle's path did not include much noise which allowed for the interpolation of the 1 Hz path data to match the 102 Hz of the remaining channels using a piece-wise cubic spline. This was possible due to the fact that it takes a significant force to cause an 80,000 lb vehicle to deviate from a given trajectory; so it could be safely presumed that the vehicle followed a smooth arc from one data point to the next. The raw GPS data for a circle test can be seen in figure 37.

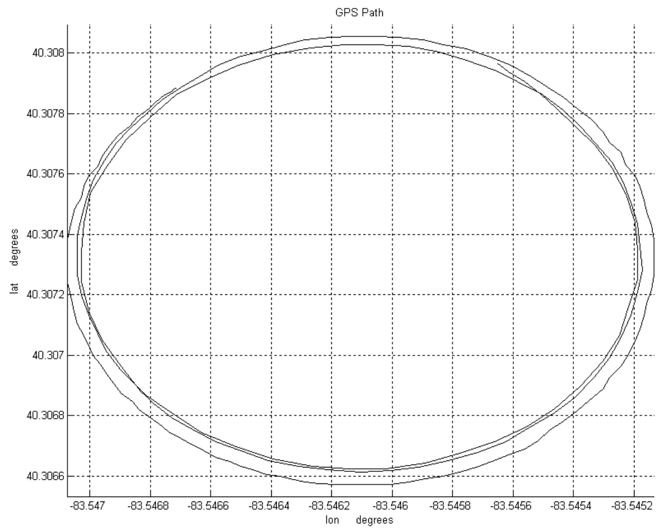


Figure 37. Graph. Raw GPS Data for a Circle Test.

Once the GPS data was interpolated, the GPS data was passed through a Universal Transverse Mercator grid transformation (UTM) to convert the latitude and longitude values into physical distances. This enabled the path of the vehicle to be tracked in real engineering units. The figure 37 circle test is shown in figure 38 in X- and Y-displacements, where X is derived from longitude and Y is derived from latitude. Note: this type of transformation defines the X-axis as a distance from the central meridian and the Y-axis as a distance from the equator. Because only the relative path of the vehicle was needed, these were zeroed to the initial position of the vehicle at time "0" to highlight the change in position of the vehicle rather than the location of the vehicle on the planet.

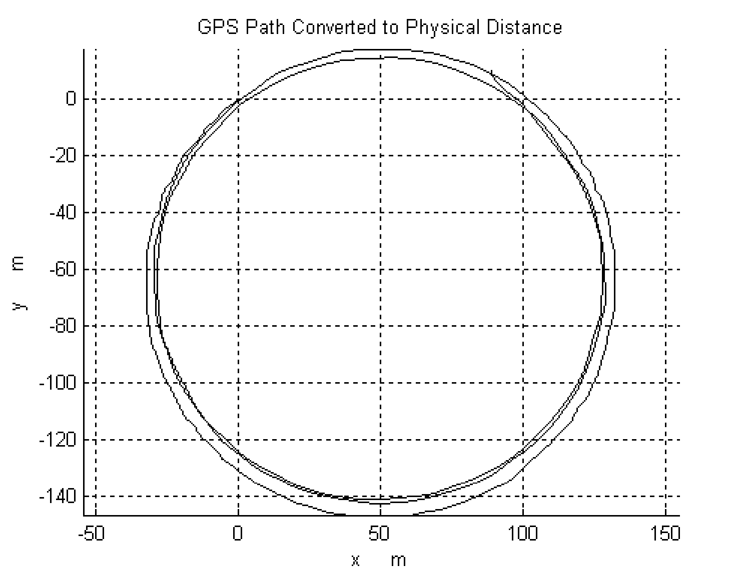


Figure 38. Graph. Vehicle Path Converted to Physical Distance (X- and Y-Displacements).

Calculated Channels

The height sensors were used with known lateral spacing between the sensors (read from a text file containing the sensor location data), to calculate the roll angle of the vehicle.

Table 6. Sensor Location File

```
% height sensor space
trk.space1       = 40.75;
trl.space1       = 40.5;
trl.space2       = 40.25;
```

$$Roll\_Angle = \tan^{-1}\left(\frac{Right\_Sensor\_Height - Left\_Sensor\_Height}{Sensor\_Spacing}\right) \qquad \mathbf{5}$$

Figure 39. Equation 5.

The vehicle roll was calculated at three locations: on the tractor, just ahead of the drive axles; on the flatbed-trailer at the landing gear; and on the flatbed-trailer, just ahead of the flatbed-trailer's axles.

Using the roll angles, the measured lateral acceleration of the vehicle could be corrected to remove the false acceleration measured due to the roll of the vehicle. The correction was applied to the tractor and the flatbed-trailer using the tractor roll angle and the rear flatbed-trailer roll angle, respectively. Note: Ay (lateral acceleration) is measured in units of gravitational acceleration (g) (Note: Ayc denotes the corrected lateral acceleration).

$$Ayc = Ay - \sin(roll\_angle) \qquad \mathbf{6}$$

Figure 40. Equation 6.

Vehicle Radius Estimation

For the circle tests, it is necessary to know the vehicle turning radius to properly evaluate the data. To determine the radius, the vehicle path was evaluated over a 30 sec window. Using this path segment, an arc was fitted, using the origin (x and y center coordinates) and the radius as the three independent parameters. These parameters were optimized such that the error between the path and the arc was minimized. The radius estimation was evaluated for each data point using 15 sec on each side of the point through which the arc was to be fit.

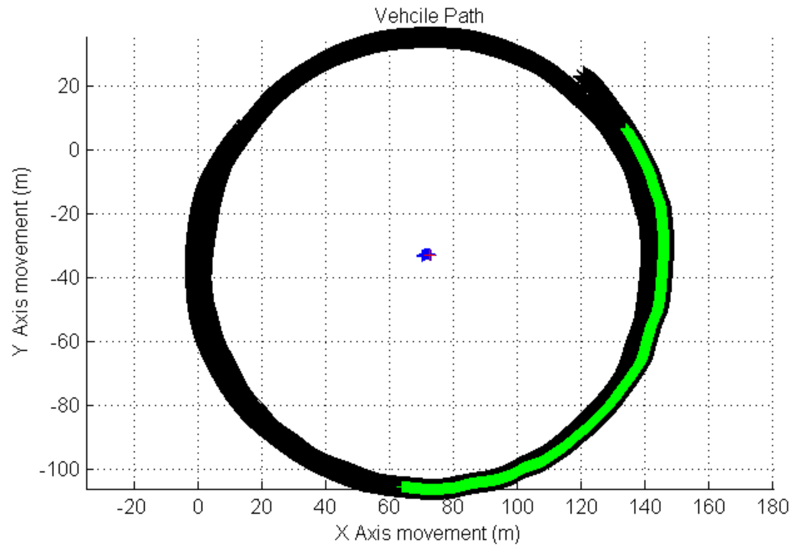


Figure 41. Graph. Circle Test Radius.

Data Filtering

Once the data had been collected and corrected for sensor errors, the data was filtered using a third-order 5 Hz low-pass Butterworth filter. This was done to remove noise from the data so that the observed vehicle response was smoother, and the trends in the data were clearer. Because any response of interest of the vehicle would occur below 5 Hz, this filtering did not affect the conclusions of the analysis.

Field Data Plotting

Using the data structure defined above, MATLAB-based plotting tools were developed to easily plot the raw or filtered data for any of the channels where the sequential passes were plotted individually. These tools were very useful in evaluating the data in the field to identify sensor issues and for maneuver tuning.

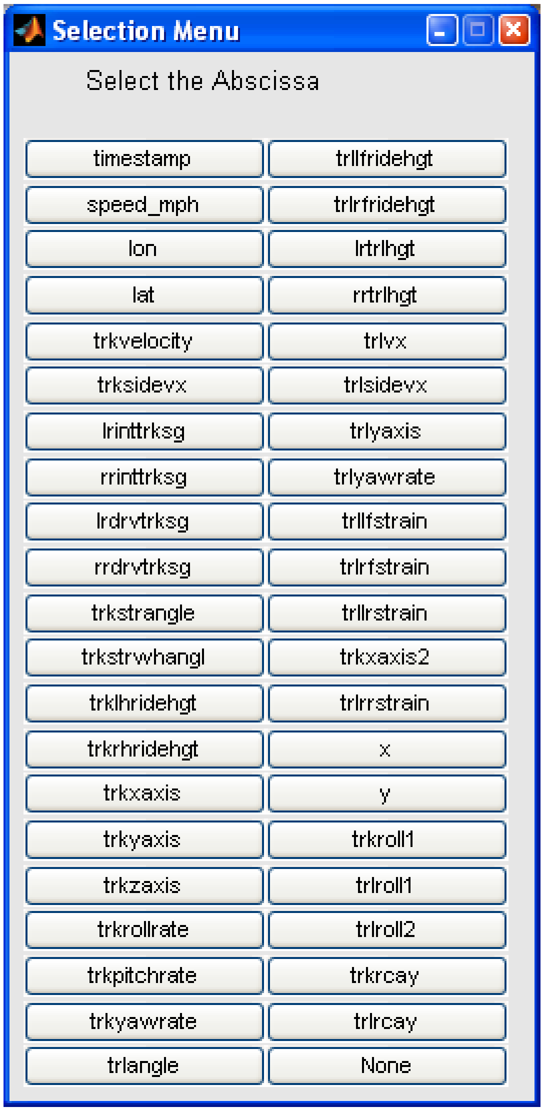


Figure 42. Screen Shot. Abscissa Data Selection.

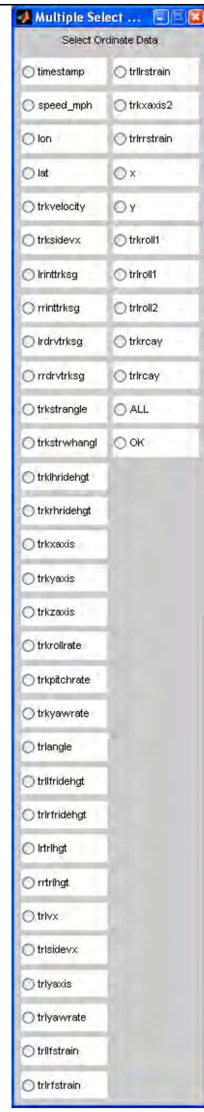


Figure 43. Screen Shot. Ordinate Data Selection.

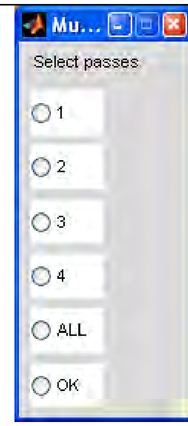


Figure 44. Screen Shot. Data Pass Selection.

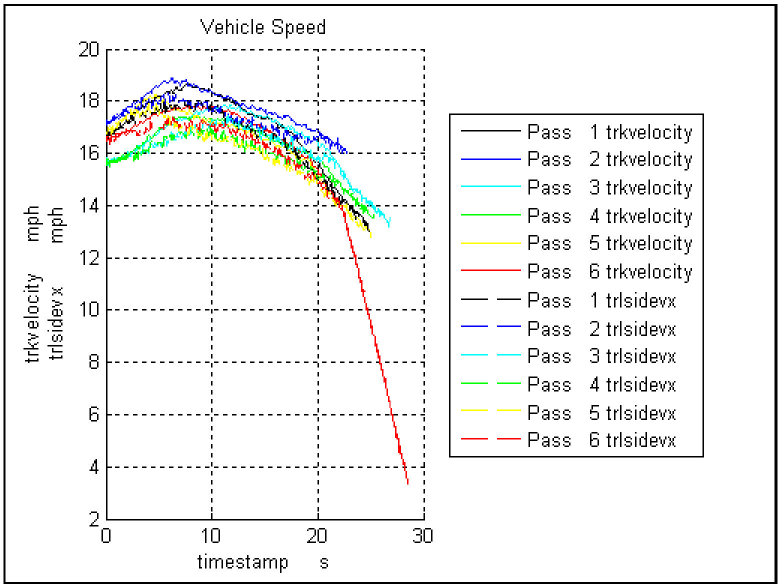


Figure 45. Graph. Tractor and Flatbed-Trailer Velocity vs. Time for Six Passes.

Using this tool, multiple passes from a maneuver could be compared for assessment of repeatability. Passes from differing test conditions (e.g., payload or tire configuration) could be compared to check for maneuver consistency, and multiple channels from one or more passes could be plotted to check vehicle response characteristics.

It was through the use of this tool that several bad sensors were discovered during testing. This tool was also used to review the driver's actual steering input to determine what adjustments, if any, were needed to meet the desired steering input. Finally, the field plotting tool also helped the test team determine what maneuver modifications to make for cases where the vehicle responses were not what were expected.

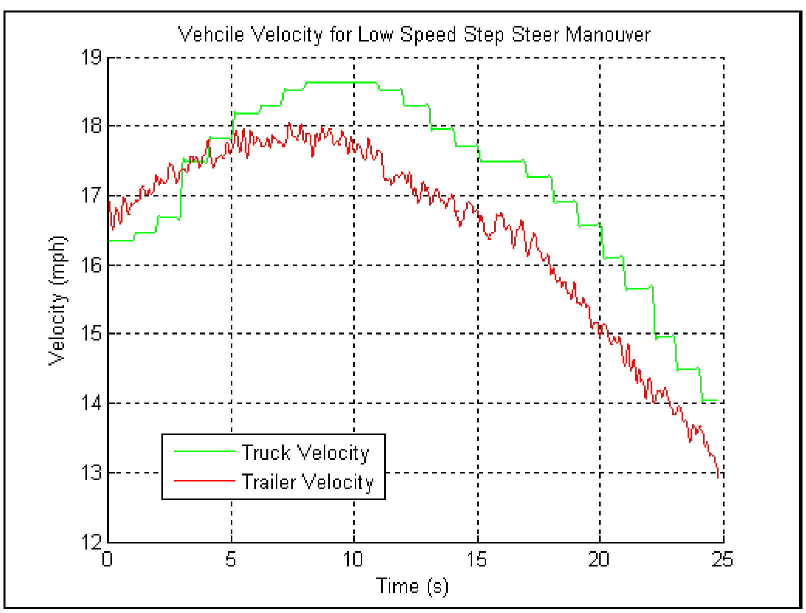


Figure 46. Graph. Two Channel Comparison.

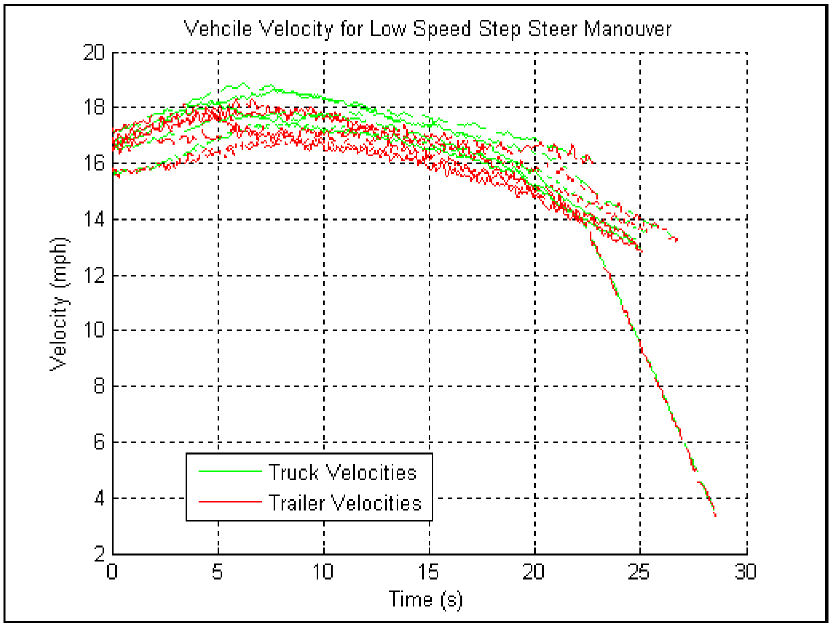


Figure 47. Graph. Two Channel Overlay.

CHAPTER 10. DATA ANALYSIS

BACKGROUND

This chapter is divided into three main sections. The first one focuses on theoretical concepts related to vehicle dynamics and derives the parameters that were used in the analysis of the maneuvers. The second section presents a detailed analysis of the three maneuvers that were tested; that is, constant radius, step steer, and lane change maneuvers. Finally, the last section of this chapter presents the conclusions of the data analysis.

THEORETICAL CONCEPTS

Rollover Threshold

Many factors affect the rollover threshold of a tractor-flatbed trailer combination. Rollover threshold is defined as the maximum lateral acceleration, usually specified in units of g, that the vehicle can experience before becoming unstable. Analytical simplification of rollover can be used to obtain insights into the rollover phenomenon, even if these simplified solutions may not always predict accurately the actual vehicle behavior. Since all cornering forces (Fc) act at the ground (see figure 48), the overturning moment (Mo) is established by the CG height (h), the lateral acceleration that the CG experiences (ay), the magnitude of the load (w) and the instantaneous load position (side-to-side)) as presented in equation 7. A restoring moment (MR) counters the overturning moment and ultimately is established by the radial load on the tires (Fr) and the vehicle track width (t).

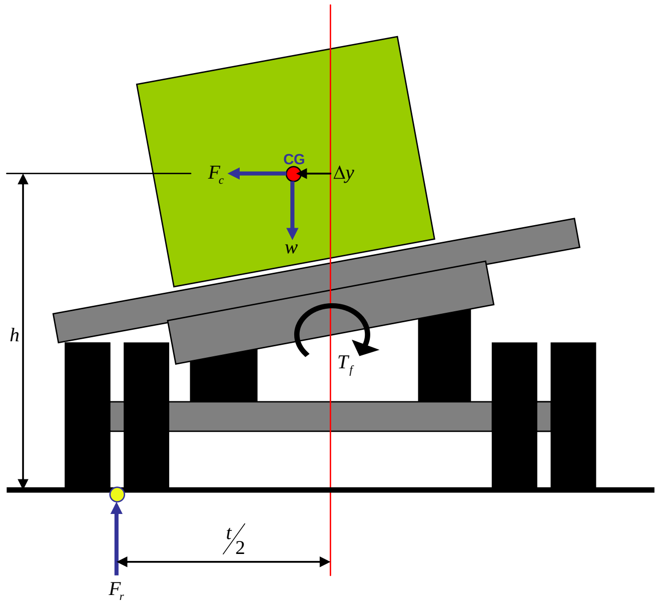


Figure 48. Illustration. Simplified Roll Physics.

$$M_o = (m) * h * a_y = F_c * h \qquad \textbf{7}$$

Figure 49. Equation 7.

$$M_R = (m * g) * \left(\frac{t}{2} - \Delta y\right) = w * \left(\frac{t}{2} - \Delta y\right) \qquad \textbf{8}$$

Figure 50. Equation 8.

$$RS = \frac{T}{2 * h_{cg}} \qquad \textbf{9}$$

Figure 51. Equation 9.

For the flatbed-trailer under study, the payload was not a single load but was either distributed along the spine of the trailer (flat load) or discretely segmented at elevated heights in specially designed load racks (low-CG payload and high-CG payload). The very first tests were performed with the load distributed along the trailer spine at a height of approximately 16 in above the flatbed-trailer bed (flat load). This condition produced static stability margins (equation 9) in the 0.6 g range which was high enough that test speeds were considered uncomfortably high for wheel lift; to the point of questioning the safety of the test.

The next test utilized payloads positioned in load racks that were effectively positioned longitudinally above the drive axles of the tractor and above the flatbed-trailer axles. The load rack's horizontal positions as measured from the rear of the flatbed-trailer to the middle of the load racks were: Front Rack = 530.5 in; Rear Rack = 69.5 in. Two payload heights were used in this testing; the high-CG payload configuration which positioned the CG of the payload at 9’8” and 9’3” in the front and rear load racks, respectively, and the low-CG payload where the racks were approximately 12 in lower, resulting in payload heights of 8’7” and 8’3” in the front and rear load racks, respectively. The third set of tests utilized the load racks with the payloads lowered by approximately 12 in, resulting in payload heights of 8’7” and 8’3” in the front and rear load racks, respectively. While the load was configured in the high-CG payload configuration, the tires were changed from the NGSWBTS to dual tires to allow back-to-back testing. In the high-CG payload configuration, both NGSWBTS and dual tires were tested. The low-CG payload configuration was tested with the dual tires only. These load configurations are listed in table 7.

Table 7. Basic Test Configurations

Purpose	Tires	Front and Rear Load Cage Heights		Outrigger Positions Measured From the Rear of the Flatbed-Trailer (in)
Preliminary System Check and Evaluation	NGSWBTS	High-CG	9’8” & 9’3”	156.5 and 389.5
Data Collection	NGSWBTS	High-CG	9’8” & 9’3”	156.5 and 389.5
Data Collection	Dual Tires	High-CG	9’8” & 9’3”	
Data Collection	Dual Tires	Low-CG	8’7” & 8’3”	

Payload position along the flatbed-trailer axis influences the rollover threshold, especially when a trailer with a relatively low torsional stiffness is examined. The restoring moment, depending on the position of the load, can come from the factors shown in equation 8 as well as the torsional characteristics of the frame. The flatbed-trailer frame could add restoring moment through its connection at the fifth wheel. A number of additional factors then come into play including suspension roll stiffness of the tractor and/or flatbed-trailer, tire characteristics, frame torsional stiffness and load position relative to each suspension system. The restoring moment is created by a combination of gravity and the axial torque produced in the vehicle’s frame. The overturning moment is created by forces from centripetal acceleration.

Many factors therefore contribute to the rollover threshold. Although the standard track width (t) has increased over the years, it is still limited by road width and the DOT maximum width limit of 102 in. The CG height could be considered a design criterion, but is conventionally limited by loading dock height, load placement/distribution and tire clearance. CG lateral shift is caused by tractor and trailer component compliances that result in a rotation of the sprung mass as a function of its distance from the suspension roll axis. Off-center load placement also results in lateral CG location offsets. Compliances such as suspension roll, tire radial stiffness, frame torsional stiffness, fifth-wheel lash, and off-center cargo all contribute to changes in Δy (the horizontal displacement of the load relative to the center of the track). When analyzing one set of axles, such as the flatbed-trailer axles, restoring frame torque (T_f) results from an angular difference in axial rotation at the analyzed axles, to the axial rotation at the nearest axles or at the fifth wheel. In this case, the frame restoring moment would result from the difference between the angular roll of the bed at the trailer axles and the angular roll at the drive axles, or fifth-wheel assuming the fifth wheel is in total contact. In a coupled vehicle (fifth wheel in full contact), the axle sets do not act independently of each other, restoring moments at each axle set are additive and transfer to one another through the vehicles frame. At some degree of roll, it is possible to partially decouple the roll of the tractor and the flatbed-trailer because the flatbed-trailer starts to separate from the tractor's fifth wheel through vertical translation of the king pin relative to the fifth wheel.

Establishing the Equivalent Tractor Wheelbase

In establishing the steady-state dynamics, one of the first concerns was to establish the equivalent tractor wheelbase. Since the tractor has dual drive axles, competing slip angles are induced between the drive wheels even at very low velocities, when in a turn (see figure 55). This induced slip is a result of forcing two separated but parallel axles to rotate about a common center, in a turn. This induced slip complements side slip on one drive axle (forward) and opposes it on the other (rear), of an axle set. This induced slip acts to increase the effective wheel base of the tractor beyond that of the conventional location, which is assumed to be centered between the drive axles. The increase in effective wheel base means that the front wheels need to steer into the turn further to negotiate a given radius.

The equivalent wheelbase length (Le) is approximated by equation 11. The tandem factor (T) in equation 10, relates the axle spacing to the mean or average position [11]. In the case of two drive axles the term (Δ) is equivalent to half of the spacing between the axles.

$$T = \left[\frac{1}{N} \sum_{i=1}^{N} {\Delta_i}^2 \right] \qquad \textbf{10}$$

Figure 52. Equation 10.

$$L_e = L \left[1 + \frac{T}{L^2} \left(1 + \frac{C_{F\alpha 2}}{C_{F\alpha 1}} \right) \right] \qquad \textbf{11}$$

Figure 53. Equation 11.

Table 8 shows the equivalent wheelbase length factors and compares the difference in length of the "customary" length with the equivalent wheelbase length.

Table 8. Equivalent Wheel Base Evaluation.

Equivalent Wheelbase				
Steer Axle-to-Drive Axle 1 Spacing	5.829	m	229.5	in
Steer Axle-to-Drive Axle 2 Spacing	7.173	m	282.4	in
Number of Drive Axles	2			
Steer Axle-to-the the Average of the Drive Axle Positions	6.501	m	255.945	in
Length from the Steer Axle to the Center of the Rear Drive Axle	0.672	m	26.457	in
Tandem Factor (T)	0.452	m^2	700.0	in^2
Cornering Stiffness of the Steer Tires	3,500	N/deg	785	lb/deg
	200,550	N/rad	44,976	lb/rad
Number of Tires on the Steer Axle	2			
Number of Tires per Drive Axle	4			
Cornering Stiffness of the Drive Tires	2,500	N/deg	561	lb/deg
	143,250	N/rad	32,125	lb/rad
Cornering Stiffness Factor	3.857		3.857	
Tandem factor divided by L^2	0.010685		0.010685	
Equivalent Wheelbase	6.769	m	266.5	in
L_{equiv}-L_{cust} =	0.268	m	10.55	in

The steered wheel angle (δ) needed to negotiate a turn is a combination of the Ackermann steer angle, shown in equation 12, and the collective tire steer resulting from the tire slip angle differences, and any suspension-related geometry steer. The steered wheel angle is shown in equation 13, which includes Ackerman Steer angle and the tire related steer. A vehicle with multiple drive axles has a turn induced slip angle at the drive axles, which is a function of the axle separation and the tire cornering stiffness. The force for this turn induced slip at the tandem axles must be generated at the front steer axle. As a result the ratio of the tire cornering stiffness between the steer axle and the tandem axles plays a large role in establishing the equivalent wheelbase. The equivalent wheelbase and the role of the tire cornering stiffness in the equivalent wheelbase determination is presented in equation 11.

Using the above equivalent wheel base model, the vehicle can be treated as a two axle vehicle with 8 rear tires. This makes it possible to use more traditional two axle vehicle dynamics models such as the one below to evaluate basic vehicle dynamics.

The theoretical amount of steering needed for a vehicle to negotiate a given turn is related to the turn radius and the vehicle wheelbase (or effective wheel base in this case). There is a geometric relationship defining this steering requirement. For a “zero speed” case, or a case where no lateral forces need to be generated to force the vehicle around the turn, this relationship is defined as the Ackerman angle. The Ackermann steer angle is shown in equation 12.

$$\delta_{\deg} = \frac{L}{R}\left(\frac{180}{\pi}\right) \qquad \textbf{12}$$

Figure 54. Equation 12.

where:

L = effective wheel base as defined above for the three axle vehicles

R = radius of the turn

L/R = Ackermann Angle (at very low, "0 slip angle" speeds)

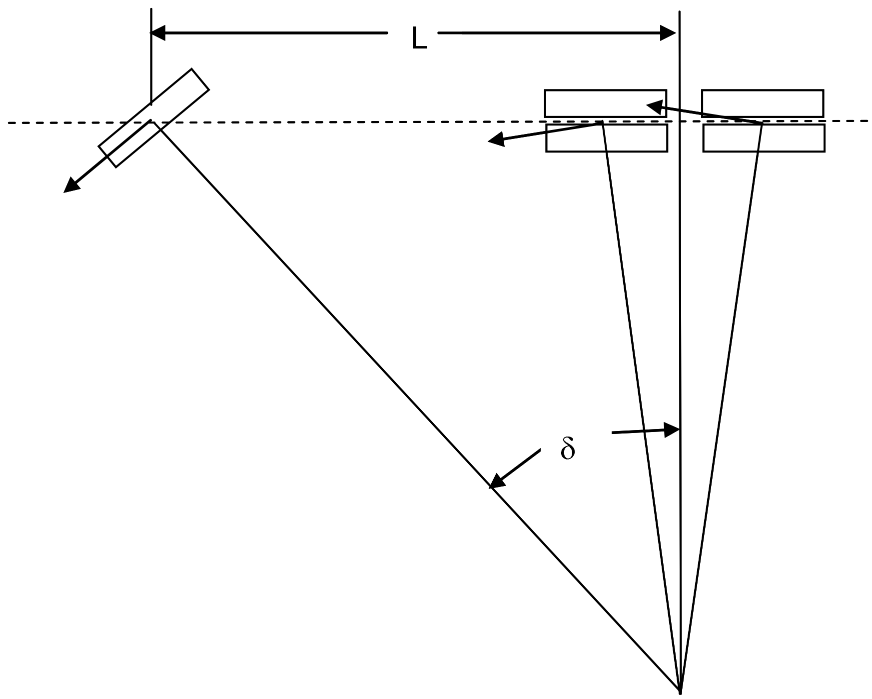


Figure 55. Diagram. Equivalent Wheelbase.

Since the weight on the axle sets (Wf, Wr), multiplied times the lateral acceleration represents the cornering forces (Fy1, Fy2), in a linear model the tire steer angle can be represented as a combination of the Ackerman steer angle plus tire slip angle relationships as shown in equation 13. The first term in the right side of equation 13 (i.e., [(L/R)*(57.3)] represents the Ackermann steer angle, while the second term $\eta\left(\frac{v^2}{Rg}\right)$ represents the dynamic steer gradient. The sign of the dynamic steer gradient indicates either an understeer (for positive values) or oversteer (for negative values) characteristic.

$$\delta_{\deg} = \frac{L}{R}\left(\frac{180}{\pi}\right) + \left[\frac{W_f}{C_{F\alpha 1}} - \frac{W_r}{C_{F\alpha 2}}\right]\left(\frac{v^2}{Rg}\right) = \frac{L}{R}(57.3) + \eta\left(\frac{v^2}{Rg}\right) = \frac{L}{R}(57.3) + \eta(a_y) \qquad \mathbf{13}$$

Figure 56. Equation 13.

where

- η = understeer gradient (deg/g)
- $C_{F\alpha1}$, $C_{F\alpha2}$ = total front and rear tire cornering stiffness respectively
- V = longitudinal (tangential) velocity
- W_f= weight over one front wheel (half of the front axle weight)
- $C_{F\alpha1}$= cornering stiffness of the front tires (force/degree)
- W_r= weight over one rear wheel (one-eighth of the drive axle total)
- $C_{F\alpha2}$= cornering stiffness of the rear tires (force/degree)
- a_y= lateral acceleration of the vehicle

With the equivalent wheelbase established, the turn radius can be established from the yaw velocity (ω_y) and the longitudinal velocity (v), and can also be established from the roll corrected lateral acceleration as indicated in equation 14.

$$a_y = v * \omega_y = v * \frac{v}{R} = \frac{v^2}{R} \qquad \mathbf{14}$$

Figure 57. Equation 14.

$$R = \frac{v^2}{a_y} \quad or \quad R = \frac{v}{\omega} \qquad \mathbf{15}$$

Figure 58. Equation 15.

Determination of the Under/Oversteer Gradient

With the equivalent wheelbase established, the understeer gradient (η) can be extracted from the experimental data with the relationships between yaw velocity (ω), forward velocity (v) and roll corrected lateral acceleration (ay) as shown in equation 14 and equation 15. Equation 16 and equation 17 provide the relationships leading to the understeer gradient (η) in equation 18 below.

$$\delta_{\deg} = \frac{L_e}{R}\left(\frac{180}{\pi}\right) + \left[\frac{W_f}{C_{F\alpha 1}} - \frac{W_r}{C_{F\alpha 2}}\right]\left(\frac{v^2}{Rg}\right) = \frac{L_e}{R}(57.3) + k\left(\frac{v^2}{Rg}\right) \qquad \mathbf{16}$$

Figure 59. Equation 16.

$$\delta_{\deg} = \frac{L_e}{R}(57.3) + k(a_y) \qquad \textbf{17}$$

Figure 60. Equation 17.

$$k = \frac{\delta_{\deg} - \frac{L_e}{R}(57.3)}{a_y}(\deg/g) \qquad \textbf{18}$$

Figure 61. Equation 18.

Note: 57.3 is 180/pi and it denotes the transition from radians to degrees for angular measurement.

<u>Transverse Weight transfer Due to Engine Torque</u>

The tractor frame is also the mechanism by which the drive shaft torque is resisted by, and distributed to, the front and rear suspensions. Since the engine and transmission are located toward the front of the frame, and the drive axles to the rear, the drive shaft torque is passed to the engine-frame mounts and resisted by the front suspension roll stiffness, and the coupling that occurs between the engine mounts and rear suspension, through the frame. A torsionally compliant frame facilitates little coupling to the drive axle suspension, resulting in a force imbalance between the right-front and left-front tires. This force imbalance is present whenever a drive shaft torque is present. A low selected gear (high ratio) coupled with a high engine torque output, results in high drive shaft torque and considerable transverse weight transfer across the front axle.

The force imbalance is reflected to the drive axles as well, and results in a diagonal weight change (the left rear and the right front increases, and the right rear and left front decreases) while torque is applied with a conventional rotation engine and drive line. Although this force difference can be considerable under high torque situations while in the low selected gears, it is less significant under steady-state, higher selected gear conditions. Estimates were made to determine the steady-state drive shaft torque at the speeds at which the maneuvers were encountered.

The force imbalance across the steer axle can result in an asymmetric understeer gradient in CW vs. CCW turns. Typically in a CCW turn, with drive line torque applied and conventional drive line rotation, the lateral acceleration weight transfer and the transverse weight transfer due to the torque are complementary on the steer axle, increasing the differential loading across the axle, and increasing understeer. In a CW turn the lateral acceleration weight transfer and the transverse weight transfer are in opposition on the steer axle, reducing the differential loading on the axle, and reducing understeer. The opposite effects are true on the drive axles (there is a complementary weight transfer in a CW turn and an opposing weight transfer in a CCW turn). The combined result of the drive axle weight transfer and the steer axle weight transfer in a CCW turn increases the front slip angle and decreases the rear slip angle, which are complementary effects for understeer. The combined result of the drive axle weight transfer and the steer axle weight transfer in a CW turn decreases the front slip angle and increases the rear slip angle, which are complementary effects for reduced understeer.

The magnitude of the differential between the CW and CCW turns are a function of the non-linear slip angle characteristics of the steer axle tires, the drive line torque, and the proximity to the traction limits on both steer and drive axles. The procedures used in the on-track testing did not produce consistent and accurate zero's for the steering transducer, which prevented decisive assessment of any asymmetries found in the steering in CW vs. CCW maneuvers. Analysis was performed to extract the understeer characteristics in the maneuvers, however confidence in the asymmetries was lacking due to the perceived inaccuracy of the initial zero of the transducer. Further study is needed to accurately extract this characteristic.

FIELD TESTING

The combined tractor and flatbed-trailer was tested using three basic maneuvers: a constant steering wheel angle maneuver (which is similar to a constant radius maneuver), a lane change maneuver, and a step steer maneuver. Each of these maneuvers was performed in both a low- and high-speed setting to gage the vehicle's responses to inputs and roll stability. Details of the test maneuvers and the execution of the test maneuvers can be found in the test maneuver section of this report (see 0).

As can be seen from figure 62, the loading consisted of two discrete masses centered over the drive and flatbed-trailer axles. This type of loading was designed to simulate the transportation of large bulk items with relatively dense material properties. The resulting vehicle axle loads are shown in figure 63. The vehicle was tested with the payload CG at two different heights: 9.5 ft above the ground and 8.5 ft above the ground.

In addition to the two payload heights, there were two sets of tires used in testing. Both the low-CG and high-CG payload conditions were tested using conventional 275/80 R22.5 dual tires on the drive and flatbed-trailer axles. An additional test with the high-CG payload case only was evaluated using 445/50 R22.5 NGSWBTs.

Figure 62. Photograph. Test Vehicle.

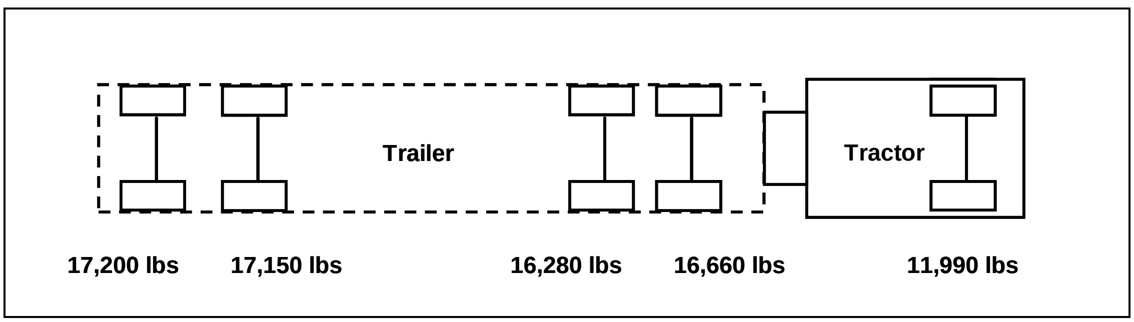


Figure 63. Diagram. Axle Loads.

Constant Radius Test

Both the low- and high-speed constant radius tests used a steering angle of approximately 100 deg at the steering wheel. This steering input resulted in a path with a radius that was reasonably close to a common highway exit ramp. Using GPS data, the radius of the vehicle could be evaluated for each test. The effective radius of the vehicle was evaluated over a moving 30 sec window as shown in figure 64. In this example, the vehicle was traveling in a CW direction. Note the red + marker indicating the current center of curvature and the blue + marker for indicating the center of curvature for previous segments of the test.

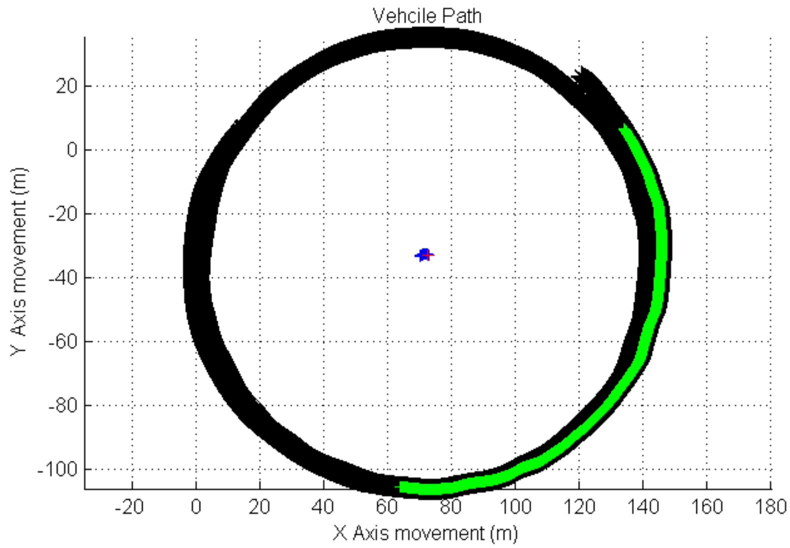


Figure 64. Graph.Vehicle Path for a 75 m Clockwise Constant Radius Test.

As can be noted in figure 64, the average radius of 75 m did not vary significantly over the test. Similar results were noted for all of the other test conditions in both the CW and CCW directions. To verify the radius calculation, the theoretical lateral acceleration (Ay) was calculated for each test case using Equation 19.

$$Ay = \frac{Velocity^2}{Radius} \qquad \textbf{19}$$

Figure 65. Equation 19.

The theoretical lateral acceleration was then compared to the measured lateral acceleration. The theoretical and roll-corrected lateral accelerations agreed fairly well as can be seen in figure 66.

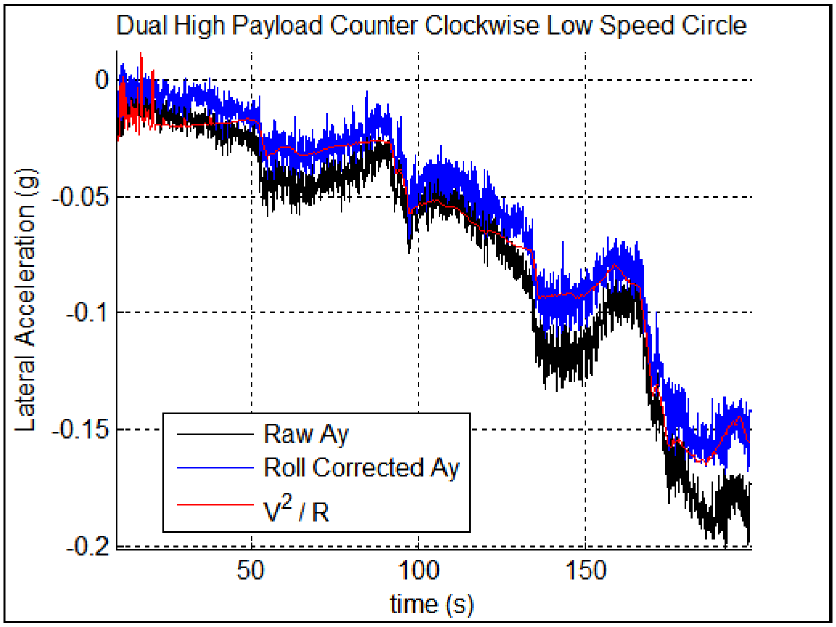


Figure 66. Graph. Vehicle Lateral Acceleration for a Clockwise Low-Speed Constant Radius Test (Dual Tires, High-CG Payload).

Low-Speed Constant Radius Test

The low-speed circle test was conducted over a speed range of 10-to-25 mph. The intent of the testing was to have a stepped speed test with increments of 5 mph per step, but the 1 percent slope in the test-track made this difficult, resulting in significant variation in the velocity as shown below in figure 67. However, the rate of change in the velocity was slow enough as to not impact the steady-state characteristics of the vehicle, which allowed the test data to be usable.

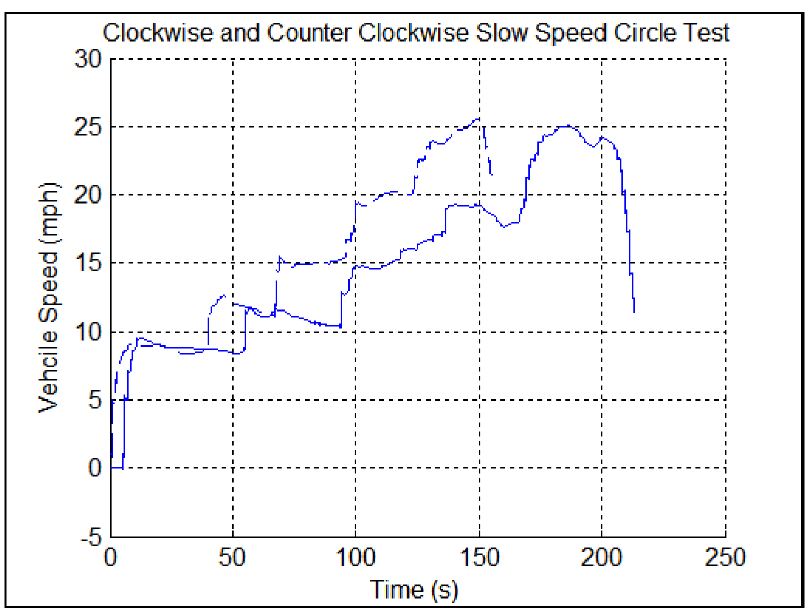


Figure 67. Graph. Speed Profile for a Slow-Speed Clockwise Constant Radius Test.

The main objective of this testing was to evaluate the understeer properties of the vehicle (see Determination of the Under/Oversteer Gradient above for more details). The vehicle exhibited very mild understeer for all cases evaluated. As an example, the steering response of the dual tire, high-CG payload case is shown below in figure 68.

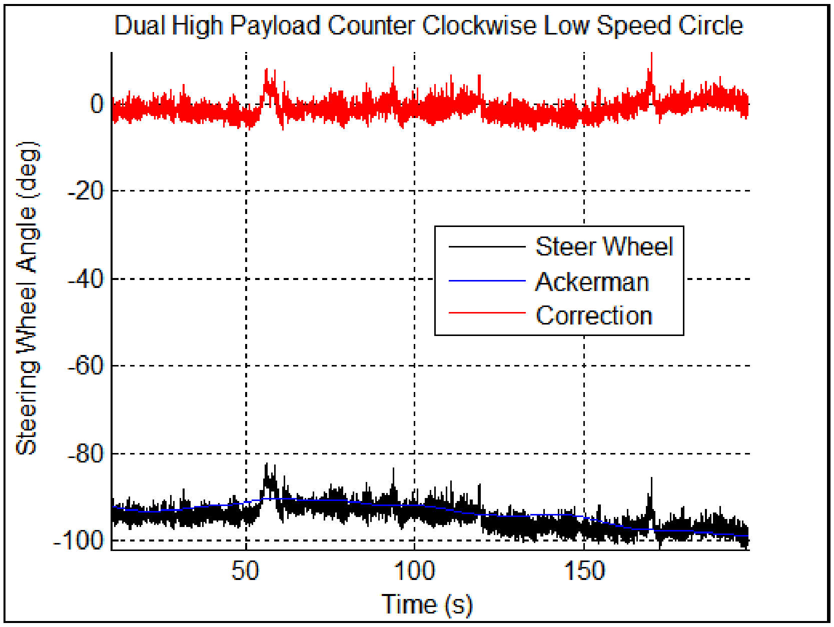


Figure 68. Graph. Test Input Steering Plot for a Counter-Clockwise Low-Speed Constant Radius Test (Dual Tires, High-CG Payload).

The black line is the actual measured steering input, the (thin) blue line is the Ackerman or neutral steering angle, and the red line is the error between the two. A neutral steering vehicle will have a steering ratio equal to the vehicle wheel base divided by the radius of curvature, multiplied by the steering system ratio (see Establishing the Equivalent Tractor Wheelbase above for a description of the vehicle's effective wheel base).

$$Ack = \frac{Eff.\_Wheel\_Base}{Path\_Radius} \qquad \mathbf{20}$$

Figure 69. Equation 20.

The red line is the actual steering minus the Ackerman angle. If the actual steering magnitude is greater than the Ackerman steering magnitude, as in this case, it indicates understeer, and more steering input is needed than Ackerman dictates, to hold the course.

As the vehicle speed increases with time, the above plot can be viewed as a function of the lateral acceleration (see figure 70).

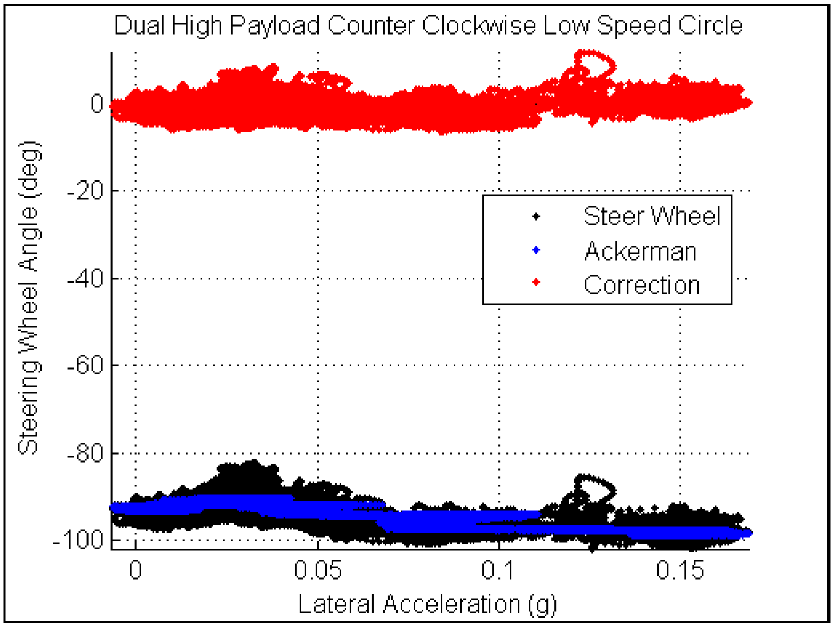


Figure 70. Graph. Test Understeer Plot for a Counter-Clockwise Low-Speed Constant Radius Test (Dual Tires, High-CG Payload).

The test data indicates that at low lateral acceleration levels, the vehicle needed approximately three degrees of steering input at the steering wheel above that which would be dictated by the Ackerman geometry in order to maintain the measured radius of curvature. At higher lateral acceleration levels, the steering input approaches the Ackerman requirement. This indicates that the vehicle is slightly understeering but at a very slowly decreasing rate with respect to lateral acceleration.

The value of the above plot is that it is easier to see the effect of changes in lateral acceleration. The amount of understeer exhibited by this vehicle does not seem to be influenced significantly by lateral acceleration. So the understeer gradient (K in equation 21, referred to as η in previous equations) is nearly zero for all tested lateral acceleration values.

$$\eta = \frac{d(\text{Ackerman})}{da_y} (\text{deg}/\, g) \qquad \textbf{21}$$

Figure 71. Equation 21.

All of the tested payload and tire combinations had very similar understeer curves. Changing tires or payload heights did not change the understeer response significantly.

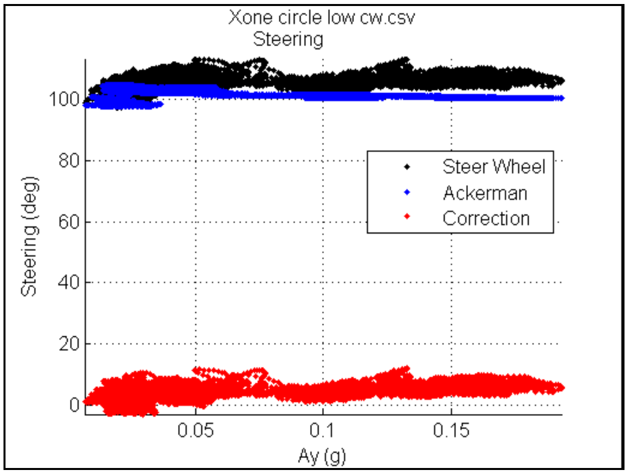


Figure 72. Graph. Understeer Test, NGSWBTS – High-CG Payload – Clockwise.

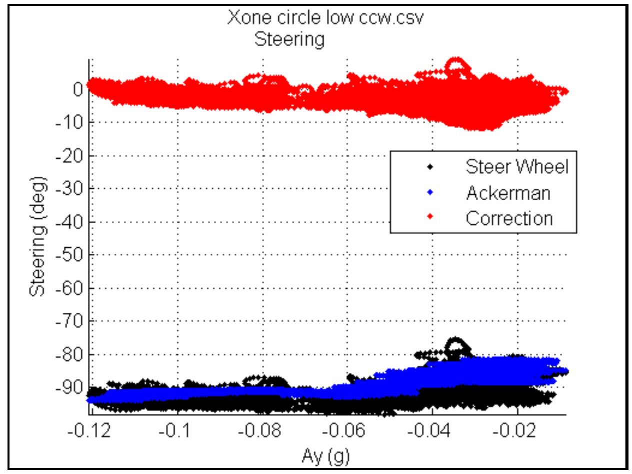


Figure 73. Graph. Understeer Test, NGSWBTS – High-CG Payload- Counter-Clockwise.

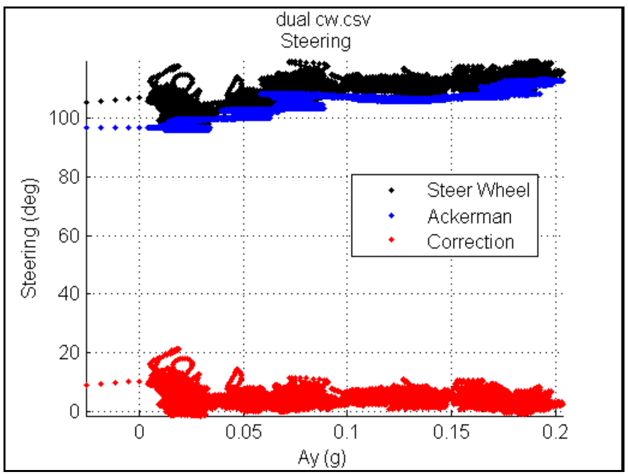


Figure 74. Graph. Understeer Test, Dual Tires – High-CG Payload – Clockwise.

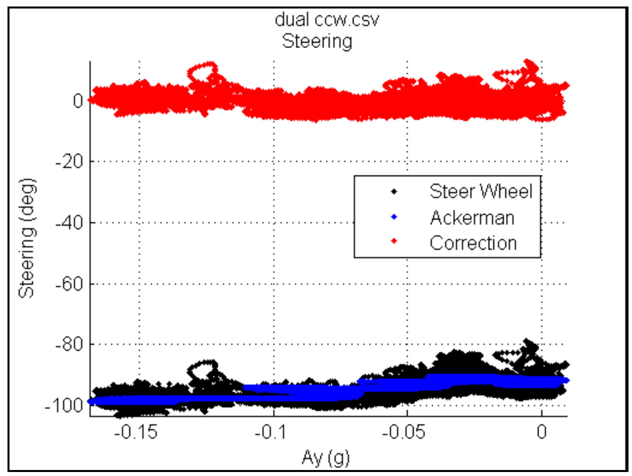


Figure 75. Graph. Understeer Test, Dual Tires – High-CG Payload-Counter-Clockwise.

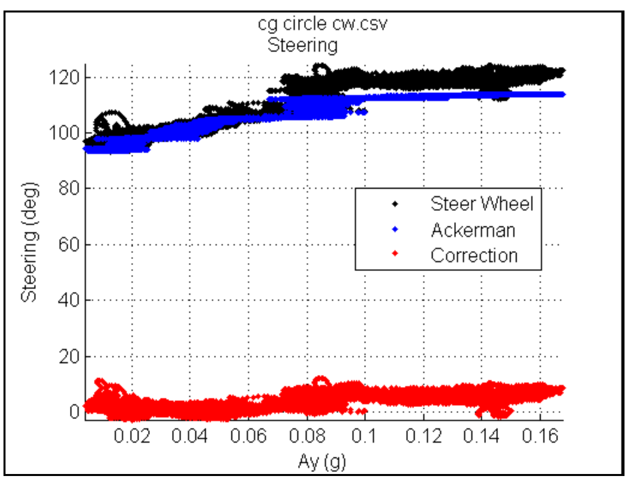


Figure 76. Graph. Understeer Test, Dual Tires – Low-CG Payload- Clockwise.

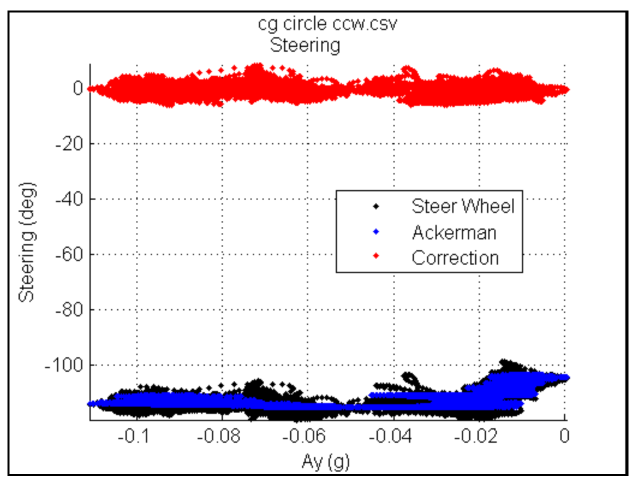


Figure 77. Graph. Understeer Test, Dual Tires – Low-Cg Payload- Counter-Clockwise.

High-Speed Constant Radius Test

The high-speed constant radius test demonstrated the same vehicle path characteristics as that of the low-speed constant radius test with a nearly constant 75 m radius for all speeds. Note that since lateral acceleration is the velocity squared divided by the radius, the path characteristics are not significantly affected by lateral acceleration. Figure 78 shows a trace of one of the dual high-CG payload high-speed circle runs:

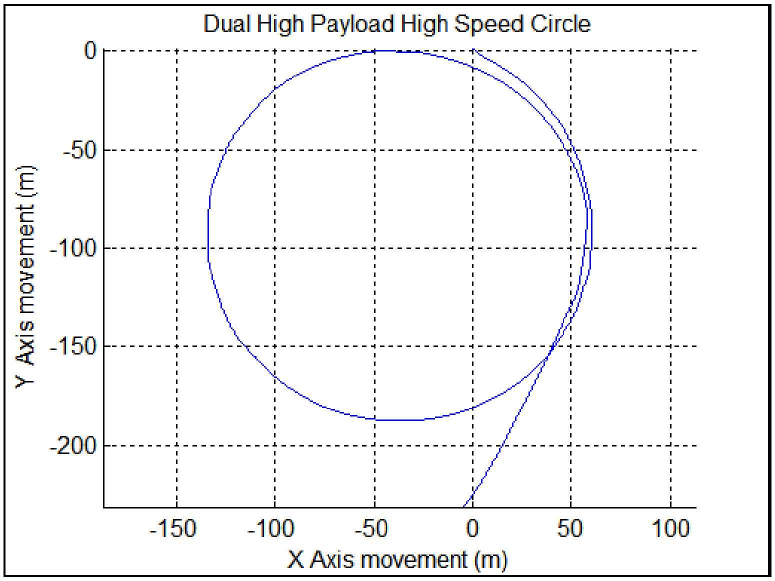


Figure 78. Graph. High-Speed Constant Radius Test Path.

A statistical review of the high-speed circle test data using a Duncan Multiple Range test with least significant ranges (evaluates each parameter individually and keeps only statistically significant parameters based on a significant score) shows that the selection of tires was not statistically significant. The test does show that the reduction of the payload height by 1 ft is significant. This is shown by grouping by letter. Cases with the same grouping are not statistically distinguishable.

Table 9. Constant Radius Test Drive Axle Lift

Tractor Ay Lift			
Duncan Grouping	**Mean**	**N**	**NAME**
A	0.37g	4	Dual Tires Low Load
B	0.31g	4	NGSWBTs High Load
B	0.31g	4	Dual Tires High Load

Table 10. Constant Radius Test Trailer Axle Lift

Trailer Ay Lift			
Duncan Grouping	**Mean**	**N**	**NAME**
A	0.37g	4	Dual Tires Low Load
B	0.33g	4	NGSWBTs High Load
B	0.33g	4	Dual Tires High Load

Also of interest is the fact that the higher CG payload allows the drive axles to lift before the trailer axles, but the lower CG payload has the trailer axles lifting with the drive axles. This indicates that the front and rear portions of the flatbed-trailer have very similar roll limits since the flatbed-trailer is flexible and is not capable of effectively transmitting torque from one end to the other in order to force the lifting of the axles in unison. Note: This would not be true of a rigid chassis such as a tanker where the chassis could effectively pass moments from the front to the rear axle sets.

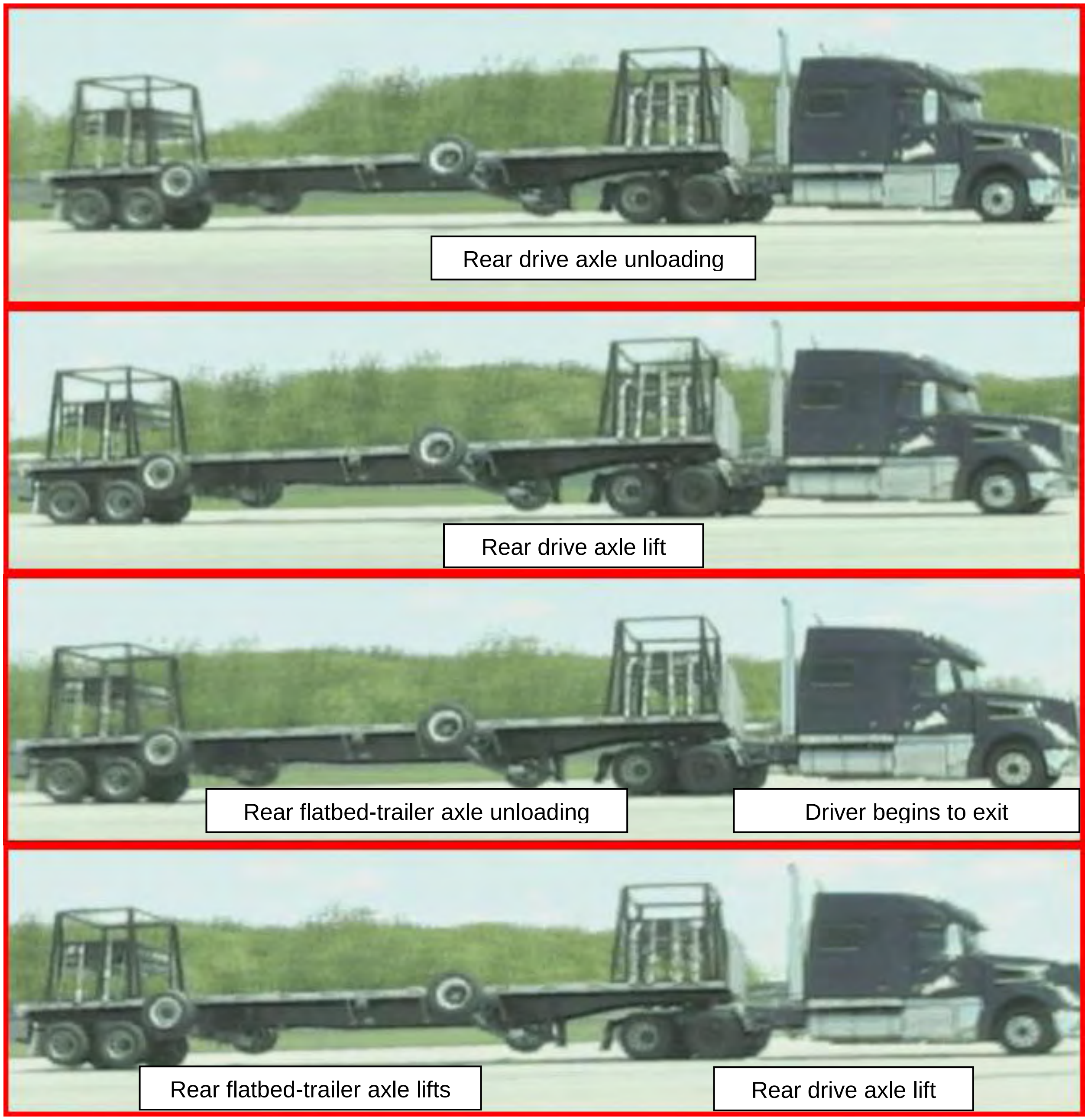


Figure 79. Photograph. High-CG Payload Constant Radius Test – Wheel Lift Sequence.

The photos in figure 79 are from the NGSWBT/high-CG payload testing. Note that the drive axles lift first, then the flatbed-trailer axles. Also, the drive axle lift amplitude was considerably less than that of the flatbed-trailer, but the measurements indicate that the drive axles lifted before the flatbed-trailer axles. Note: in some cases, only one axle from the group lifted, in others both lifted. The lifting of one axle from a group constituted a lift event.

The lift sequence of the high-CG payload cases can be contrasted to the low-CG payload cases where the flatbed-trailer axle lifts just before the drive axle. Again, however, the flatbed-trailer axle lift amplitudes are larger than that of the tractor axles.

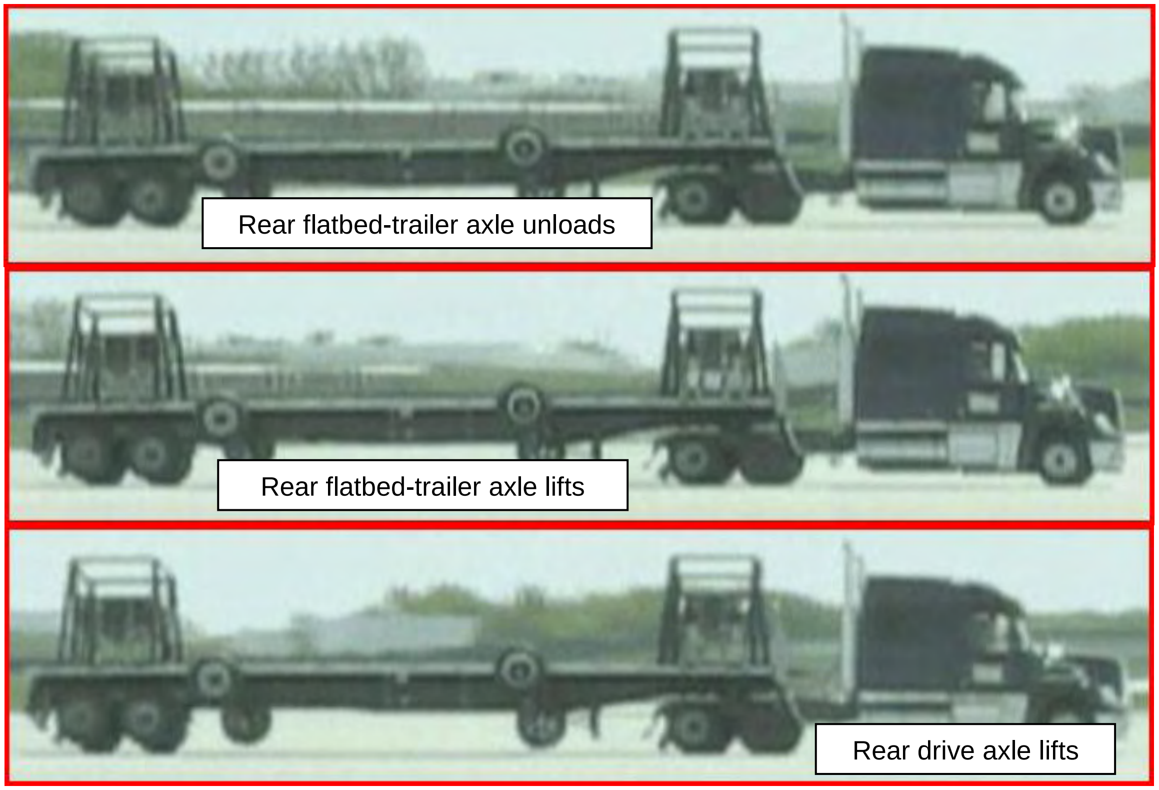


Figure 80. Photograph. Low-CG Payload Constant Radius Test – Wheel Lift Sequence.

The final runs in this test sequence were done with the tractor ESC system engaged. The system engaged on all test combinations at around 0.2 g of lateral acceleration. This is well below the lift threshold of the vehicle.

<u>Step Steer Test</u>

The step steer test was designed to evaluate the transient behavior of the vehicle using a repeatable steering input. The test involved reaching a steady speed and then executing a 170 deg steering input at approximately 170 deg/sec. Testing was performed without the tractor ESC on in order to evaluate the vehicle's stability, and then with the tractor ESC on in order to gage the benefits of the tractor ESC system.

Wheel lift was noted visually and by measurement of the bending strain in the axle. As part of the vehicle instrumentation, strain gages were applied near the wheel hubs on each side of the drive and flatbed-trailer axle housings to indicate wheel lift (see 0). Prior to testing, a calibration test was performed during which the strain was measured with the axles suspended off the ground, providing a zero condition. During the track testing, the axle strain decreased to zero as the wheel lifted, and the axle vertical load decreased to zero.

Because the steering input was constant for each repetition of the maneuver, the lateral acceleration seen by the vehicle was directly proportional to the vehicle speed for each test run. Since rollover threshold is usually defined in terms of lateral acceleration, the lift point was defined as the lateral acceleration level at which any wheel first left the ground. Using lateral acceleration as the lift indicator separates the measurement from the specifics of the maneuver (steering, speed) and provides a more generalized lift metric.

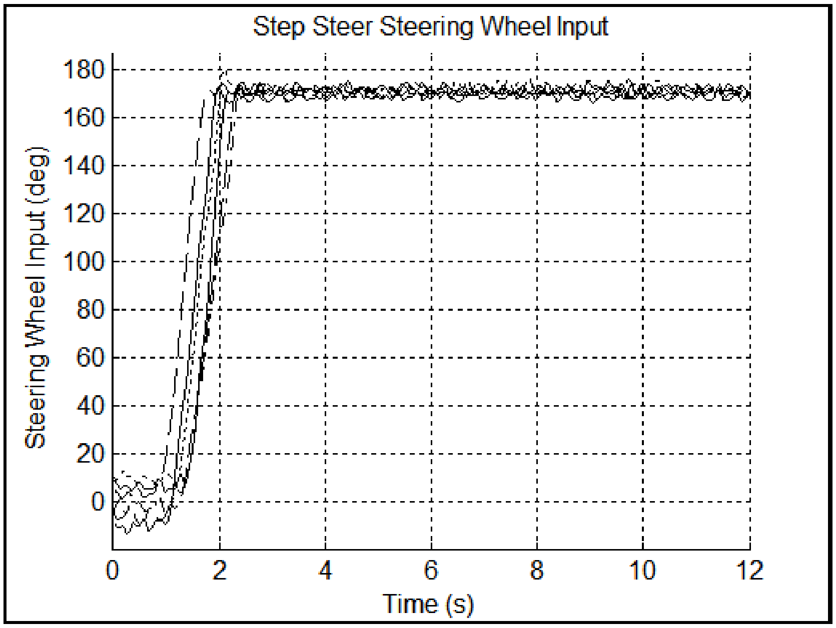


Figure 81. Graph. Step Steer Input.

Step Steer Testing With Tractor ESC Off

The step steer test was performed using progressively increasing speeds until a wheel lift occurred. This set the target speed for the test condition and the test was re-run multiple times with speeds slightly above and below the target speed such that the lift point was bracketed by lift and non-lift events.

The lateral acceleration level recorded at lift was the average of a one-second window about the lift point. Due to dispersion in the testing, the entry speeds for the vehicle were such that the lateral acceleration level needed to lift the vehicle's wheels were, on occasion, exceeded by a significant margin. For these cases, the lateral acceleration level of the vehicle continued to increase after wheel lift rather than maintaining a value just above the wheel lift threshold. The introduction of larger than necessary lateral accelerations has a slight tendency to skew the lift threshold to a higher lift value.

Below are the results for the dual tire testing with the high-CG payload\ height (see figure 82 to figure 85). When a lift event occurs, the strain in the axle housing goes to zero as shown in figure 83.

Several key vehicle metrics were selected as potential lift indicators. These included vehicle velocity, lateral acceleration, roll rate, yaw rate, roll angle, and vehicle side slip angle. Despite the attempts to bracket the lift point, there were too few over-lapping lift and non-lift points to statically determine the lift threshold for the three vehicle configuration cases . Because of this, it was decided to try to determine if the three cases could be statistically separated.

The statistical analysis was performed by considering all of the test runs together with an indicator representing the condition (NGSWBTs/high-CG payload, dual tires/high-CG payload, and dual tires/low-CG payload). Other indicators such as the lateral acceleration, yaw rate, roll rate, roll angle, and vehicle side slip were included in the model to determine if any of these indicators were associated with the lifting of a wheel.

To visualize the vehicle's response, the following plots were used: Vehicle speed, axle strain which indicates wheel lift when the strain goes to zero, lateral acceleration, and axle strain vs. lateral acceleration.

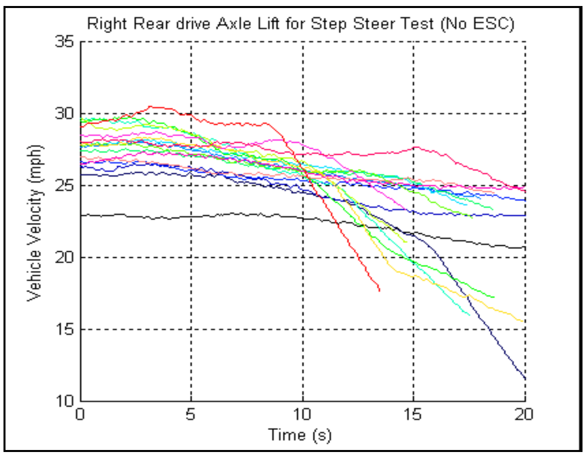


Figure 82. Graph. Dual Tires – High-CG Payload Step Steer Velocity.

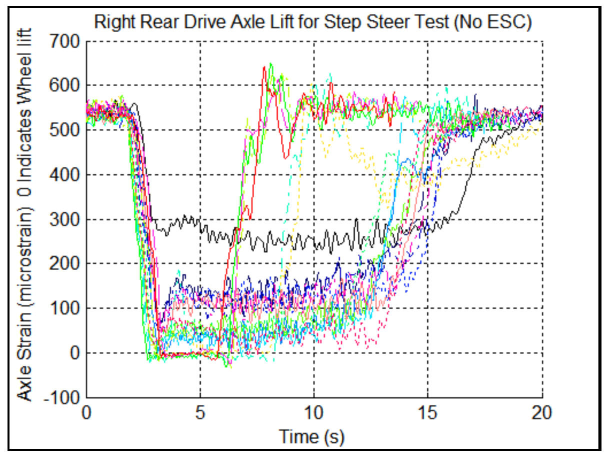


Figure 83. Graph. Dual Tires – High-CG Payload Step Steer Drive Axle Lift Indicator.

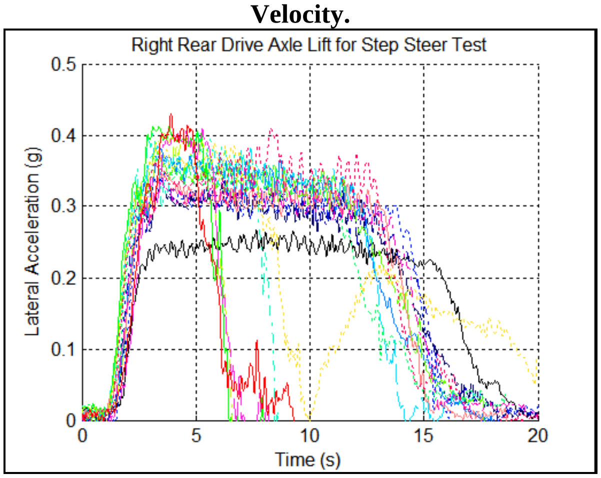


Figure 84. Graph. Dual Tires – High-CG Payload Step Steer Drive Ay.

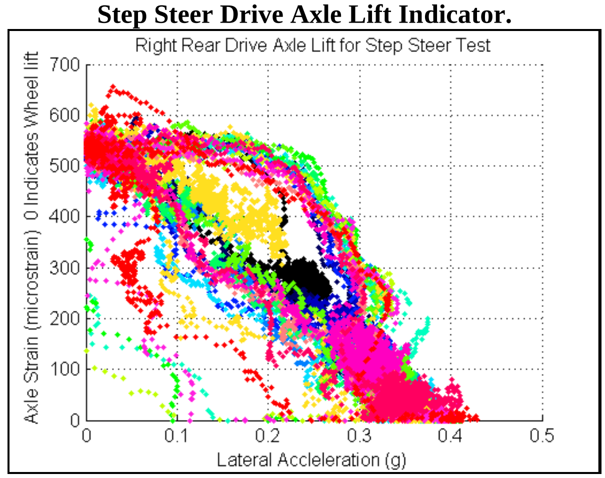


Figure 85. Graph. Dual Tires – High-CG Payload Step Steer Drive Axle Strain.

Note: The colors represent sequential passes through the maneuver

Evaluation of the data was done using logistic regression methods employing a stepwise procedure to determine which, if any, of the indicators predicted the vehicle lift (a significance test). The model evaluated the indicators one at a time to determine if the given indicator correlated with the lift events and if the indicator was correlated to any other indicator. The indicators kept were the strongest correlating indicators that were not correlated to a stronger indicator. An example of a strong indicator not selected was vehicle velocity because it was correlated to lateral acceleration, and the lateral acceleration had a higher correlation to the lift event. The results showed that payload height, and lateral acceleration were well correlated to the lift event. The remaining parameters were not determined to be statistically correlated with the lift event.

The lack of association of the remaining vehicle dynamics parameters with the lifting of the wheel may be a result of dispersion in the measurement, a function of too few sample points, potential correlation between the indicators themselves (which results in the elimination of lesser indicators), or any combination of the above issues.

Table 11. Tractor Lift Probability Estimates

Analysis of Maximum Likelihood Estimates					
Parameter	**Degrees of Freedom**	**Estimate**	**Standard Error**	**Wald Chi-Square**	**Pr > ChiSq**
Intercept	1	-85.6064	27.3203	9.8184	0.0017
ay	1	2.6257	0.8346	9.8976	0.0017
low	1	-13.0519	4.2527	9.4194	0.0021

An analysis of the quality of the statistical fit of the data shows that the model agreed with the data in 99% of the cases.

Table 12. Tractor Step Steer Observed Response Probability

Association of Predicted Probabilities and Observed Responses			
Percent Concordant	99	Somers' D	0.973
Percent Discordant	1.3	Gamma	0.973
Percent Tied	0	Tau-a	0.495
Pairs	675	c	0.987

A similar analysis was also performed on the right rear flatbed-trailer axle to evaluate the flatbed-trailer lift point. The results obtained were similar to the drive results indicating that only lateral acceleration and payload height were significant parameters related to wheel lift. The summary is provided below.

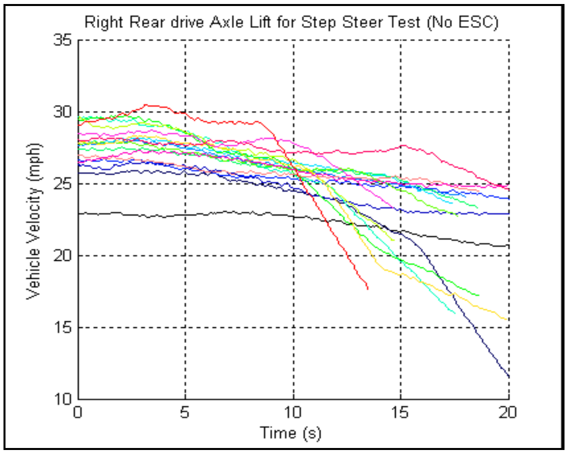


Figure 86. Graph. Dual Tires – High-CG Payload Step Steer Velocity.

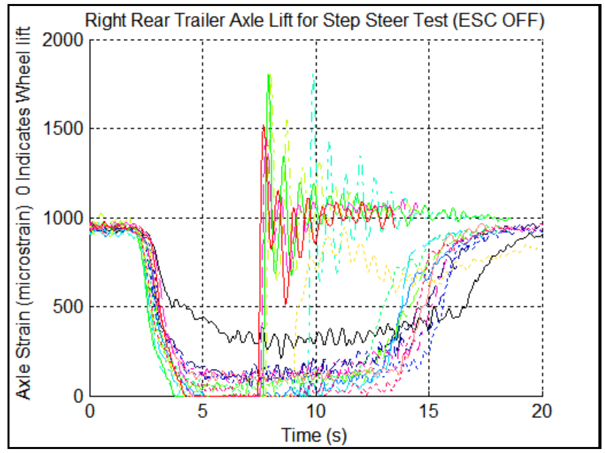


Figure 87. Graph. Dual Tires – High-CG Payload Step Steer Flatbed-Trailer Lift.

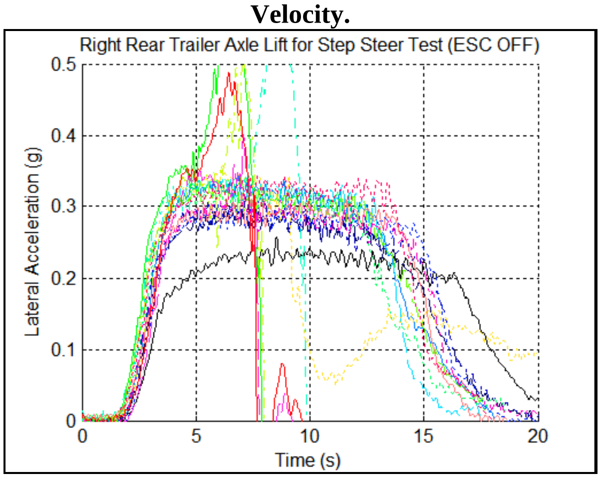


Figure 88. Graph. Dual Tires – High-CG Payload Step Steer Flatbed-Trailer Ay.

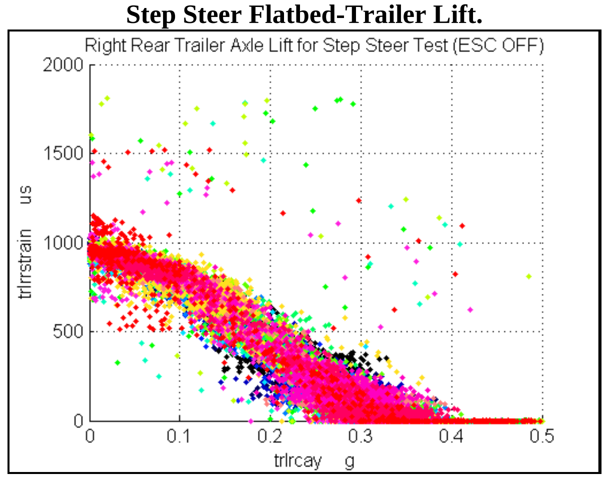


Figure 89. Graph. Dual Tires – High-CG Payload Step Steer Flatbed-Trailer.

Table 13. Trailer Lift Probability Estimates

Analysis of Maximum Likelihood Estimates					
Parameter	**Degrees of Freedom**	**Estimate**	**Standard Error**	**Wald Chi-Square**	**Pr > ChiSq**
Intercept	**1**	**-200.4**	**86.9908**	**5.3061**	**0.0213**
low	**1**	**-16.6453**	**7.0388**	**5.5922**	**0.018**
Tractor Velocity	**1**	**7.3099**	**3.1781**	**5.2902**	**0.0214**

An analysis of the quality of the statistical fit of the data shows that the model agreed with the data in 99% of the cases.

Table 14. Step Steer Observed Response Probability

Association of Predicted Probabilities and Observed Responses			
Percent Concordant	**99**	**Somers' D**	**0.979**
Percent Discordant	**1**	**Gamma**	**0.979**
Percent Tied	**0**	**Tau-a**	**0.499**
Pairs	575	c	0.99

Based on the analysis above it was possible to determine with a 95% confidence that the low-CG payload vehicle response is statistically different from the response of the high-CG payload vehicle for both drive and flatbed-trailer axle sets. It was not possible to distinguish between the behavior of the dual tires and NGSWBTs for either the drive or flatbed-trailer axle sets. The indication here is that dual tires and NGSWBTs have similar performance in roll stability, and that lower CG payload heights improve roll stability.

The statistical analysis above indicates only that the low-CG payload is a significant parameter, but gives no indication of the lateral acceleration lift points for the differing payload and tire cases. However, because the statistical analysis shows that the dual tire/high-CG payload case and the NGSWBT/high-CG payload case are equivalent, they can be treated as having the same lift point. Therefore, the need is to determine an estimation of the lift threshold for the low-CG and high-CG payload cases.

In figure 90 through figure 93, the lateral acceleration of the vehicle is plotted with the non-lift events on the left side (0 on the abscissa) and the lift events on the right side (1 on the abscissa). The lift value is estimated as the minimum lateral acceleration at which a lift event occurred. Note: this is not the true lift threshold because test dispersion has not been addressed in this analysis. The minimum was used as opposed to the average of the measured values due to the above-mentioned skew in the data. The test dispersion also accounts for the apparent difference in minimum Ay for the NGSWBT and dual tire sets. Later analysis will show the effect of the dispersion on the data more clearly and demonstrates that the two cases are truly equivalent and have the same lift point.

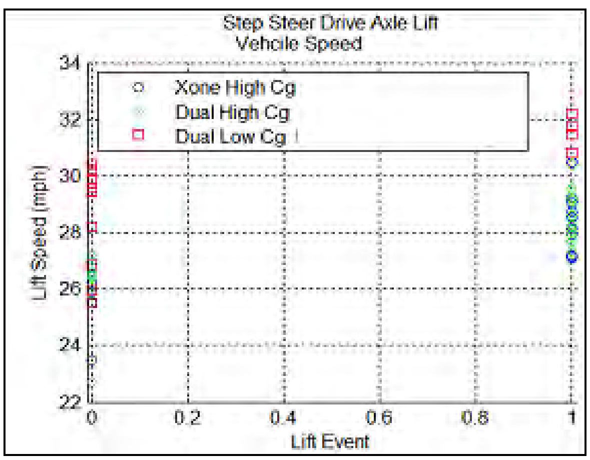


Figure 90. Graph. Drive Axle Step Steer Lift Comparison (Velocity).

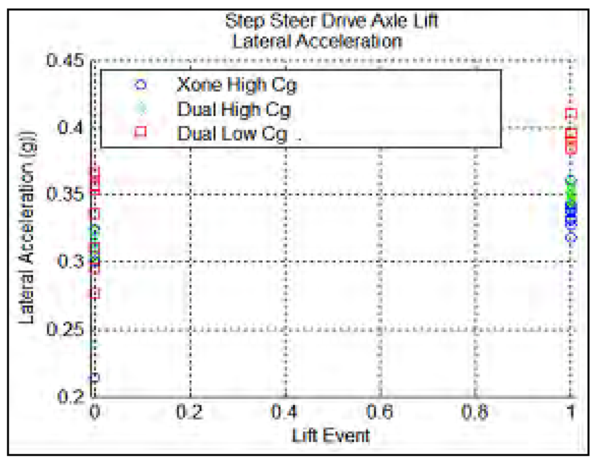


Figure 91. Graph. Drive Axle Step Steer Lift Comparison (Ay).

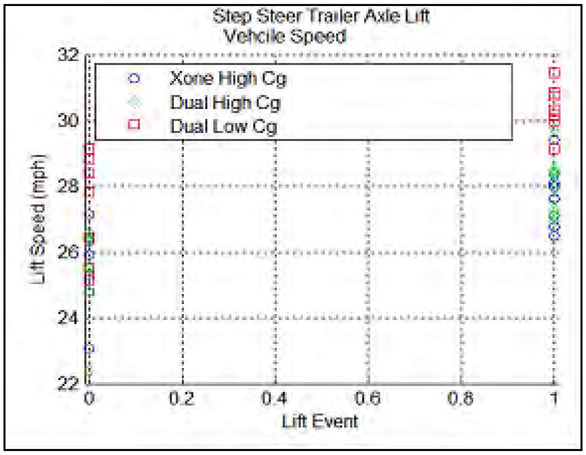


Figure 92. Graph. Flatbed-Trailer Axle Step Steer Lift Comparison (Velocity).

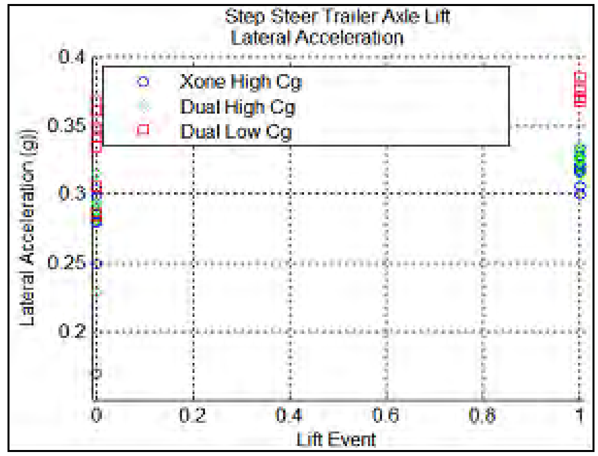


Figure 93. Graph. Flatbed-Trailer Axle Step Steer Lift Comparison (Ay).

Note that the dual tire/high-CG payload and NGSWBT/high-CG payload lift velocity and acceleration values are grouped together and the dual tire/low-CG payload is isolated.

While the data is not sufficiently coherent to identify a statistically sound lift threshold value, the lift point of the vehicle can be estimated from the data above. By observing the lift trends for the differing test cases, it was estimated that the high-CG payload cases (dual tires and NGSWBTs) lifted the drive axle tires around 0.32 g, while the low-CG payload case lifted the drive axle tires around 0.39 g. The flatbed-trailer exhibited lift for the high-CG payload cases around 0.3 g and the low-CG pay payload case lifted around 0.37 g. These results are summarized in table 15.

Table 15. Estimated Step Steer Lift Threshold (Based on Roll-Corrected Lateral Acceleration)

	High-CG Payload Case	Low-CG Payload Case
Drive Axle Lift	0.32 g	0.39 g
Flatbed-Trailer Axle Lift	0.30 g	0.37 g

Step Steer Testing With Tractor ESC On

The step steer test was performed using progressively increasing speeds until a wheel lift occurred. This set the target speed for the test condition, and the test was re-run multiple times with speeds slightly above and below the target speed such that the lift point was bracketed by lift and non-lift events.

Once the lift test speed was established for the vehicle, the step steer tests were repeated with the tractor ESC engaged. The starting velocity for the tractor ESC on cases was the lift threshold determined from the tractor ESC off testing (see previous section). Below are the results from testing of the dual tire case with the higher CG payload, and with the tractor ESC system enabled. For this vehicle configuration, the vehicle's lift threshold was determined to be the same of both the ESC off and ESC on cases.

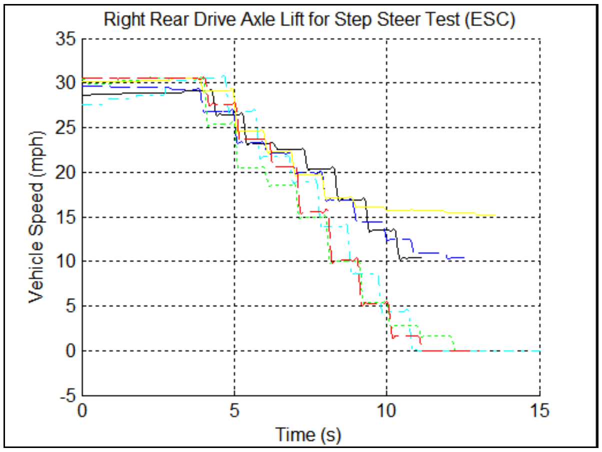


Figure 94. Graph. Dual Tires – High-CG Payload Step Steer Velocity (Tractor Esc On).

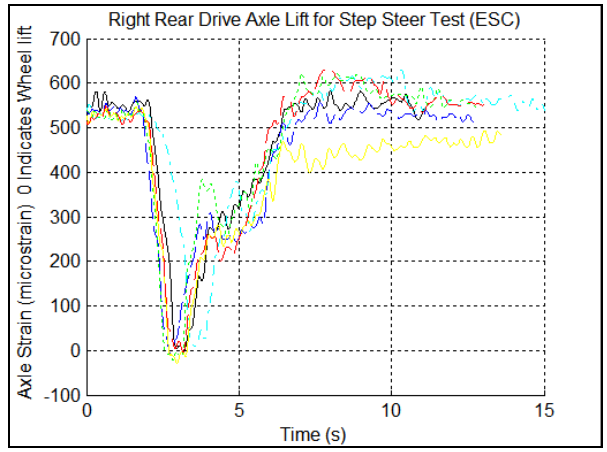


Figure 95. Graph. Dual Tires – High-CG Payload Step Steer Wheel Lift (Tractor Esc On).

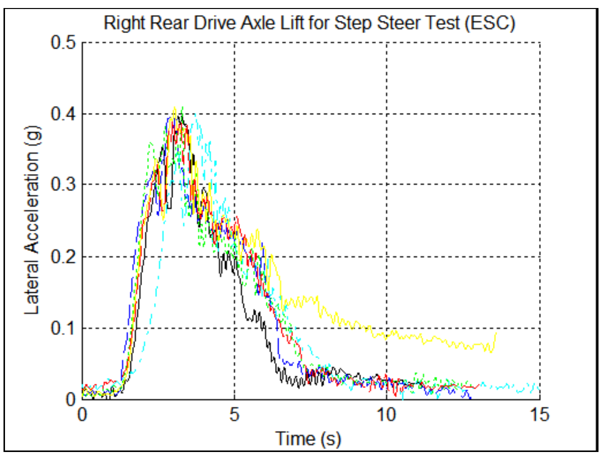


Figure 96. Graph. Dual Tires – High-CG Payload Step Steer Ay (Tractor Esc On).

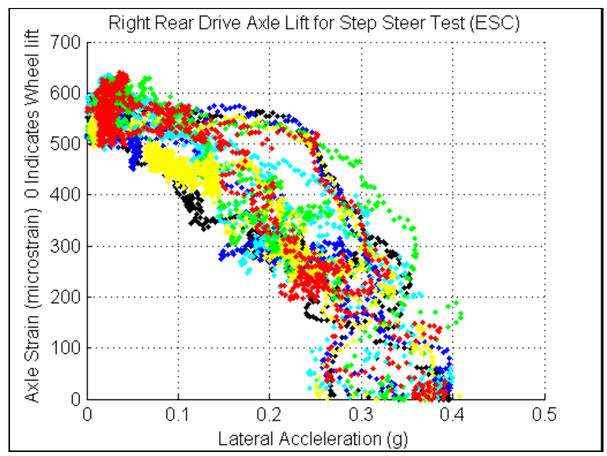


Figure 97. Graph. Dual Tires – High-CG Payload Step Steer Lift Point (Tractor Esc On).

For the dynamic step steer maneuver, axle lift thresholds for speed or lateral acceleration, with the high-CG payload are similar between the tractor ESC on case and the tractor ESC off case. However, the duration of the observed wheel lift is decreased, as can be seen by comparing the tractor ESC off test results, shown in figure 83, to the results of the tractor ESC on case, shown in figure 95. This would make sense as the engagement of the tractor ESC system should tend to bring the vehicle back into a stable operating situation, sooner.

While the wheel lift duration for the high-CG payload cases was decreased, the data cannot be used to quantify the lift duration reduction because the driver's response after a wheel lift occurred varied significantly. Factors that influenced the driver's response included the amplitude of the lift, the notification to abort by the test administrator, and the roll of the flatbed-trailer, among other factors. In some cases, the driver held the steering so that the vehicle traversed a 180 deg arc. There were some cases the test administrator told the driver to abort for safety reasons after seeing the tractor and flatbed-trailer lift. If the driver could see the outriggers touch the ground, he was instructed to slowly exit the maneuver. If the driver felt the vehicle rolling excessively or rolling too quickly, he was instructed to steer out of the maneuver. Likewise, if the vehicle's yaw rate was deemed to be excessive, the driver was instructed to begin steering out of the maneuver. All of these variations in steering after a lift occurred make defining a quantitative value for lift duration infeasible.

For the low-CG payload case, the tractor ESC system was effective in preventing wheel lift under the same conditions that wheel lift was observed in the tractor ESC off testing for this maneuver.

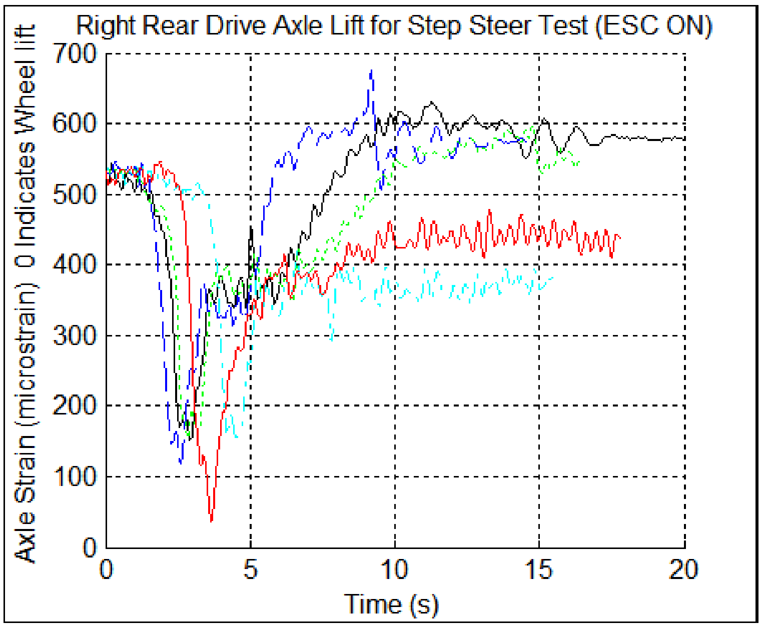


Figure 98. Graph. Step Steer Drive Axle Lift (Tractor ESC On) Dual Tires Low-Payload.

As with the tractor ESC off testing, the lift threshold for each case was plotted to aid assessment of the tractor ESC system's impact on the differing cases. As can be seen in figure 99 and figure 100, both high-CG payload cases are roughly equal, with lift thresholds in the 0.33 g range.

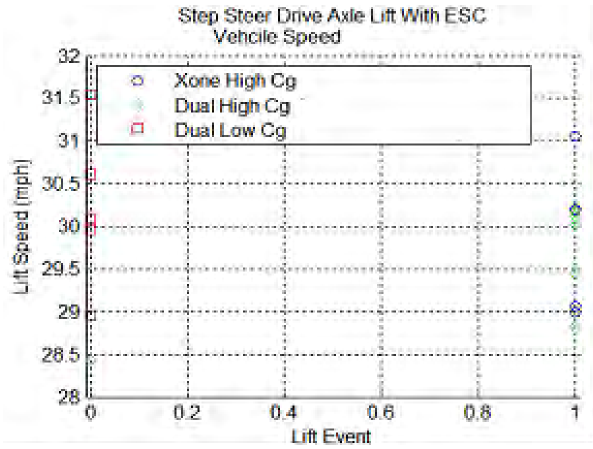


Figure 99. Graph. Step Lift Comparison With Tractor ESC On (Velocity).

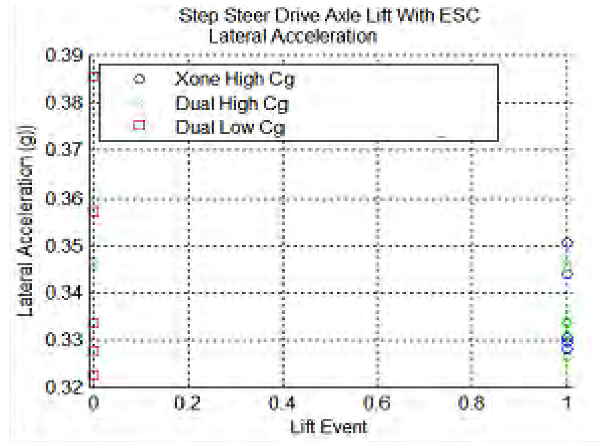


Figure 100. Graph. Step Lift Comparison With Tractor ESC On (Ay).

Note that qualitatively, the lift lateral acceleration for the dual tire high-CG payload case with tractor ESC on is lower than the lift value with tractor ESC off. The difference in roll threshold was deemed to be a product of the test dispersion in the data. This helped to explain the statistical results indicating equivalent behavior between the dual tires and NGSWBTs.

Evasive Maneuver Test

The objective of this test was to assess the vehicle's overall transient response and roll characteristics when subjected to a sudden lane change. The evasive, or lane-change maneuver set-up was similar to the one used in the previous phases of this project. It consisted of two gates delimited by four pylons on a level asphalt area of sufficient size to accommodate safe entry and exit of the course for the size and weight vehicle being tested. Gate 2 was located 100 ft downstream of gate 1, and 12 ft to the left (i.e., gate 1 mimicked a rightmost lane on a three-lane freeway, with gate 2 representing the leftmost lane. The test consisted of negotiating a left-hand lane change by entering gate 1 at a constant speed, making an abrupt steering maneuver upon exiting gate 1, gaining position for straight entry to gate 2, and maintaining the initial speed until exiting gate 2.

Similarly to the other two tests (i.e., the constant radius and step steer maneuvers) the lane maneuver was performed:

- In both a low-speed (LoSp) and a high-speed (HiSp) setting,
- With the payload CG at two different heights, i.e., 9.5 ft above the ground (HiCG) and 8.5 ft above the ground (LoCG), and
- With two different type of tires, i.e., conventional 275/80 R22.5 dual tires on the drive and flatbed-trailer axles for both the HiCG and LoCG cases, and 445/50 R22.5 wide-based single tires (NGSWBTs) for the HiCG case only.

All the high-speed tests were conducted with and without the tractor ESC engaged.

A total of 70 runs were conducted for the lane change maneuver tests. Forty-one runs were conducted with the high-CG payload setting and 29 runs were conducted with the low-CG payload CG setting. Out of the 41 HiCG runs, 23 were conducted with the vehicle equipped with dual tires:

- 13 runs with HiSp, tractor ESC off,
- 3 runs with HiSp, tractor ESC on,
- 7 runs with LoSp tractor ESC off,

and 18 were conducted with NGSWBTs:
- 10 runs with HiSp, tractor ESC off,
- 2 runs with HiSp tractor ESC on, and
- 6 runs with LoSp tractor ESC off.

Ten of the 41 runs produced liftoff events[3]:
- 7 runs with dual tires, HiSp, tractor ESC off,
- 1 run with NGSWBTs, HiSp, tractor ESC off,
- 1 run with dual tires, HiSp, tractor ESC on,
- 1 NGSWBTs, HiSp, tractor ESC On),

and another seven where close-to-liftoff events[4] (i.e., very low traction):
- 4 runs with NGSWBTs, HiSp, tractor ESC off,
- 2 run with dual tires, HiSp, tractor ESC on, and
- 1 run with NGSWBTs, HiSp, tractor ESC on.

All of the 29 LoCG runs were conducted with the vehicle equipped with dual tires:
- 15 runs with HiSp, tractor ESC off,
- 8 runs with HiSp, tractor ESC on; and
- 6 runs with LoSp, tractor ESC off.

Out of the 29 LoCG runs, two runs produced liftoff events (both HiSp, tractor ESC off), and another seven where close to wheel lift events:
- 5 runs with HiSp, tractor ESC off, and
- 2 runs with HiSp, tractor ESC on.

Table 16 presents a summary of the runs for the evasive maneuver by payload CG height, speed, type of tires, and whether the tractor ESC system was engaged or not; showing also those runs with liftoff or close-to-liftoff events. For the runs identified as close-to-liftoff events, the table presents the maximum value reported by the strain gages and the particular run in which this was

3 Wheel liftoff was determined by checking whether the strain-gages mounted on the axles of the tractor and flatbed-trailer showed a null or negative reading.
4 Close-to-wheel-liftoff events were determined by checking whether the strain gages mounted on the axles of the tractor and flatbed trailer showed a null or small positive reading, indicating a loss of traction in the corresponding axle. For the HiCG case, the average values reported by the strain gages were in the 510-580 lb interval for axles 2 and 3, and 925-990 lb for axles 4 and 5 when the vehicle was equipped with Dual Tires; and in the 625-660 lb and 1050-1080 lb intervals for axles 2-3 and 4-5, respectively for NGSWBTs. For the LoCG case, these values were in the 570-600 lb and 935-965 lb intervals for axles 2-3 and 4-5, respectively. Events were considered as being close-to-wheel-liftoff when the readings in the strain gages were less than 20 lb.

observed; all the other runs listed showed strain gage values that were smaller than the maximum. Figure 101 shows the trajectory of two of the runs (runs 8 and 10) of the HiCG/HiSp/tractor-ESC-off/dual tires case (Note: the trajectories have been slightly displaced to avoid overlapping in the graph).

Table 16. Summary of Evasive Maneuver Runs

<table>
<tr><th colspan="3" rowspan="2"></th><th colspan="2">Tractor Esc Off</th><th colspan="2">Tractor Esc On</th><th rowspan="2">Row Total</th></tr>
<tr><th>Dual Tires</th><th>NGSWBTs</th><th>Dual Tires</th><th>NGSWBTs</th></tr>
<tr><td rowspan="6">High-CG</td><td rowspan="3">High-Speed</td><td>Total</td><td>13</td><td>10</td><td>3</td><td>2</td><td>28</td></tr>
<tr><td>Liftoff</td><td>7 (runs 1, 5, 7, 8, 9, 10, 12)</td><td>1 (run 5)</td><td>1 (run 1)</td><td>1 (run 1)</td><td>10</td></tr>
<tr><td>Close to Wheel Lift Events</td><td>0</td><td>4 (runs 1, 2, 7, 10; max = 18.33 us, run 7)</td><td>2 (runs 2, 3; max = 9.14 us , run 3)</td><td>1 (run 2; max = 19.2 us)</td><td>7</td></tr>
<tr><td rowspan="3">Low-Speed</td><td>Total</td><td>7</td><td>6</td><td>N/A</td><td>N/A</td><td>13</td></tr>
<tr><td>Liftoff</td><td>0</td><td>0</td><td>N/A</td><td>N/A</td><td>0</td></tr>
<tr><td>Close to Wheel Lift Events</td><td>0</td><td>0</td><td>N/A</td><td>N/A</td><td>0</td></tr>
<tr><td rowspan="6">Low-CG</td><td rowspan="3">High-Speed</td><td>Total</td><td>15</td><td>N/A</td><td>8</td><td>N/A</td><td>23</td></tr>
<tr><td>Liftoff</td><td>2 (runs 4, 12)</td><td>N/A</td><td>0</td><td>N/A</td><td>2</td></tr>
<tr><td>Close to Wheel Lift Events</td><td>5 (runs 1, 5, 10, 13, 14; max = 12.14 us, run 10)</td><td>N/A</td><td>2 (runs 3, 7; max = 19.2 us, run 7)</td><td>N/A</td><td>7</td></tr>
<tr><td rowspan="3">Low-Speed</td><td>Total</td><td>6</td><td>N/A</td><td>N/A</td><td>N/A</td><td>6</td></tr>
<tr><td>Liftoff</td><td>0</td><td>N/A</td><td>N/A</td><td>N/A</td><td>0</td></tr>
<tr><td>Close to Wheel Lift Events</td><td>0</td><td>N/A</td><td>N/A</td><td>N/A</td><td>0</td></tr>
<tr><td colspan="2" rowspan="3">Column Total</td><td>Total</td><td>41</td><td>16</td><td>11</td><td>2</td><td>70</td></tr>
<tr><td>Liftoff</td><td>9</td><td>1</td><td>1</td><td>1</td><td>12</td></tr>
<tr><td>Close to Wheel Lift Events</td><td>5</td><td>4</td><td>4</td><td>1</td><td>14</td></tr>
</table>

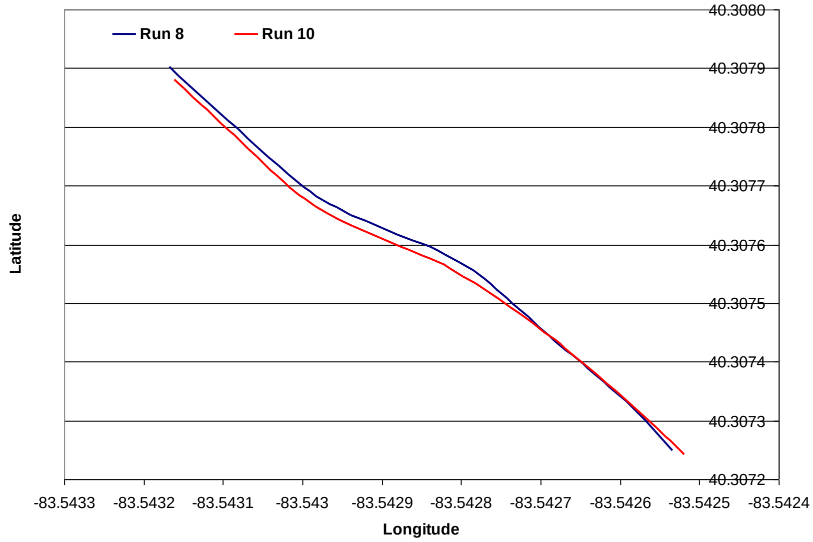


Figure 101. Graph. Trajectories for Runs 8 and 10 (HiCG/HiSp/Tractor ESC Off/Dual Tires Case).

In order to study the stability of each of the configurations tested in this phase of the project for the lane change maneuver, the maximum roll angle registered in each test and the maximum lateral acceleration that caused that roll angle were identified. As was the case with the box-trailer (previous phases of the project), there was a time difference between the instant at which the maximum lateral acceleration (in absolute value) occurred, and the moment at which the maximum (in absolute value) roll angle was registered. In almost all of the cases (96% of all of the cases for the first part of the maneuver, and 82% of all of the cases for the second half of the maneuver), the sequence of events was as follows: 1) tractor maximum lateral acceleration, 2) tractor maximum roll angle and trailer roll angle 1 (front of the flatbed-trailer), and 3) flatbed-trailer maximum lateral acceleration. For the rear of the trailer, the sequence of events varied with the speed and the payload CG height, mostly due to the low torsional stiffness of the trailer.

This time lag is illustrated for the tractor in figure 102. During the first part of the test, the tractor-flatbed-trailer moved in a straight line (i.e., constant lateral acceleration, constant vertical forces, and insignificant roll angle). At about 1.3 sec, the driver started to turn the vehicle to the left and then began to turn to the right at about 2.6 sec. At 2.62 sec, the maximum lateral acceleration of the first half of the maneuver for this run was registered (the driver gave a sudden jolt to the steering wheel to make corrections to the truck path and guide it through the second gate of this maneuver). This sudden jolt produced a liftoff of the third axle at 2.89 sec and also resulted in the maximum (in absolute value) observed roll angle for this part of the maneuver at 3.16 sec (i.e., 0.54 sec after the maximum lateral acceleration was observed). During the second half of the maneuver, axle 3 lifted at about 4.25 sec, the maximum lateral acceleration occurred at 4.53 sec, and the maximum roll angle occurred at 4.67 sec, showing a time lag of 0.14 sec.

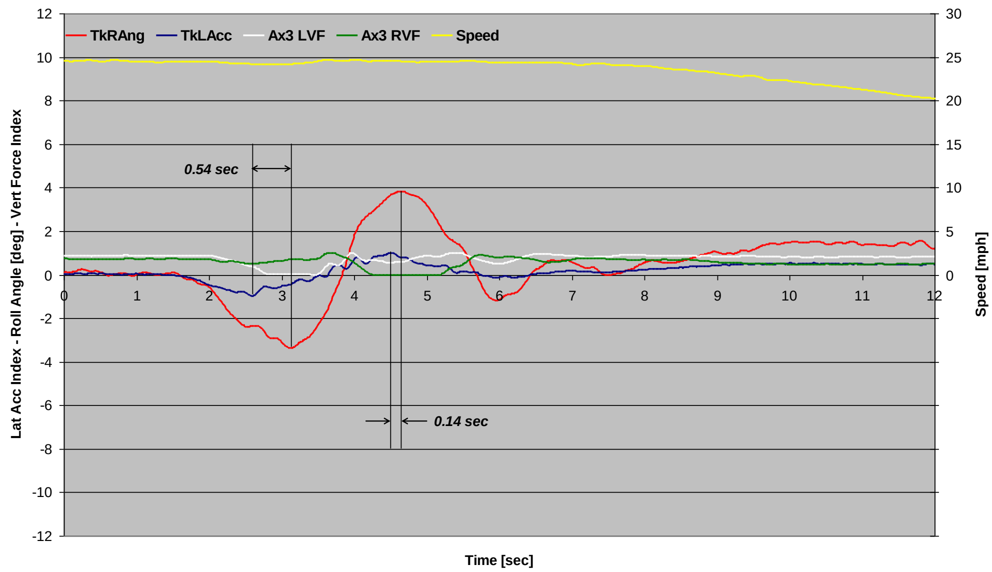


Figure 102. Graph. Time Lag Between the Tractor Maximum Lateral Acceleration and the Tractor Maximum Roll Angle (Run 7 – HiCG/HiSp/Tractor ESC Off/Dual Tires Case).

In order to compare the results of the lane change maneuver tests, the variable ***RS*** (maximum roll stiffness, in units of deg/g) was generated by computing the ratio between the maximum roll angle and the maximum lateral acceleration; which tries to capture the vehicle roll stability of each of the configurations studied. For each run, the variable ***RS*** was computed for both the first (left steering wheel turn) and second part of the maneuver (right steering wheel turn), for the tractor and the front and rear of the flatbed-trailer. Since in the majority of the cases studied (more than 85%) the distributions of the variable ***RS*** had very dissimilar variances for the different treatments analyzed, a Smith-Satterthwaite Test [12] was used to test the null hypothesis that the means of each pair of compared distributions of ***RS*** (e.g., the ***RS*** distribution for dual tires vs. the ***RS*** distribution for NGSWBTs, for high-CG payload and high-speed) were the same, against the alternative hypothesis that these means were different. A rejection of the null hypothesis with a high level of confidence (greater than 99%) would indicate that the treatment studied (e.g., type of tires for high-CG payload and high-speed) had an effect in the distribution of the variable ***RS,*** and in consequence, in the roll stability of the vehicle.

The distributions of the maximum lateral accelerations were also compared to determine whether or not the maneuvers were performed in a sufficiently similar way by the driver for the treatment studied. Again, the hypothesis that the means of the maximum lateral acceleration distributions were the same was tested. A rejection of this hypothesis would indicate that the distributions of the maximum lateral accelerations were different and the maneuvers were not performed in a similar way for the two treatment studied (e.g., dual tires vs. NGSWBTs, for high-CG payload and high-speed). As expected, the null hypothesis was rejected with almost 100 % (more than 99+%) confidence for the cases in which high-speed maneuvers were compared against low-speed maneuvers (all other treatments being equal), for both the first and second part of the lane-change maneuver. The null hypothesis of similar distribution of maximum lateral accelerations could also be rejected, although it was rejected at a much lower confidence level (i.e., greater than 90%) for comparisons between high- and low-CG payload heights with tractor ESC off. In all other cases, the null hypothesis of similar distributions of maximum lateral accelerations could not be rejected, thus indicating that the maneuvers were performed in a similar way by the driver.

Evasive Maneuver with Tractor ESC Off

Fifty-seven runs were conducted with the tractor ESC system not engaged. The information collected in these runs was used to study the effects that the type of tires and payload CG height had on the roll stability of the vehicle during an evasive maneuver.

In order to study the effect of the different type of tires in vehicle stability, the HiCG/HiSp runs were selected to generate the ***RS*** variable for the tractor, as well as the front and rear of the flatbed-trailer, for both the first and second half of the maneuver. Figure 103 shows the flatbed-trailer lateral acceleration, roll angle at the front of the flatbed-trailer, strain gage readings (a proxy for axle vertical force) for the third axle, and speed as a function of time, for the same run shown in figure 102 (i.e., Run 7 - HiCG/HiSp/tractor ESC off/ dual tires Case). Figure 104 shows the same information as figure 103, but for the rear end of the flatbed trailer. Notice that the flatbed-trailer's maximum front roll angle is several times larger than the maximum roll angle at the end, indicating a low torsional stiffness.

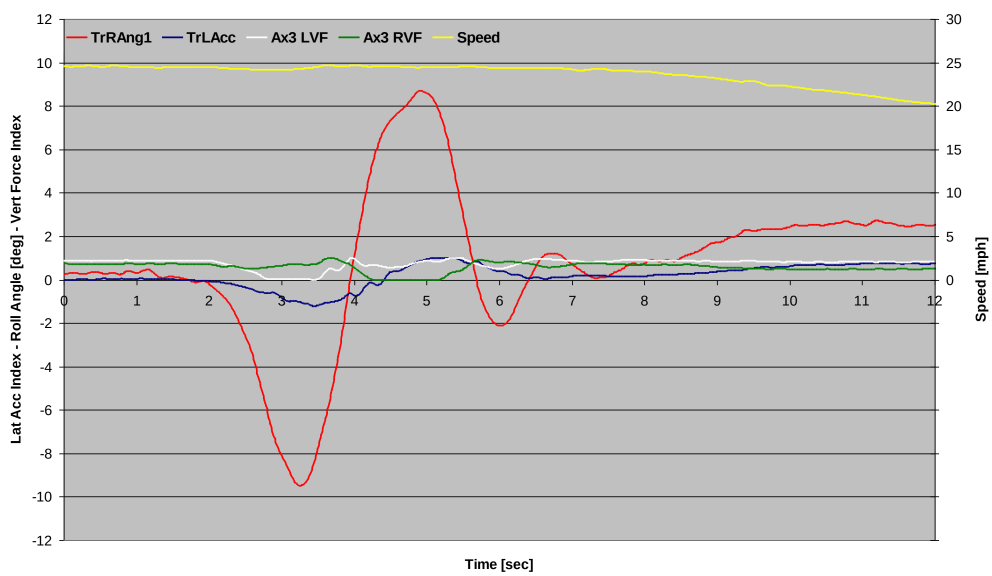


Figure 103. Graph. Flatbed-Trailer Lateral Acceleration, Front Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 7 – HiCG/HiSp/Tractor ESC Off/Dual Tires Case).

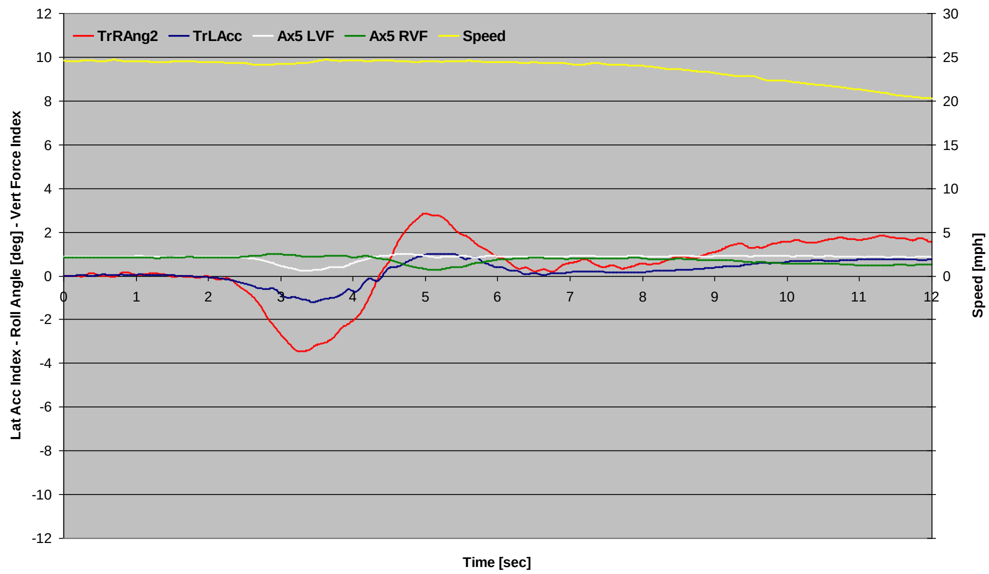


Figure 104. Graph. Flatbed-Trailer Lateral Acceleration, Rear Roll Angle, Axle 5 Vertical Force, and Speed vs. Time (Run 7 – HiCG/HiSp/Tractor ESC Off/Dual Tires Case).

Similar behavior can be observed in figure 105 to figure 107, which show the same information as figure 102 to figure 104, but for a run in which the tractor-flatbed-trailer was equipped with NGSWBTs. Notice also that the speed of the tractor-flatbed-trailer remains more-or-less constant during the entire maneuver and only begins to decrease after the driver has exited gate 2.

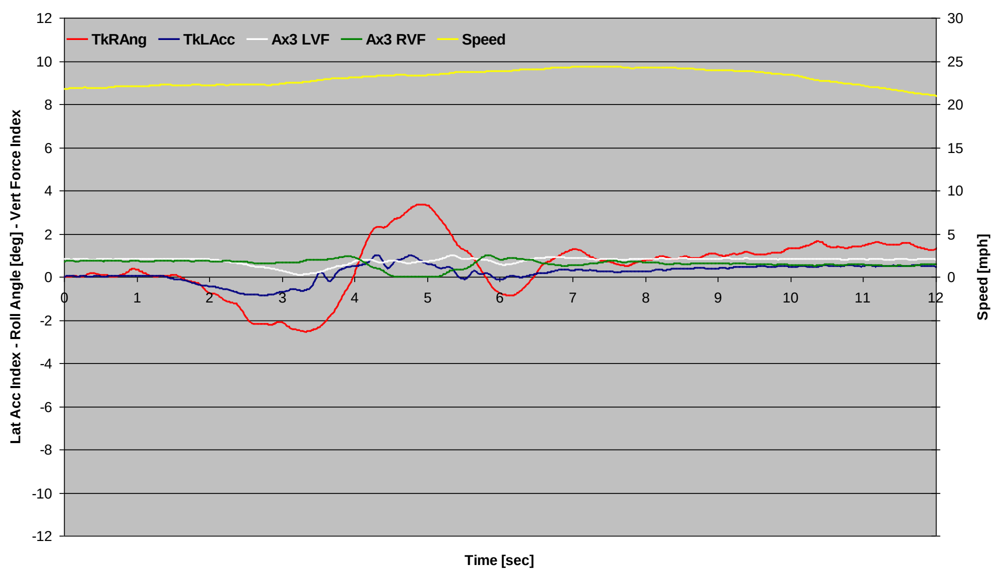


Figure 105. Graph. Tractor Lateral Acceleration, Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 5 – HiCG/HiSp/Tractor ESC Off/NGSWBTs Case).

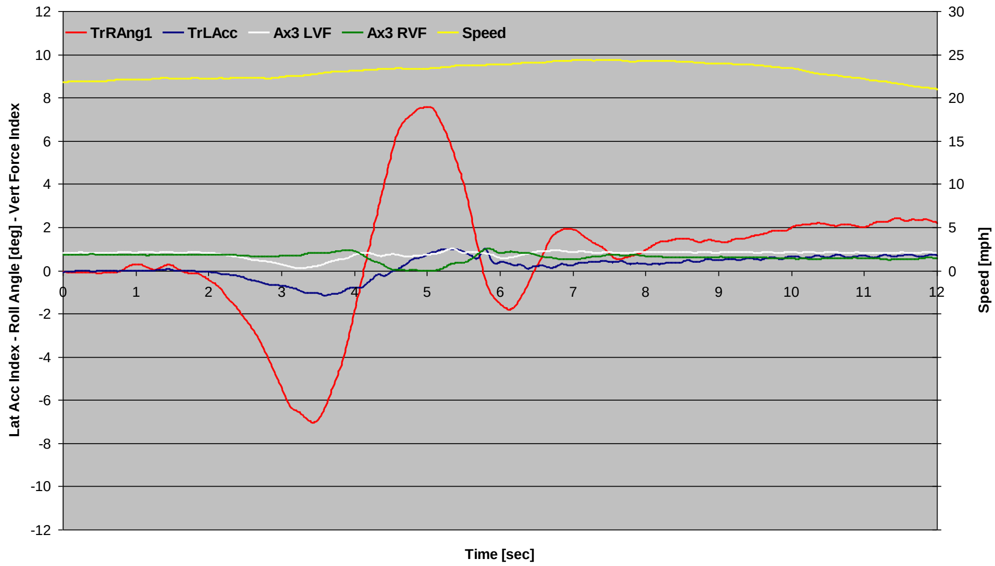


Figure 106. Graph. Flatbed-Trailer Lateral Acceleration, Front Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 5 – HiCG/HiSp/Tractor ESC Off/NGSWBTs Case).

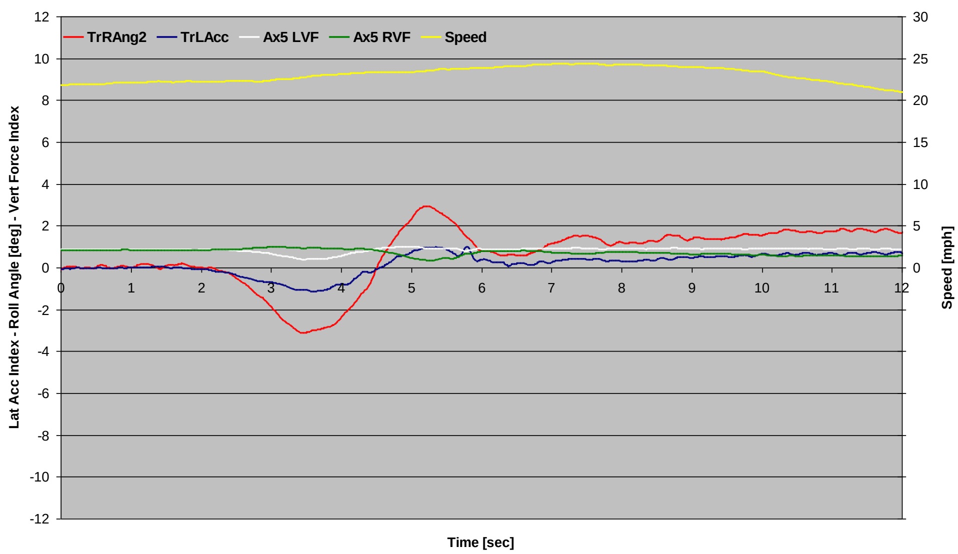


Figure 107. Graph. Flatbed-Trailer Lateral Acceleration, Rear Roll Angle, Axle 5 Vertical Force, and Speed vs. Time (Run 5- HiCG/HiSp/Tractor ESC Off/NGSWBTs Case).

As described above, a Smith-Satterthwaite Test was used to test the null hypothesis that the means of the ***RS*** distribution for the dual tires vs. the ***RS*** distribution for the NGSWBTs, for high-CG payload and high-speed, were the same; against the alternative hypothesis that these means were different. Table 17 shows the results of the test. The table is divided into two parts covering the first and second half of the lane change maneuver. In each part, the three main columns show the variable ***RS*** for the tractor (***TkRS***), the front of the flatbed-trailer (***TrRS1***) and the back of the flatbed-trailer (***TrRS2***), for both the runs in which the vehicle was equipped with dual tires and with NGSWBTs. For the first half of the maneuver, the null hypothesis could not be rejected with a high level of confidence (only at 82%-92% confidence level), thus indicating that the type of tires did not make any difference in terms of roll stability. For the second half of the maneuver, the null hypothesis could not be rejected for the tractor's ***RS***, but it could be rejected at 98% and 99.6% confidence levels for the front and rear of the flatbed-trailer, respectively. For the second portion of the lane change maneuver with this high-CG payload configuration, the results seem to indicate that the roll stability was higher in the case in which the tractor-flatbed-trailer was equipped with dual tires than when it was using NGSWBTs (98% and 99.7% confidence level for ***TrRS1*** and ***TrRS2***). Notice, however, that the flatbed-trailer's low torsional stiffness could have played a large role on the ***RS*** variable for axle 5, more so than for axle 3.

Table 17. Comparison of Dual Tires vs. NGSWBTs *RS* (HiCG/HiSp/ESC Off Case)

	1st Part of the Lane Change Maneuver						2nd Part of the Lane Change Maneuver					
	TkRS		TrRS1		TrRS2		TkRS		TrRS1		TrRS2	
	D*	W**	D*	W**	D*	W**	D*	W**	D*	W**	D*	W**
Mean	8.15	9.13	33.10	31.27	13.08	13.67	7.89	8.58	37.45	42.38	11.98	14.74
Variance	0.15	1.06	3.59	1.76	0.25	0.57	0.85	1.95	4.60	9.22	0.29	1.20
Count	7	5	7	5	7	5	7	5	7	5	7	5
v (degrees of freedom)	5		10		6		6		7		5	
Computed t' Value	2.0308		1.9667		1.5429		0.9590		3.1132		5.1951	
Reject at Confidence Level =	90.20%		92.24%		82.62%		62.54%		98.30%		99.65%	

* D = Dual Tires; ** W = NGSWBTs

As described previously, the lane change maneuver was also conducted with a lower CG payload (i.e., 8.5 ft above the ground), with the vehicle equipped with dual tires. Figure 108 to figure 110 show the vehicle dynamics information used to compute the ***RS*** variable for run 12 of the lane change maneuver with LoCG, high-speed, tractor ESC off tests. Notice that, as expected, the maximum roll angle and lateral acceleration are lower than in the high- CG payload case (figure 102 to figure 104).

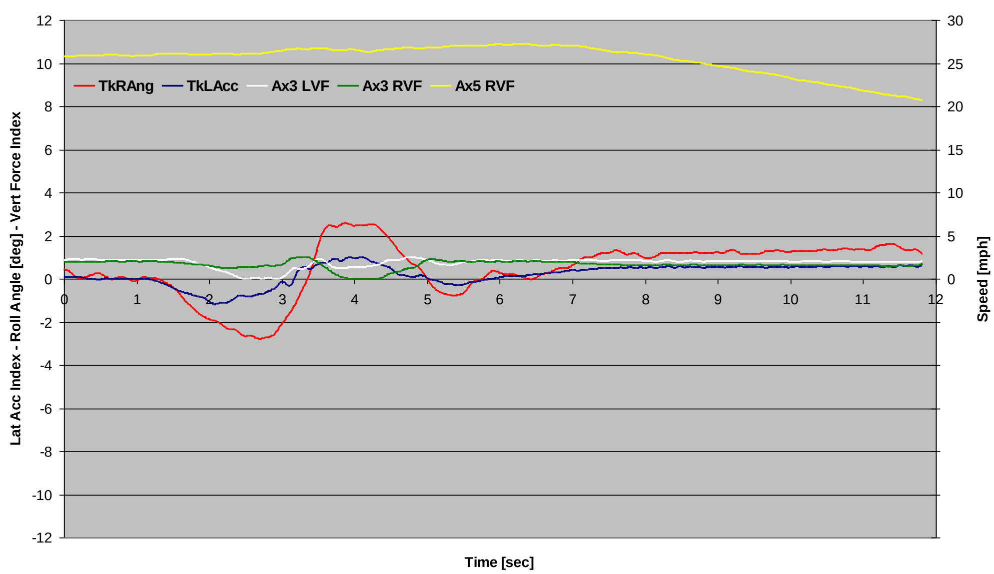


Figure 108. Graph. Tractor Lateral Acceleration, Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 12 – LoCG/HiSp/Tractor ESC Off/Dual Tires Case).

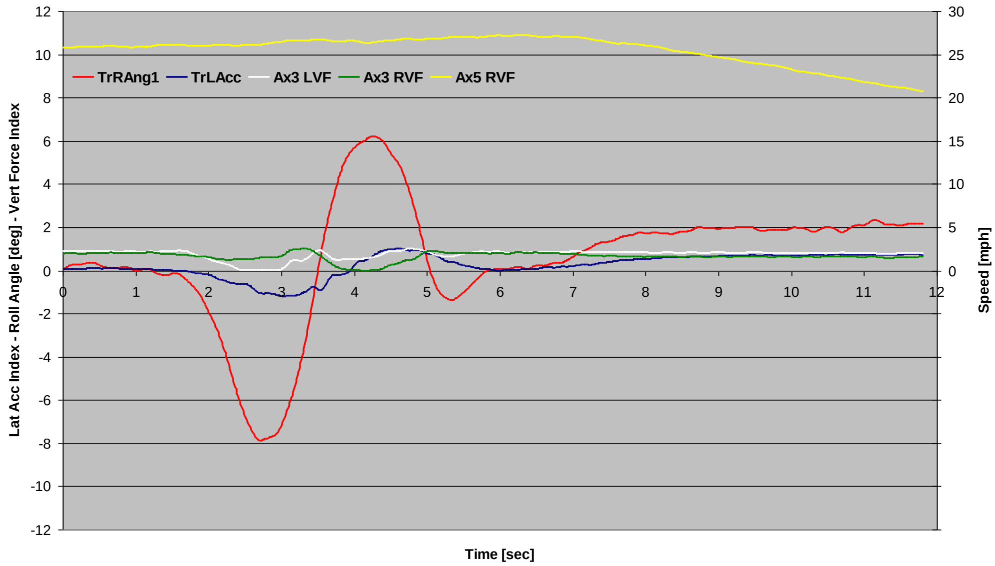


Figure 109. Graph. Flatbed-Trailer Lateral Acceleration, Front Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 12 – LoCG/HiSp/Tractor ESC Off/Dual Tires Case).

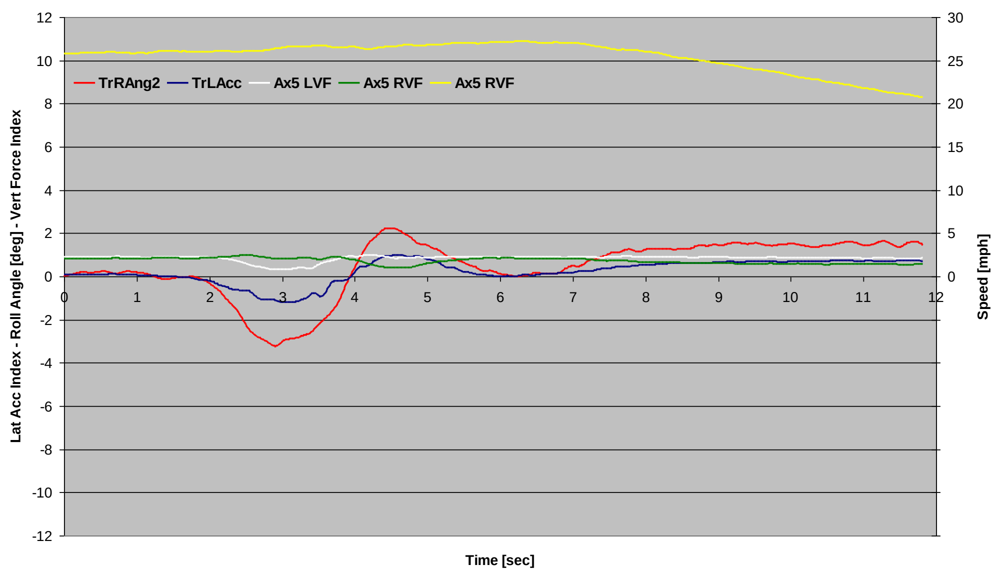


Figure 110. Graph. Flatbed-Trailer Lateral Acceleration, Rear Roll Angle, Axle 5 Vertical Force, and Speed vs. Time (Run 12 – LoCG/HiSp/Tractor ESC Off/Dual Tires Case).

The results of the statistical tests comparing the ***RS*** variable for the low- CG payload versus the high-CG payload (tractor ESC off, high-speed, dual tires) case are presented in table 18. In all of the cases analyzed (i.e., the tractor and flatbed-trailer ***RS*** for both the first and second half of the maneuver), the null hypothesis could be rejected at a 99.99% confidence level, thus clearly indicating that the CG height makes a difference (i.e., increase) in the roll stability of the vehicle during an evasive maneuver. A similar analysis, conducted for the low-speed case, did not show (as expected) such a strong difference in roll stability due to CG height variation. In that case, the null hypothesis could only be rejected at 99.9%, only for the case of the rear of the flatbed-trailer during the second half of the maneuver.

Table 18. Comparison of High-Payload CG vs. Low-Payload CG *RS* (Dual Tires/HiSp/Tractor ESC Off Case)

	1st Part of the Lane Change Maneuver						2nd Part of the Lane Change Maneuver					
	TkRS		TrRS1		TrRS2		TkRS		TrRS1		TrRS2	
	HiCG	LoCG	HiCG	LoCG	HiCG	LoCG	HiCG	LoCG	HiCG	LoCG	HiCG	LoCG
Mean	8.15	6.81	33.10	27.26	13.08	11.33	7.89	6.61	37.45	25.28	11.98	9.51
Variance	0.15	0.24	3.59	1.42	0.25	0.12	0.85	1.06	4.60	7.96	0.29	0.23
Count	7	7	7	7	7	7	7	7	7	7	7	7
v (degrees of freedom)	**11**		**10**		**11**		**12**		**11**		**12**	
Computed t' Value	**5.7147**		**6.9097**		**7.6147**		**2.4355**		**9.0845**		**9.0062**	
Reject at Confidence Level =	**99.99%**		**100.00%**		**100.00%**		**96.86%**		**100.00%**		**100.00%**	

Evasive Maneuver with Tractor ESC On

A total of 13 runs were conducted with the tractor ESC system engaged. These runs were used to study the effects that the type of tires and CG height had on the roll stability of the vehicle during an evasive maneuver, when the tractor ESC system was engaged (Note: all of the tractor ESC on tests were conducted at high-speed).

The effect that the different type of tires has on the vehicle stability was studied using the information collected during the HiCG runs. As in the previous subsection, the collected information was used to generate the ***RS*** variable for the tractor as well as the front and rear of the flatbed-trailer, for both the first and second half of the maneuver. As an illustration of the data collected, figure 111 to figure 116 show the vehicle dynamics information used to compute the ***RS*** variable for run 12 of the lane change maneuver with HiCG and dual tires test (figure 111 to figure 113), and for run 12 of the lane change maneuver with HiCG and NGSWBTs test (figure 114 to figure 114). Notice that in the second half of the maneuver, when the tractor ESC system has had time to engage, the maximum roll angle and maximum acceleration are smaller than in the first half of the maneuver (note that with the tractor ESC off, the roll angle and acceleration were almost symmetrical for the first and second half of the maneuver). This is particularly noticeable for the flatbed-trailer.

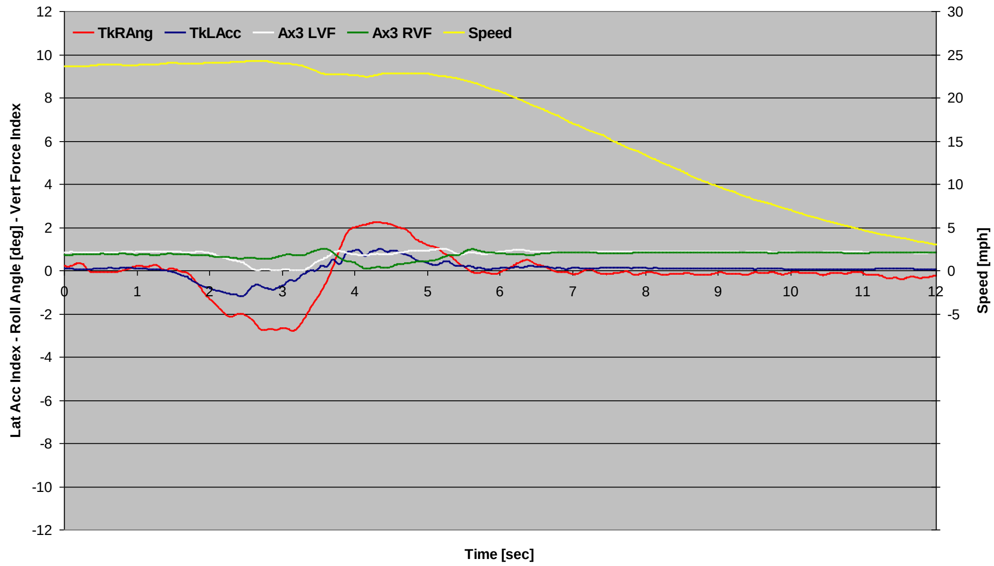


Figure 111. Graph. Flatbed-Tractor Lateral Acceleration, Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 1 – HiCG/HiSp/Tractor ESC On/Dual Tires Case).

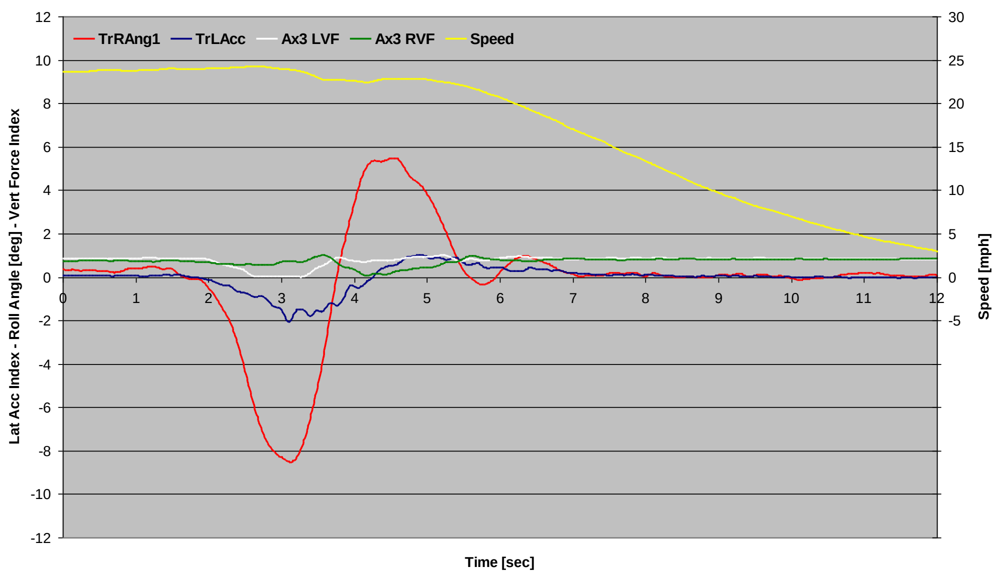


Figure 112. Graph. Flatbed-Trailer Lateral Acceleration, Front Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 1 – HiCG/HiSp/Tractor ESC On/Dual Tires Case).

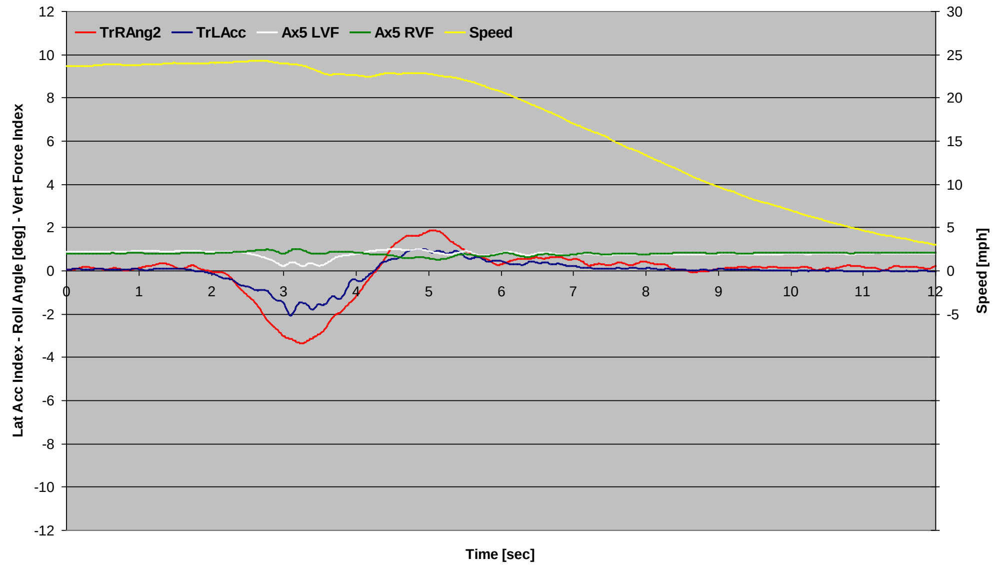


Figure 113. Graph. Flatbed-Trailer Lateral Acceleration, Rear Roll Angle, Axle 5 Vertical Force, and Speed vs. Time (Run 1 – HiCG/HiSp/Tractor ESC On/Dual Tires Case).

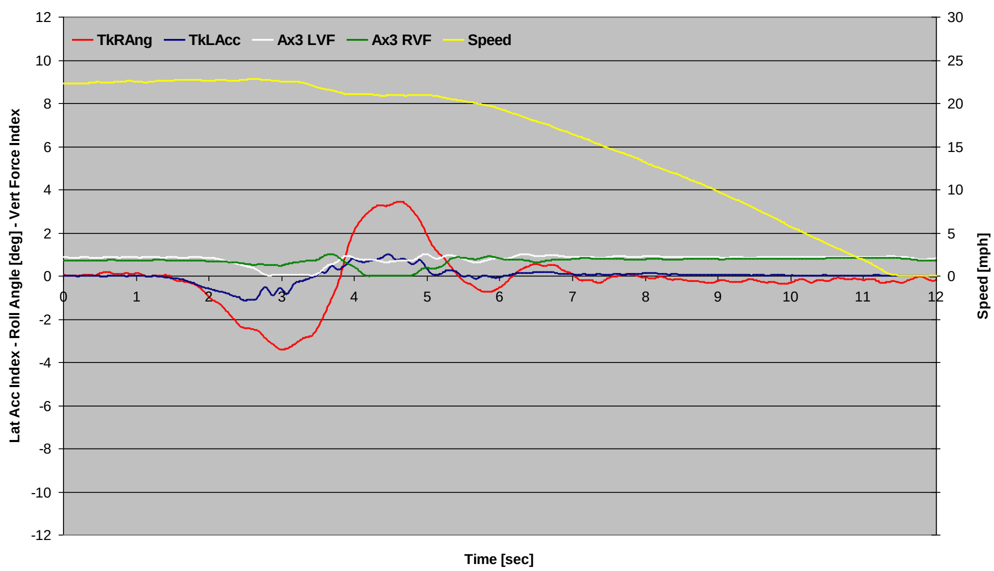


Figure 114. Graph. Tractor Lateral Acceleration, Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 1 – HiCG/HiSp/Tractor ESC On/NGSWBTs Case).

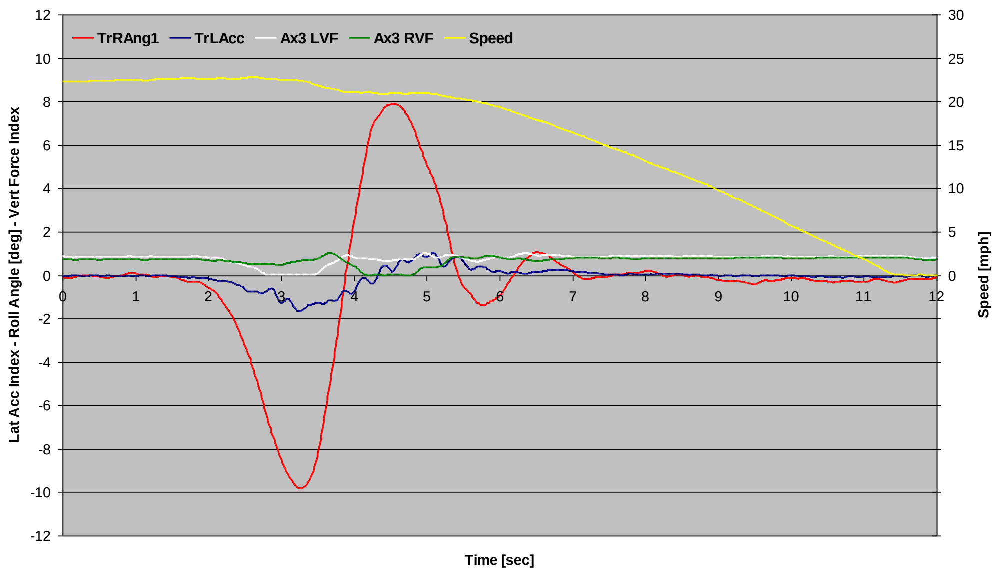


Figure 115. Graph. Flatbed-Trailer Lateral Acceleration, Front Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 1 – HiCG/HiSp/Tractor ESC On/NGSWBTs Case).

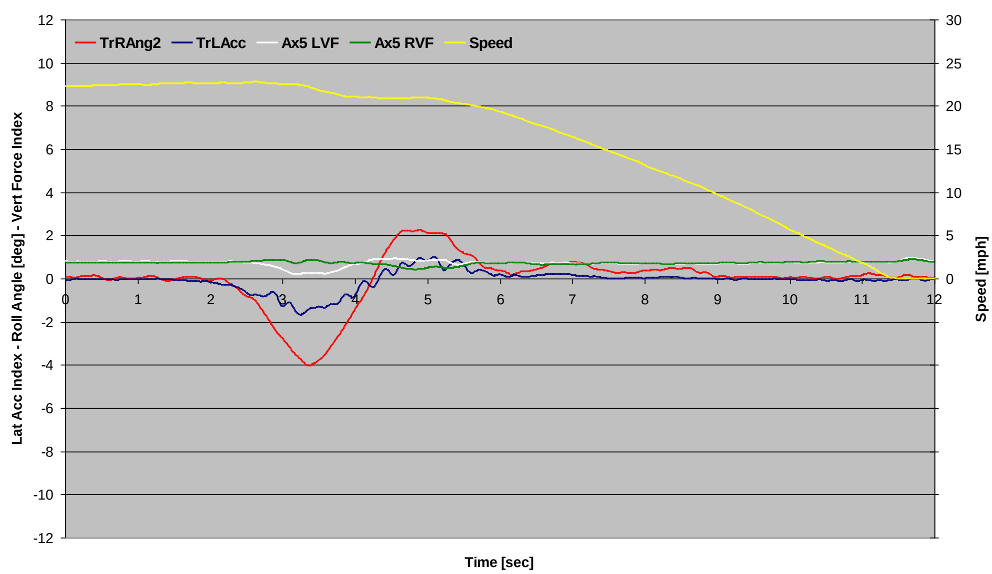


Figure 116. Graph. Flatbed-Trailer Lateral Acceleration, Rear Roll Angle, Axle 5 Vertical Force, and Speed vs. Time (Run 1 – HiCG/HiSp/Tractor ESC On/NGSWBTs Case).

The results of the statistical tests comparing the ***RS*** variable for the dual tires versus NGSWBTs with the tractor ESC on case, are presented in table 19. Those results indicate that, in any of the cases analyzed (i.e., looking at the tractor and flatbed trailer ***RS*** for both the first and second half of the maneuver), the null hypothesis could be rejected with a high confidence level, thus indicating that the type of tires does not have a noticeable effect on the roll stability of the tractor-flatbed-trailer during an evasive maneuver when the tractor ESC is engaged.

Table 19. Comparison of the *RS* Variable for Dual Tires vs. NGSWBTs (HiCG/HiSp/Tractor ESC On Case)

	1st Part of the Lane Change Maneuver						2nd Part of the Lane Change Maneuver					
	TkRS		TrRS1		TrRS2		TkRS		TrRS1		TrRS2	
	D*	W**	D*	W**	D*	W**	D*	W**	D*	W**	D*	WBST
Mean	7.11	9.05	31.21	33.34	12.68	14.11	8.03	8.87	43.53	47.14	11.91	13.90
Variance	0.09	0.99	3.10	1.73	0.92	0.01	0.64	1.45	15.79	2.11	1.23	1.18
Count	3	2	3	2	3	2	3	2	3	2	3	2
v (degrees of freedom)	**1**		**3**		**2**		**2**		**3**		**2**	
Computed t' Value	**2.6890**		**1.5431**		**2.5495**		**0.8671**		**1.4351**		**1.9780**	
Reject at Confidence Level =	**77.33%**		**77.95%**		**87.45%**		**52.27%**		**75.33%**		**81.35%**	

* D = Dual Tires; ** W = NGSWBTs

A similar analysis to the one presented above was performed using the information collected in both the low- and high-CG payload runs. The results of the analysis were the same; that is, the null hypothesis could only be rejected at a low level of confidence (94% in the best case; looking at TrS2 for the first half of the maneuver). Notice that in both cases, the number of runs was very small (i.e., three vs. two, and five vs. two, for the high-CG payload and the lowCG-payload cases, respectively) and therefore the statistical tests do not have the same power as those conducted in the previous subsection; i.e., the lane change maneuver without tractor ESC on. As discussed above, the lane change maneuver test was also conducted with the payload CG height at 8.5 ft above the ground (LoCG condition), with the vehicle equipped with dual tires. Figure 117 to figure 119 show the vehicle dynamics information used to compute the ***RS*** variable for run 3 of the lane change maneuver with LoCG, high-speed, tractor ESC on tests. Notice that, as might be expected, the maximum roll angle and lateral acceleration are lower than in the high- CG payload case (figure 111 to figure 113).

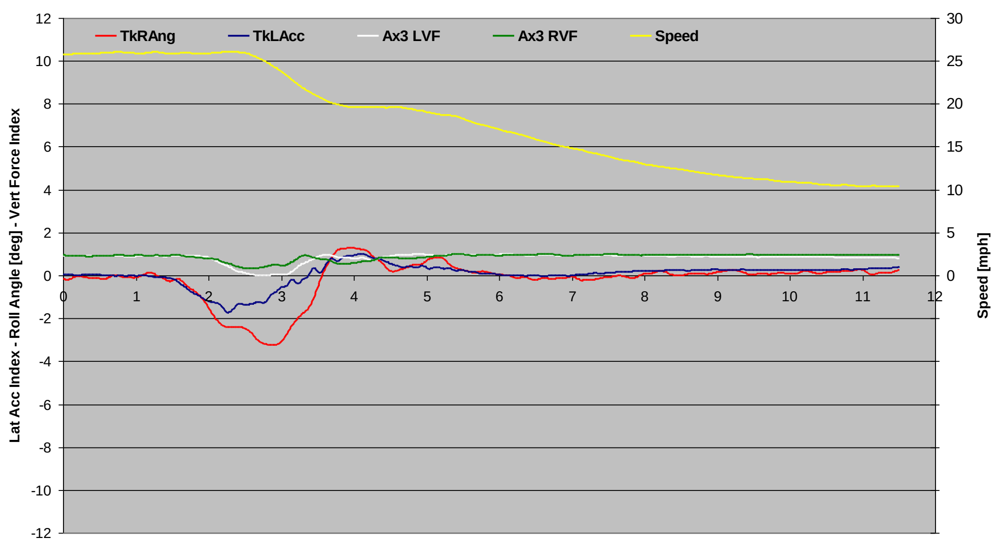


Figure 117. Graph. Tractor Lateral Acceleration, Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 3 – LoCG/HiSp/Tractor ESC On/Dual Tires Case).

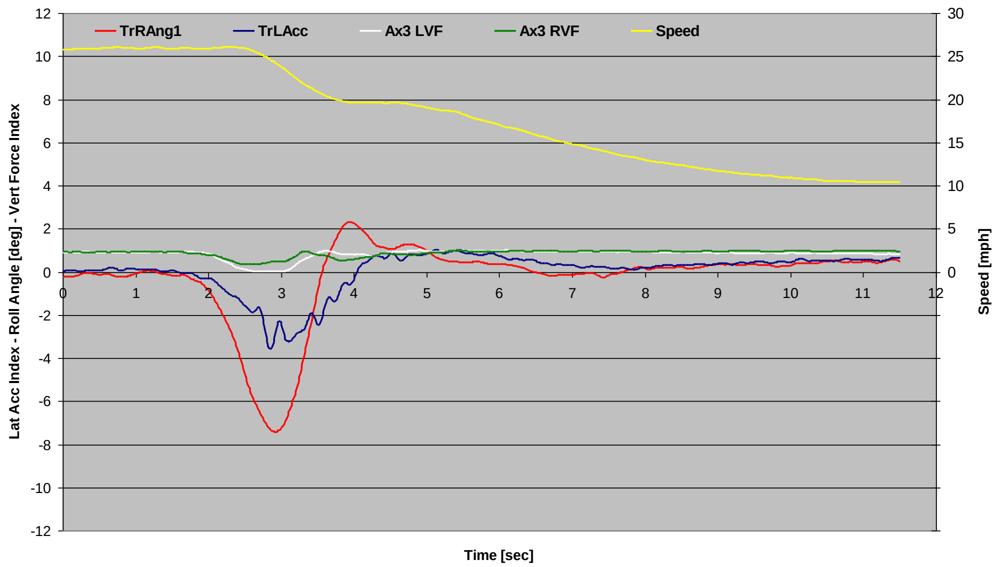


Figure 118. Graph. Flatbed-Trailer Lateral Acceleration, Front Roll Angle, Axle 3 Vertical Force, and Speed vs. Time (Run 3 – LoCG/HiSp/Tractor ESC On/Dual Tires Case).

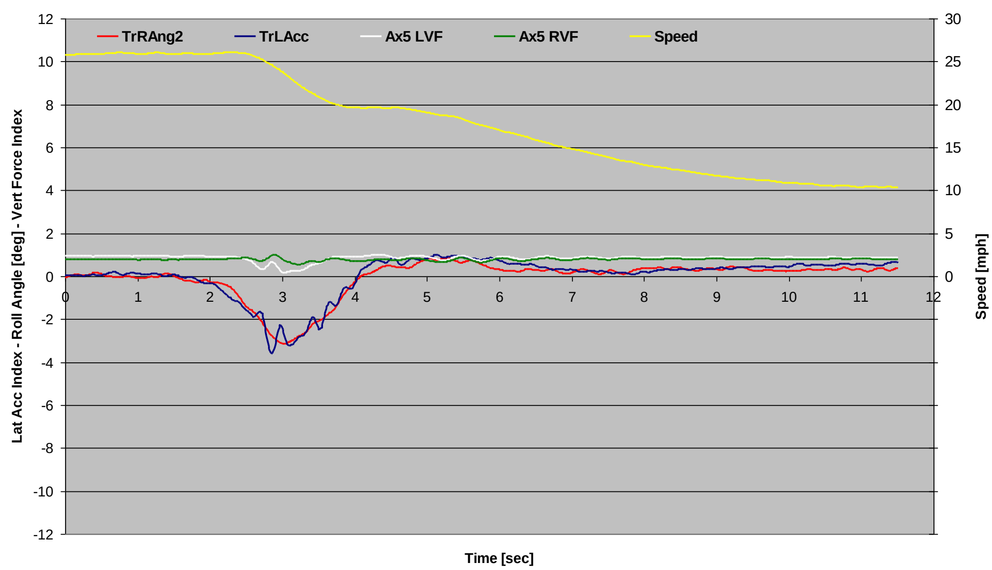


Figure 119. Graph. Flatbed-Trailer Lateral Acceleration, Rear Roll Angle, Axle 5 Vertical Force, and Speed vs. Time (Run 3 – LoCG/HiSp/Tractor ESC On/Dual tires Case).

The statistical tests comparing the ***RS*** variable for the low-CG payload versus the high-CG payload (tractor ESC on, high-speed, dual tires case) showed that the null hypothesis could be rejected only at the 95% confidence level (looking at ***TrRS1*** in the first half of the maneuver) and 97.7% (looking at ***TkRS*** in the second half of the maneuver). These levels of confidence were lower than in the tractor ESC off case; however, the number of runs was also lower (seven vs. seven in the tractor ESC off case against three vs. two in the tractor ESC on case), and therefore the statistical tests were not as powerful.

A noticeable effect of the tractor ESC system was the sharp decrease in the speed of the vehicle in the second half of the maneuver. Compare, for example, the graph of the vehicle speed shown in figure 114 (run 1 of the HiCG/HiSp/Tractor ESC On/ NGSWBTs test) against that of figure 105 (i.e., run 5 of the HiCG/HiSp/Tractor ESC Off/ NGSWBTs tests). In the latter (tractor ESC off) the speed is almost constant (slightly increasing) during the entire lane change maneuver, while in the former, the tractor-flatbed-trailer speed is sharply reduced, particularly during and after the second half of the maneuver when the tractor ESC system has had time to react to the rapid changes in lateral acceleration generated during the first half of the evasive maneuver. In the run shown in figure 114, the tractor-flatbed-trailer went from a speed of about 20 mph after exiting gate 2, to a complete stop in approximately 6 sec. Although the driver was instructed to maintain speed and not braking, brake pedal position was not a piece of information collected in the tests, and therefore it was not possible to determine if this sharp reduction in speed was attributed to the tractor ESC system or to the driver (although a video of this particular run seems to indicate that the driver did not activate the brakes). In future tests, the brake pedal position will be added to the list of channels to be collected to further investigate the effects of the tractor ESC system in the overall behavior of the tractor-flatbed-trailer. Nevertheless, when the maximum lateral accelerations registered with and without the tractor ESC system engaged were compared, it was observed that during the first half of the lane change maneuver the null hypothesis stating that the distributions of maximum lateral accelerations were the same, could not be rejected. However, during the second half of the maneuver, when the tractor ESC system has had time to react, the null hypothesis could be rejected at a 99.9% confidence level, showing that the tractor ESC system reduced the maximum lateral acceleration. The results of these t-tests are shown in table 20 below.

Table 20. Comparison of the Lateral Accelerations Observed Without and With the Tractor ESC Engaged

	1st Part of the Lane Change Maneuver				2nd Part of the Lane Change Maneuver			
	TkLA		TrLA		TkLA		TrLA	
	ESC Off	ESC On	ESC Off	ESC On	ESC Off	ESC On	ESC Off	ESC On
Mean	-0.37	-0.37	-0.26	-0.27	0.37	0.28	0.20	0.12
Variance	0.003	0.002	0.001	0.001	0.002	0.003	0.001	0.001
Count	19	7	19	7	19	7	19	7
v (degrees of freedom)	**24**		**24**		**24**		**24**	
Computed t Value	**0.9832**		**0.2239**		**0.0004**		**0.0000**	
Reject at Confidence Level =	**1.68%**		**77.61%**		**99.96%**		**100.00%**	

Lateral Accelerations at Liftoff, Sequence of Axle Lift and Lift Duration

The experimental data collected from the testing provided great insight into the handling of the tractor-flatbed-trailer and produced some challenges as well. In the analysis of the data, a few challenges were presented that were not unlike any experimental data collection. The challenges included height sensors that would go out of range with roll, and the lack of a means by which to produce an absolute zero for the steering transducer. The axle mounted strain gages proved to be extremely valuable as an indicator of axle lift.

The collected csv files were imported into Excel and grouped by test set. Correction columns were added to allow roll and zero corrections as appropriate. Lift indicator columns were added that allowed lift and lift duration to be identified with an adjustable threshold for each column.

Lift columns were used for all axles with the exception of the steer axle. A roll angle column was added that allowed roll determination, and because of the dropout problem, a maximum adjustable threshold was set to determine when dropout was present, and then the value of that angle was omitted. Roll corrected lateral accelerations for the tractor and flatbed-trailer were established. Tractor and flatbed-trailer lateral accelerations were calculated from the measured yaw rate and the velocity. Understeer was calculated from the measured steer angle and the calculated Ackermann steer required for the turn.

The tractor roll was calculated from the height sensors on the tractor. Two pairs of height sensors on the flatbed-trailer were used. By comparing the tractor roll, with the front flatbed-trailer roll, determination of the 5th wheel separation could be made. Values that were in the range of 3 deg were an indication that 5th wheel separation could be present. This occurred in some tests. Comparing the front flatbed-trailer roll with the rear flatbed-trailer roll indicated the amount of flatbed-trailer twist.

In all of the testing, the rear-most axle of a set was most likely to lift. On the tractor the rear-most drive axle was the axle which most consistently lifted first. That would be followed by either the lift of the rear flatbed-trailer axle or the front drive axle on the tractor. The data tables that follow in the next sections indicate when lift was present. The data field was parsed according to the lift of each axle. For the lift duration row, for each axle lifting, an average duration was calculated. The tables indicate which axle lifted first, and their sequence (axle 1 is the first drive axle, axle 2 is the second drive axle, axle 3 is the front trailer axle and axle 4 is the rear trailer axle. The "delay" indicated in the tables is the time from when the first axle lifts to the lift of any subsequent axles. The tables include a "representative run" and the average for the set. The "delay in the tractor - flatbed trailer lift" indicates the time duration for the flatbed-trailer to respond, again for a "representative" case and on-average. As such, the data in the following tables therefore are not directly predicting the rollover threshold but are determining an average g loading during the duration of the lift event.

<u>High Speed Step Steer – High-CG Payload Height (NGSWBTS)</u>

In the step steer maneuver the average of the values for the duration of lift and the maximum value at which lift occurred were closely aligned. In the lane change maneuver the lateral acceleration for wheel lift and the average lateral acceleration during the wheel lift event differed, and therefore the average values for the event are lower than the threshold values.

Some sample data for the high-speed step steer maneuvers presented in table 21. Figure 120 presents the delay in lateral accelerations between the tractor and the trailer.

Table 21. Sample Data for the High-Speed Step Steer Maneuver with High-CG Payload and NGSWBTs

High-Speed Step Steer	Load Height = High-CG				Tractor ESC = Off	
	Tires = NGSWBTs					
	Run Number 20				Average of Runs 15-30	
	Axle 1	Axle 2	Axle 3	Axle 4	Tractor	Flatbed-Trailer
Axle Lift Acceleration (g)	0.31	0.31	0.32	0.32	0.32	0.31
Lift Sequence	2	1	4	3		
Lift Duration (sec)	.97	2.22	1.59	1.66	4.47	3.92
Delay (sec) Between Axles of a Set for Liftoff	1.08		03		1.0	1.45
Delay in the Tractor-Flatbed-Trailer Lift (sec)	1.34				1.87	
High-Speed Step Steer	Load Height = High-CG				Tractor ESC = On	
	Tires = NGSWBTs					
	Run Number 32				Average of Runs 31-35	
	Axle 1	Axle 2	Axle 3	Axle 4	Tractor	Flatbed-Trailer
Axle Lift Acceleration (g)		0.33		0.29	0.34	
Lift Sequence (sec)		1		2		
Lift Duration (sec)		0.93		.02	0.75	0.13
Delay (sec) Between Axles of a Set for Liftoff	NA		NA		NA	NA
Delay in the Tractor-Flatbed-Trailer Lift (sec)	1.02				0.75	

Comparing the high-speed step steer maneuver with tractor ESC off and tractor ESC on demonstrates the influence of the tractor ESC. The wheel lift accelerations are approximately the same but the duration of the event is considerably shortened when the tractor ESC is engaged.

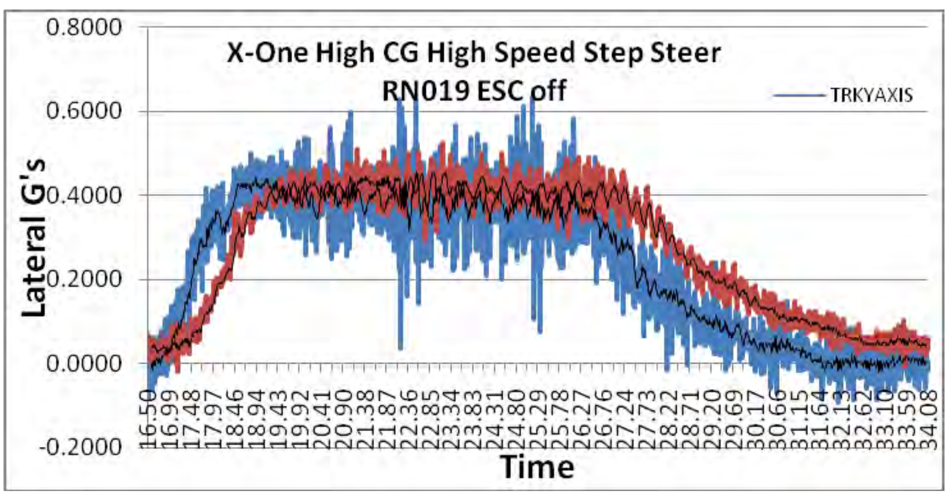


Figure 120. Graph. High Speed Step Steer Lateral Accelerations (NGSWBTs).

High-Speed Step Steer – High-CG Payload Height (Dual Tires)

Some sample data for high-speed step steer maneuver with dual tires on both the tractor drive and the flatbed-trailer axles, is presented in table 22.

Table 22. Sample Data for the High-Speed Step Steer Maneuver with High-CG Payload and Dual Tires

High-Speed Step Steer	Load Height = High-CG				Tractor ESC = Off	
	Tires = Duals					
	Run Number 11				Average of Runs 5-17	
	Axle 1	Axle 2	Axle 3	Axle 4	Tractor	Flatbed-Trailer
Axle Lift Acceleration (g)	0.24	0.31	0.37	0.36	0.31	0.35
Lift Sequence	4	1	3	2		
Lift Duration (sec)	0.61	3.61	0.28	0.20	4.54	3.41
Delay (sec) Between Axles of a Set for Liftoff	2.71		0.25		3.02	0.86
Delay in the Tractor-Flatbed-Trailer Lift	1.21				1.81	
High-Speed Step Steer	Load Height = High CG				Tractor ESC = On	
	Tires = Duals					
	Run Number 18				Average of Runs 18-23	
	Axle 1	Axle 2	Axle 3	Axle 4	Tractor	Trailer
Axle Lift Acceleration (g)		0.32			0.32	
Lift Sequence		1				
Lift Duration (sec)		0.33			0.40	NA
Delay (sec) Between Axles of a Set for Liftoff	NA		NA		NA	NA
Delay in the Tractor-Flatbed-Trailer Lift	NA				NA	

Comparing the high-speed step steer maneuver with ESC off and ESC on demonstrates again the influence of the ESC. The wheel lift accelerations are approximately the same, but the duration of the event is considerably shortened when the ESC is engaged. Virtually no difference appears between the dual tires and the NGSWBTs.

High-Speed Step Steer – Low-CG Payload Height (Dual Tires)

Some sample data for high-speed step steer maneuver with dual tires on both the tractor drive and flatbed-trailer axles, is presented below.

Table 23. Sample Data for the High-Speed Step Steer Maneuver with Low-CG Payload and Dual Tires

High speed step steer	Load Height = High-CG				Tractor ESC = Off	
	Tires = Duals					
	Run Number 18				Average of Runs 13-31	
	Axle 1	Axle 2	Axle 3	Axle 4	Tractor	Flatbed-Trailer
Axle Lift Acceleration (g)		0.36	0.40	0.40	0.36	0.40
Lift Sequence		1	3	2		
Lift Duration (sec)		2.39	1.43	3.19	3.32	3.72
Delay (sec) Between Axles of a Set for Liftoff			1.61		2.19	0.64
Delay in the Tractor-Flatbed-Trailer Lift	0.18				0.82	

High-Speed Step Steer	Load Height = High-CG				Tractor ESC = On	
	Tires = Duals					
	Axle 1	Axle 2	Axle 3	Axle 4	Tractor	Flatbed-Trailer
Axle Lift Acceleration (g)	No Lift Present					
Lift Sequence						
Lift Duration						
Delay (sec) Between Axles of a Set for Liftoff						
Delay in the Tractor-Flatbed-Trailer Lift (sec)						

Comparing the high-speed step steer maneuver with ESC off and ESC on demonstrates again the influence of the ESC. At low-CG payload heights no lift was present in any of the ESC tests with dual tires.

High-Speed Lane Change Maneuver – High-CG Payload Height, (NGSWBTs)

Because the high-speed lane change maneuver was considered to be a relatively violent maneuver that required more driver repeatability, the results produced were less consistent. The variation in each phase of the event resulted in inconsistency in the averaging of results. Some sample data for the high-speed lane change maneuver, with NGSWBTs on both the tractor drive and flatbed-trailer axles, is presented in table 24. It should be noted that the lower lateral acceleration for the 2nd axle lift in Run Number 6 is related to the roll response of the system.

Table 24. Sample Data for the High-Speed Lane Change Maneuver with High-CG Payload and NGSWBTs

High-Speed Lane Change	Load Height = High-CG			Tractor ESC = Off	
	Tires = Duals				
		Run Number 6			
		Axle 1	Axle 2	Axle 3	Axle 4
Axle Lift Acceleration (g)	CCW				
	CW	0.307	0.423		
Lift Sequence	CCW				
	CW	2	1		
Lift Duration (sec)	CCW				
	CW	0.6	0.9		
Delay (sec) Between Axles of a Set for Liftoff	CCW				
	CW	0.24			
Delay in the Tractor-Flatbed-Trailer Lift	CCW	No Trailer Lift			
	CW	No Trailer Lift			

High Speed Lane Change		Ld Ht	High		ESC
		Tires	NGSWBTs		On
		Run Number 12			
		Axle 1	Axle 2	Axle 3	Axle 4
Axle Lift Acceleration (g)	CCW		0.454		
	CW		0.411		
Lift Sequence	CCW		1		
	CW		1		
Lift Duration (sec)	CCW		0.23		
	CW		0.69		
Delay (sec) Between Axles of a Set for Liftoff	CCW	Only Axle 2 Lifted		Only Axle 2 Lifted	
	CW	Only Axle 2 Lifted		Only Axle 2 Lifted	
Delay in the Tractor – Flatbed-Trailer Lift	CCW	No trailer lift			
	CW	No trailer lift			

Comparing the high-speed lane change maneuver with ESC off and ESC on is much more difficult to discern when compared to the step steer tests. This may be partially due to the greater difference between the tractor and flatbed-trailer lateral accelerations. The lift threshold approached 0.45 g for the tractor for this event but the resolution of the data was not as good. The high-speed, high-CG payload lane change maneuver tests produced no trailer lift.

DATA ANALYSIS SUMMARY AND CONCLUSIONS

The field testing of the tractor-flatbed-trailer was a successful enterprise with good data being collected and good insights being gained into the operation of the tractor-flatbed-trailer and its limitations. The constant radius maneuver was completed at a low-speed to quantify the understeer characteristics of the tractor-flatbed-trailer combination. For the low-speed testing, the data indicated that at low lateral acceleration levels, the vehicle needed approximately 3 deg of steering input at the steering wheel above that which would be dictated by the Ackerman geometry in order to maintain the measured radius of curvature. At higher lateral acceleration levels, the steering input approached the Ackerman requirement. This indicates that the vehicle was slightly understeering, but at a very slowly decreasing rate with respect to its lateral acceleration. All of the tested payload and tire combinations had very similar understeer curves. Due to the high-CG payload condition, the steady-state testing was limited to accelerations below 0.2 g. Changing tires or payload heights did not change the understeer response significantly.

For the high-speed constant radius maneuver, the Duncan Multiple Range test was used to analyze the data. It showed that the reduction of the payload CG height by 1 ft was statistically significant, but the selection of tires was not (similar conclusions were derived from the step steer and lane change maneuvers analysis). The ability to discern a statistically significant change in vehicle lift response of a 1 ft payload CG height change indicated a reasonably well-controlled test method. The tests were repeated with the tractor ESC system engaged with the expected result that wheel lift was prevented for all of the constant radius runs at higher speeds.

The high-speed constant radius test data also showed that for the higher CG payload case, the drive axles lifted at a slightly lower lateral acceleration level than the flatbed-trailer axles. This is in contrast to the lower CG payload case wherein the flatbed-trailer axles lifted with the drive axles. Because of the trailer's torsional flexibility, which severely limits its ability to effectively transmit torque from one end to the other, it was deduced that the front and rear portions of the flatbed-trailer have very similar roll limits.

Drive axle lift sensitivity was determined to be 0.06 g/ft of CG height change in the payload, and 0.12 g/ft in system CG height change (includes the masses of the axle, outriggers, weight rack, etc.). The flatbed-trailer axle lift sensitivity was 0.04 g/ft of CG height change in the payload, and 0.067 g/ft in system CG height change. Note that the tractor has significantly more mass than the flatbed-trailer so that the payload over the flatbed-trailer is larger than the payload over the drive axles.

The step steer test maneuver was designed to evaluate the transient behavior of the tractor-flatbed-trailer using a repeatable steering input. Testing was initially performed without the tractor ESC engaged to evaluate the tractor-flatbed-trailer's stability, and then with the tractor ESC engaged to assess the benefits of such a system.

For the case in which the tractor ESC was not engaged, several key vehicle metrics were selected as potential lift indicators including vehicle velocity, lateral acceleration, roll rate, yaw rate, roll angle, and vehicle side slip angle. The results of the statistical analysis showed that the payload CG height and lateral acceleration were well correlated to the lift event, while the remaining parameters were not. The analysis also showed that it was possible to determine with 95% confidence, that the low-CG payload tractor-flatbed-trailer response was statistically different from the response of the high-CG payload tractor-flatbed-trailer for both drive and flatbed-trailer axle sets. It was not possible to distinguish between the behaviors of the dual tires and NGSWBTs for either the drive or flatbed-trailer axle sets. The indication is that dual tires and NGSWBTs have similar performance in roll stability for a torsionally compliant tractor-flatbed-trailer.

For the cases where the tractor ESC system was engaged with a high-CG payload configuration, the analysis of the data showed that for the tractor, the axle lift thresholds for speed or lateral acceleration were similar to those observed with the tractor ESC off condition, but the duration of the observed wheel lift decreased. This would be expected since the engagement of the tractor ESC system should tend to bring the tractor-flatbed-trailer back into a stable operating situation more quickly. The flatbed-trailer, being delayed in time from the tractor's steering input, did not lift with the ESC system on as the vehicle was able to slow before the flatbed-trailer's lateral accelerations could begin to build. The data, however, could not be used to statistically quantify the lift duration reduction because the driver's response after a wheel lift occurred varied significantly. For the low-CG payload case, the tractor ESC system was effective in preventing wheel lift under the same conditions that wheel lift was observed for this maneuver when the tractor ESC was not engaged.

For the lane change maneuver, roll stiffness (***RS***), computed as the ratio of the maximum roll angle divided by the maximum lateral acceleration, was used as the analysis variable (the variable ***RS*** was computed for the tractor, and the front and rear of the flatbed-trailer). For each lane change maneuver (a total of 70 runs), two roll stiffness values were computed: one for the first half of the maneuver (left turn) and one for the second half (right turn). Since in the majority of the cases (more than 85%) the distributions of ***RS*** had very dissimilar variances for the different treatments analyzed, a Smith-Satterthwaite Test was used to test the null hypothesis that the means of each pair of compared distributions of ***RS*** (e.g., the ***RS*** values for the dual tires vs. the ***RS*** values of the NGSWBTs, for high-CG payload and high-speed) were the same. Concentrating on the first part of the maneuver, the results showed that the null hypothesis could only be rejected with a low level of confidence (less than 95%) when the dual tires were compared against the NGSWBTs for the cases of high-CG/high-speed/tractor ESC-off, high-CG/low-speed/tractor ESC-off, and high-CG/high-speed/tractor ESC-on. This indicates no statistically significant differences between the two types of tires regarding roll stability. The tests also seem to indicate that the engagement of the tractor ESC system made the ***RS*** distributions of dual tires and NGSWBTs "more equal" since the null hypothesis could only be rejected at lower confidence levels than when the system was not engaged. This difference was more pronounced in the second half of the maneuver, when the tractor ESC system had some time to act (i.e., the first half of a lane-change maneuver consisted of an abrupt change in the direction of travel that cannot be predicted by the tractor ESC system). Notice, however, that because only five runs were conducted with the tractor ESC on (three with dual tires and two with NGSWBTs), those tests are not as powerful as those conducted with the tractor ESC off (12 runs, seven with duals and five with NGSWBTs).

Regarding the effect of CG height; the results show that for the high-speed case, the null hypothesis could be rejected at a very high level of confidence (more than 99.99%, except for the tractor ***RS*** comparison in the second half of the maneuver where the null hypothesis could be rejected with 96.9% confidence). This strongly indicates that the 1 ft decrease in the height of the CG increases the roll stability of the tractor-flatbed-trailer substantially (average decreases in ***RS*** of 16% was observed for the tractor and 13%-17%-to-20%-30% for the flatbed-trailer was observed, for the first half and second half of the maneuver, respectively). For the low-speed runs the null hypothesis could only be rejected at the 99% confidence level when considering the ***RS*** distributions for the rear of the flatbed-trailer, indicating that the 1 ft CG height decrease showed statistically significant gains in roll stability for the flatbed-trailer. For the tractor and the front of the flatbed-trailer, the null hypotheses could be rejected with 98% and 90% confidence level, respectively.

In summary, the results of the field tests show that, in terms of roll stability, the reduction of the payload height by 1 ft was statistically significant for all of the maneuvers (and that the quality of the testing was very good), but the selection of tires did not have an impact on roll stability. Although there were very few runs conducted with the tractor ESC system on, the results of the tests seem to indicate that there is a positive contribution of this system in terms of roll stability. However, the testing also indicated that the tractor ESC system would not prevent wheel lift under some situations (i.e., for some speed, steering, and payload combinations) but appears to reduce the duration of the wheel lift. The results suggest that the effect of the tractor ESC needs to be further and specifically investigated for the lane change maneuver. Additionally, to address the issue of wheel lift duration, a different vehicle control strategy and set of measurements will be required.

CHAPTER 11. MODELING

There are two basic types of vehicle dynamics models: functional models and component models - also referred to as multi-body models. Component models use component level data (A-arms, bushings, springs, etc.) and allow for the building of a virtual vehicle based on the component properties, locations, and motion ranges. Functional models reduce the entire vehicle to wheel plane motion vs. suspension deflection, steer, and force and moment input responses that ignore the details of the motion mechanisms.

Component models permit the evaluation of each part of the vehicle and the virtual modification of the vehicle. However, they require significant fidelity in the knowledge of every part in the model and a good understanding of the connections of the members to accurately evaluate motions.

Functional models require less sophisticated data because the global response of the system to a series of applied loads is all that is needed to define the model. Functional models are also generally less expensive to run because the number of degrees of freedom in the model is significantly less than a component-level model. However, functional models do not lend themselves to the evaluation of internal components within the model, and they are more difficult to modify for vehicle component changes.

Several models were developed for this analysis, including two TruckSim® models (functional models) for the tractor and the flatbed-trailer, a solid model used in finite element analysis, and a kinematic model for the trailer. The first subsection below discusses several vehicle parameters that were used as input for the models developed in this project. The following four subsections focus on the models, including calibration and comparison of model and field test results. The last subsection presents a summary and conclusions of the modeling effort.

The data used in the development of these TruckSim® models was a combination of the kinematic and compliance data obtained using the Michelin K&C rig and vehicle kinematics and compliance data developed by Western Michigan University. The Western Michigan University data is a combination of measured test data collected by WMU and data collected from other published sources. The source of the data presented below is generally from Western Michigan University except as noted as Michelin test data.

VEHICLE PARAMETERS

Suspension Bump and Roll Stiffness Procedural Overview

Kinematic analysis of the suspension bump and roll stiffness was evaluated in a quasi-static test procedure. The procedure includes blocking the air bag inlet and exhaust mechanisms of each suspension so that suspension travel sensors do not admit nor expel air during the testing operation. Table 25 shows the representative bump stiffness of each suspension and a calculated ride rate which includes the suspension and tire interaction. The values used below are representative values that are available in published literature and simulation programs.

Table 25. Suspension Bump and Ride Stiffness

Suspension Bump and Ride Stiffness Relationships					
Individual Tire Radial Stiffness All Stiffnesses are Nominal Values for Typical Sized Tires					
Steer Axle Tire Radial Stiffness		5300	lb/in	928	N/mm
Drive Axle Tire Radial Stiffness	**Dual Tires**	4500	lb/in	788	N/mm
	NGSWBTS	6500	lb/in	1138	N/mm
Flatbed-Trailer Axle Tire Radial Stiffness	**Dual Tires**	4500	lb/in	788	N/mm
	NGSWBTS	6500	lb/in	1138	N/mm
Total Axle Rates (per axle)					
The Suspension Rates Given Below are Typical; Ride Rates are Predicted Based on the Tire Radial Rates Specified Above					
Steer Axle Suspension Rate		2850	lb/in	500	N/mm
Ride Rate Calculated		2250	lb/in	390	N/mm
Drive Axle Suspension Rate	**Dual Tires**	2500	lb/in	440	N/mm
Ride Rate Calculated	**Dual Tires**	2195	lb/in	380	N/mm
Drive Axle Suspension Rate	**NGSWBTS**	2500	lb/in	440	N/mm
Ride Rate Calculated	**NGSWBTS**	2100	lb/in	370	N/mm
Flatbed-Trailer Axle Suspension Rate	**Dual Tires**	3200	lb/in	560	N/mm
Ride Rate Calculated	**Dual Tires**	2720	lb/in	475	N/mm
Flatbed-Trailer Axle Suspension Rate	**NGSWBTS**	3200	lb/in	560	N/mm
Ride Rate Calculated	**NGSWBTS**	2570	lb/in	450	N/mm

The quasi-static nature of the suspension characterization process effectively eliminates the contributions of the air bags to the suspension roll stiffness. During modeling, this motivated the analysts to use the measured roll stiffness as the auxiliary stiffness and to ignore the air bag as a contributor to suspension roll stiffness for the drive and flatbed-trailer axles as shown in table 26. This is a reasonable assumption for the slow-speed maneuvers but may add some error in high-speed transient analysis because the air transfer rate influences the individual bag stiffness.

Table 26. Suspension Roll Stiffness for Modeling

Suspension Roll Stiffness		N-m/deg	Ft-lb/deg
Steer Axle	From Published Data		
	Contribution from Vertical Spring Rate	1867	1377
	Auxiliary Roll Stiffness (estimated)	361	267
	Total Steer Axle	2230	1643
Individual Drive Axles	From Published/Estimated Data		
	Contribution from Vertical Spring Rate	6454	4759
	Auxiliary Roll Stiffness	3628	2676
	Total per Axle (modeling-aux)	10082	7435
Individual Flatbed-Trailer Axles	From Published/Estimated data		
	Contribution from Vertical Spring Rate	8126	5993
	Auxiliary Roll Stiffness	3600	2655
	Total per Axle (modeling-aux)	11726	8647

Tractor and Flatbed Trailer Frame Torsional Stiffness

The tractor frame couples the tractor drive axle roll stiffness to the tractor steer axle roll stiffness, and hence, it is the roll moment transfer mechanism between the tractor's front and rear suspensions. To enable the analysis of the influence of the tractor frame $K_{\phi\ frame}$ the steer axle roll stiffness $K_{\phi\ SteerAxle}$ and the frame torsional stiffness were combined to reflect the torsional stiffness of the steer axle and frame combined to the drive axles (see equation 22 and equation 23).

$$\frac{1}{K_{\phi\ comb}} = \frac{1}{K_{\phi\ frame}} + \frac{1}{K_{\phi\ SteerAxle}} \qquad \mathbf{22}$$

Figure 121. Equation 22.

$$K_{\phi\ comb} = \frac{K_{\phi\ frame} \times K_{\phi\ steer\ axle}}{K_{\phi\ frame} + K_{\phi\ steer\ axle}} \qquad \mathbf{23}$$

Figure 122. Equation 23.

The tractor frame torsional contribution is also dependent upon the roll centers of the respective front and rear suspensions and the mass location relative to the tractor roll axis and the trailer Roll axis. Table 27 presents WMU developed representative values for roll center heights at each suspension and the angle of each respective roll axis.. The payload-to-roll axis height is the roll moment arm on which the lateral forces act to induce roll.

Table 27. WMU Developed Roll Axis Characteristics

Suspension	Roll Center Height		Slope	
	in	mm	Location	Angle (deg)
Steer Axle	19	480		
Drive Axle	30	760	Steer Axle-to-1st Drive Axle	2.7
1st Flatbed-Trailer Axle	19	480	Dr axle-to-1st Flatbed-Trailer Axle	-1.44
5th Wheel Height	46	1170	5th wheel-to-1st Flatbed-Trailer Axle	-3.54
2nd Flatbed-Trailer Axle	19	480	5th wheel-to-2nd Flatbed-Trailer Axle	-3.18

Load Height and Roll Center Locations

The roll angle of the tractor and flatbed-trailer relative to the axles is a function of the distance between the load heights and the roll center heights. Testing was performed at three fundamental heights which included the payload distributed along the spine of the flatbed-trailer, and the payload positioned at two different heights in specially designed load racks.

The deck height of the flatbed-trailer was nominally at 58.75 in (1490 mm). With the payload distributed along the spine of the flatbed-trailer, the nominal load height was approximately 59 +16 = 75 in (1905 mm). The condition defined as the high-CG payload height had the load positioned at approximately 116 in (2950 mm) and 111 in (2820 mm) as measured from the ground, for the front and rear, respectively. The condition defined as the low-CG payload height had the load lowered 12 in from the high-CG payload position to establish a height of approximately 104 inches (2640 mm) and 99 inches (2520 mm), for the front and rear, respectively. These loads were positioned longitudinally in very close proximity to the 5th wheel location (directly above the drive axles), and at very close proximity to the flatbed-trailer axles.

Table 28, presents the relationship between the load height and the suspension roll center heights. The load-to-roll center distance depicts the roll moment arm length that is present for the load related cornering forces to act upon. The resulting torque can twist the chassis, and will roll against the suspension in a lateral acceleration-producing turn. Table 28 presents only an analysis of the load induced roll moment characteristics which can induce roll angle into the system. In addition, the relationship of these load induced roll moment characteristics to the load position are presented. It was observed during testing that with the discrete lower CG payload configuration (2642/2515 mm load height), there tended to be a greater tendency for lift of the flatbed-trailer axles when compared with the higher CG load configuration (2946/2819 mm). In the high-CG payload configuration, the drive axles more consistently lifted first. An examination of the drive axle percent of load transfer change below may help explain the observed phenomenon.

Table 28. Load Height-to-Roll Center Relationship and Rolling Percent of Load (WMU Developed Roll Data)

<table>
<tr><th>Load
Position</th><th>Load Height
(mm)</th><th colspan="2">Roll Center Heights (mm)</th><th colspan="2">Load to Roll Center Distances (mm)</th><th colspan="2">Rolling Percent of Load Transfer</th></tr>
<tr><th></th><th></th><th>Drive Axles</th><th>Flatbed-Trailer Axles</th><th>Drive Axles</th><th>Trailer Axles</th><th>Drive Axles</th><th>Flatbed-Trailer Axles</th></tr>
<tr><td rowspan="2">Bed</td><td rowspan="2">1905</td><td>760</td><td>480</td><td>1145</td><td>1425</td><td>60%</td><td>75%</td></tr>
<tr><td colspan="2">620</td><td colspan="2">1285</td><td colspan="2">67%</td></tr>
<tr><td rowspan="2">Front =
Low-CG Payload
R =</td><td rowspan="2">2642

2515</td><td>760</td><td>480</td><td>1882</td><td>2035</td><td>71%</td><td>81%</td></tr>
<tr><td colspan="2">620</td><td colspan="2">1958</td><td colspan="2">76%</td></tr>
<tr><td rowspan="2">F =
High-CG Payload
R =</td><td rowspan="2">2946

2819</td><td>760</td><td>480</td><td>2186</td><td>2339</td><td>74%</td><td>83%</td></tr>
<tr><td colspan="2">620</td><td colspan="2">2263</td><td colspan="2">79%</td></tr>
</table>

Tire and Track Width Relationships

In an effort to examine how tire selection and track width might influence roll characteristics, a comparison with changing tire parameters was performed. The data in table 29 presents the geometric relationships for the truck with; as tested, 96 inch and with 102 inch widths. The 102 inch represents the maximum potential outside-outside width meeting current DOT highway standards. For the dual tires the average (nominal) track width is calculated as well as the effective track width. The effective track width is based on calculation which takes into account the greater track for the outer set of duals than the inner set of duals and the influence that the resulting tire spacing has on roll stiffness. The effective track width for duals is approximately 1.4% greater than the nominal track width determined by the centerlines of the two set of duals on an axle.

Effectively the track width can be increased when a single tire, limited by the maximum outside to outside width, replaces the duals. By using NGSWBTs the effective track width is increased over that of the dual tires. Table 29 shows the percentage change from the base of the different configurations for both tractor and flatbed-trailer. The base is considered the comparable outside to outside dimension of the vehicle set up with duals, therefore the baseline duals are compared to baseline NGSWBT, and the 102 inch duals compared to the 102 inch NGSWBT.

The percent increases over the base demonstrate the changes which can result by optimizing/maximizing track width. The 96 in and 102 in maximum outer dimension are included for comparison purposes, as well as the baseline. The "wider slider" is a special carriage assembly (slider) which is built with longer axles to take advantage of the single tire while allowing more conventional wheel offsets for the NGSWBT. Since track width is related to roll threshold any gains in track width are potential gains in roll threshold. The potential increase would be in the range of 3-6% as shown in table 29 below.

Table 29. Tire and Roll Stiffness Comparisons

Comparison of Effective Track and Potential Gains in Roll Threshold Based on Effective Track Only									
Tractor		Outer Track	Inner Track	Nominal Track	Outer Sidewall	Inner Sidewall	Δ from Max	Effective Track	Potential Gain
Tire Type	Width	in	in	in	in	in	in	in	
Dual Tires	As-tested	92.2	65.6	78.9	101.1	56.7		80.0	ref
Dual Tires	96	87.1	60.5	73.8	96	51.7	5.1	75.0	ref
Dual Tires	102	93.1	66.5	79.8	102	57.7	-0.9	80.9	ref
NGSWBTs	As-tested			82.4	98.1	66.6		82.4	3.0%
NGSWBTs	96			80.3	96.0	64.5	2.1	80.3	7.0%
NGSWBTs	102			86.3	102.0	70.5	-3.9	86.3	6.6%
Trailer									
Dual Tires	As-tested	91.8	65.2	78.5	100.7	56.4		79.6	ref
Dual Tires	96	87.1	60.5	73.8	96	51.7	4.7	75.0	ref
Dual Tires	102	93.1	66.5	79.8	102	57.7	-1.3	80.9	ref
NGSWBTs	As-tested			82.0	97.7	66.3		82.0	3.0%
NGSWBTs	96			80.3	96.0	64.5	1.7	80.3	7.0%
NGSWBTs	102			86.3	102.0	70.5	-4.3	86.3	6.6%
Wider-Slider	(As-tested in Prior Phase)			84.7	100.4	68.9	-4.3	84.7	4.6%

Torsional Stiffness and Suspension Roll Stiffness

The flatbed-trailer frame torsional stiffness distributes the roll moments generated by the payload and lateral accelerations to the tractor and flatbed-trailer suspensions. The magnitude of the payload roll moment is a function of the sprung mass, its distance from the suspension roll axis and the lateral acceleration it experiences. The distribution of the roll moment between the axles is a function of the payload position along the longitudinal axis, and the combination of frame torsional stiffness and suspension roll stiffness opposing the roll moment, both forward and aft of the payload. If the payload is directly above either or both suspensions (e.g., as in the lumped masses configuration used in the final portion of the on-track testing), the payload roll moment above the respective suspension is resisted by the immediate suspension with the portion shared with the alternate suspension as a function of the connecting frame torsional stiffness and the roll stiffness of the alternate suspension. Therefore, a flatbed-trailer payload is resisted by the flatbed-trailer axle roll stiffness coupled through the flatbed-trailer frame to the payload, and the tractor roll stiffness coupled through the flatbed-trailer frame to the fifth wheel.

The tractor's fifth wheel distributes the fraction of the roll couple it receives from the payload to the tractor drive axles and steer axle. The sharing of this fraction of the roll couple is a function of the drive axle roll stiffness, the tractor frame torsional stiffness and the steer axle roll stiffness. The roll moment resisted by the steer axle is obviously a function of the steer axle roll stiffness and the tractor frame torsional stiffness coupling the drive axles to the steer axle.

The flatbed-trailer typically has a lower torsional stiffness compared to other trailer designs and the tested flatbed-trailer was no exception. Long wheelbase tractors, such as that which was used in this research, have low torsional stiffness as well. The torsional stiffness of the flatbed-trailer was evaluated in both field tests and at MARC. The field tests were performed in three configurations as follows: 1) with the flatbed-trailer unloaded, 2) with the flatbed-trailer loaded with the payload distributed along the flatbed-trailer spine on the trailer bed (flat load configuration), and 3) after testing was completed at the TRC with the elevated weight racks and weights positioned at what was termed the low-CG payload position (8'7"front and 8'3"rear). The results of these tests are presented in 0. For modeling, the test results from MARC's K&C testing were used for the flatbed-trailer and for the tractor. Representative values extracted from the field testing are shown in table 30.

Table 30. Frame Torsional Stiffness Extracted from Field Testing

Torsional Stiffness		N-m/deg	Ft-lb/deg
Tractor Frame	Measured (similar tractor- field)	1150	850
Flatbed-Trailer Frame	Measured (field measurement)	2034	1500

The torsional stiffness of the flatbed-trailer influences the roll couple distribution by distributing the roll moment to the flatbed-trailer and/or tractor suspensions. Table 31 presents the ratio of the respective suspension roll properties to the frame torsional properties.

Table 31. Frame/Suspension Torsional Stiffness Percentages

Frame Torsional Stiffness		N-m/deg			
Tractor Frame	Estimated from Similarly Configured Tractor Test	1150			
Flatbed-Trailer Frame	Measured (Field measurement)	2034			
Suspension Roll Stiffness		**N-m/deg**	**Frame Torsion-to-Suspension Roll Stiffness Ratios**		
Steer Axle (measured)	from Table 26Table 26	2228	Tractor Torsion to Steer Axle Roll Stiffness	1150/2228	52%
Drive Axles (measured)	from Table 26Table 26	20164	Tractor Torsion to Drive Axle Roll Stiffness	1150/20164	6%
Total Tractor Suspension Roll Stiffness	from Table 26	22392	Tractor Torsion to Total Tractor-Flatbed-Trailer	1150/22392	5%
Total Tractor Suspension Roll Stiffness	from Table 26Table 26	22392	Flatbed-Trailer Torsion to Total Tractor-Flatbed-Trailer	2034/22392	9%
Flatbed-Trailer Axles	From Table 26	23452	Flatbed-Trailer Torsion to Flatbed-Trailer Roll Stiffness	2034/23452	8.6%

Steady-State Analysis

As part of the project analysis and parameter investigation, a “Special purpose” software analysis package was developed at WMU. Each of the vehicle masses and the payload masses are treated separately with each having its own CG. Special features included in this software package allowed the load to be positioned and distributed with up to six segments each segment having its own weight and height.

The suspension and tire properties part of the package allowed separate evaluation of the contributions of each and their impact on weight transfer. A special feature of the suspension portion of the analysis package allows the user to adjust suspension bushing compliance, trailer axle wall thickness, axle link positions as well as design and add additional anti-roll devices to assess their impact on the overall tractor and trailer suspension characteristics. After the entry of any of the design parameters, the package allows the user to select or deselect the features designed-in, and presents the overall suspension properties which are then utilized in the calculations to determine weight transfer and wheel lift.

The package also allows for the consideration of tractor and trailer torsional stiffness. These properties can be input and their properties assessed with roll and wheel lift, to evaluate the impact on roll as the trailer torsional stiffness or load position is varied. This provides the input for the analysis of load-related frequencies in a relative investigative manner.

A separate potion of the package allows engine and transmission characteristics to be input, and determines factors such as aerodynamic and rolling resistance loads. From the total road loads predicted, the tractor chassis stiffness properties allows prediction of the transverse weight transfer across the steer axle which is then integrated into the weight transfer characteristics. This portion of the package allows some of the asymmetries associated with drive line torque to be investigated in a cursory fashion. Using this analysis package allowed parameters to be varied rather simply and a relative prediction obtained, which then provided a range of acceptable parameters which could subsequently be used in the other modeling programs including TruckSim and SimMechanics.

This package was specially designed to allow for a fast cursory review of various elements of design and analysis. A quick comparison of the experimental data and the program prediction is shown in table 32.

Table 32. Steady State Prediction vs. Experimental Results

Experimental Maneuver	Measured Roll Threshold	Predicted Roll Threshold	1st Wheel Lift Experienced	1st Wheel Lift Predicted
High-CG Payload, Step Steer	0.32 – 0.34 g	0.32 – 0.34 g	Drive Axle	Drive Axle
Low-CG Payload, Step Steer	0.37 – 0.39 g	0.36 – 0.40 g	Drive Axle/Flatbed-Trailer Axle	Drive Axle
Note: The package typically presented flatbed-trailer axle lift at approximately 0.04-0.05 g higher lateral accelerations than required for drive axle lift.				

In some maneuvers, namely lane change, the threshold experimental values for rollover threshold were somewhat lower, but were also less consistent. The steady-state program results did not correlate as well for this maneuver. The inconsistent nature of the maneuver, due to driver variations, suggests that the experimental data should be more closely examined.

Structural Flex Related Load Displacement

A further concern of the flatbed-trailer torsional stiffness relates to how the load is distributed longitudinally, and the flexural displacement of the flatbed-trailer, resulting in a compounding of the load lateral displacement. As the payload moves toward the center of the flatbed-trailer and/or as the payload CG increases in height, the lateral displacement of the load away from the central axis of the flatbed-trailer tends to induce additional roll (see figure 123).

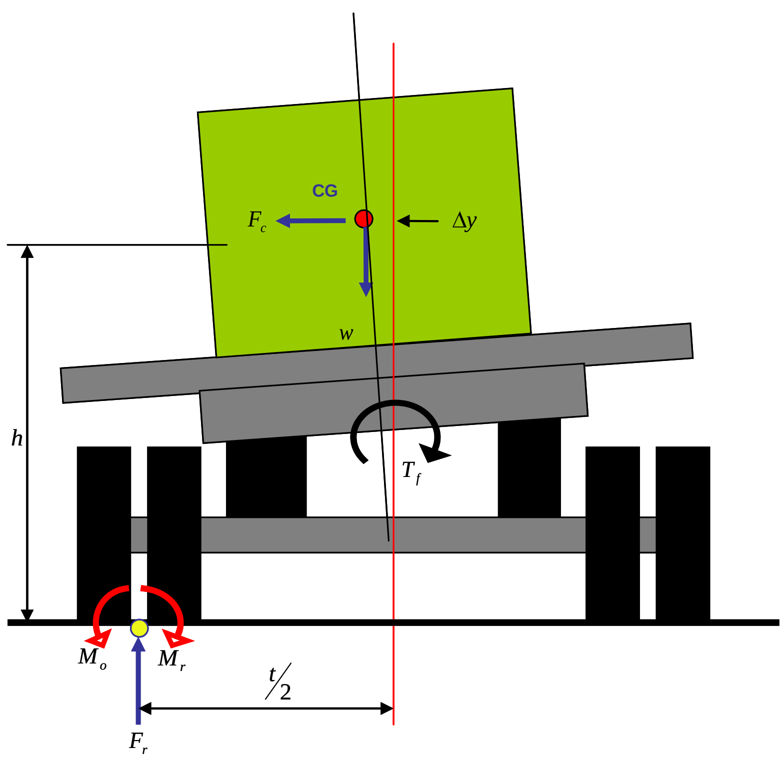


Figure 123. Illustration. Load CG Induced Roll.

The torsional stiffness of the flatbed-trailer, based on the discrete loading configuration used, influenced some of the behavior as a result of the relatively low coupling between the tractor drive axle suspension and the flatbed-trailer suspension. As a result of the load locations directly above the respective suspensions, the roll frequency was related to load height, the respective suspension roll stiffness and the geometric roll axis

Associated with this displacement is a load frequency. On a "rigid" frame, the roll frequency is directly a function of the mass, its distance from the roll axis and the roll resisting torque from the suspension roll stiffness. On a torsionally compliant frame, the roll frequency is a function of the mass, its distance from the roll axis, the roll resisting torque from the suspension roll stiffness, as well as the trailer torsional stiffness, and the inertial properties of the payload and its behavior on the structure's axis.

It was found through modeling that the roll frequency can also be influenced by the load position when a torsionally compliant vehicle structure is utilized. By placing the loads in the model further from the tractor and flatbed-trailer suspensions (toward the center of the flatbed-trailer) the roll frequency of the system could be lowered. Although additional modeling is required in this area for confirmation, some of the cases studied (centrally located loads) can reduce the roll frequency to where the roll frequencies of the loaded flatbed-trailer begin to approach the maneuver frequency. The TruckSim® modeling effort does not appear to be influenced by this characteristic because the stiffness is uniquely defined for computational purposes within the program.

The resulting frequency can be relatively low if the load placement is near the center of the flatbed-trailer between the flatbed-trailer suspension and the tractor rear suspension. In design it is always good practice to avoid having the natural frequency of the structure and its load coincide with the roll frequency of the suspension. With a torsionally compliant structure supporting the load, it appears to be possible to have the structure's frequency with the accompanying load, close to the suspension roll frequency, depending upon vertical and longitudinal load placement.

TRUCKSIM® MODEL

A functional model of the vehicle was developed using the K&C data described in 0. The model was developed in a commercial application called TruckSim® published by the Mechanical Simulation Corporation.

The TruckSim® model sets up the vehicle as a tractor, a trailer, and a load condition. As such, the parameters defining the tractor and the trailer are independent of each other so the tractor can be virtually connected to any trailer.

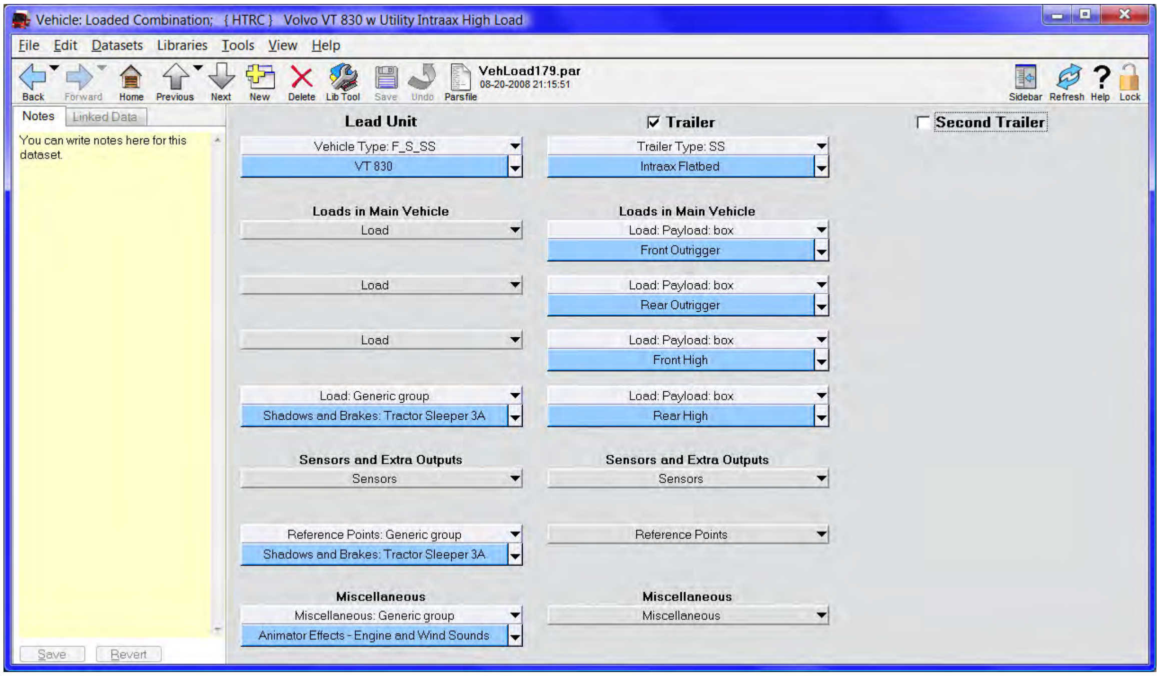


Figure 124. Screen Shot. TruckSim® Vehicle [13].

Each unit of the vehicle (see figure 125 and figure 126) is then virtually built from measured data grouped by general categories. For example, the sprung mass parameters such as CG location, vehicle width and length, etc. are in the sprung mass section. Likewise, each axle has a Kinematics category containing axle settings such as track width, unsprung mass, toe and camber settings, etc. There is also a compliance category for each axle containing its load deflection information, roll stiffness, and damper information.

Figure 125. Screen Shot. TruckSim® Tractor Main Screen [13].

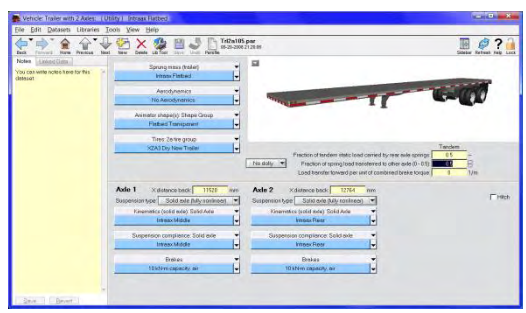


Figure 126. Screen Shot. TruckSim® Flatbed-Trailer Main Screen [13].

The data needed to develop these models was derived from MARC's K&C testing or from the manufactures' design/build data sheets.

Constants, Pull, and Roll

While all of the data needed to develop the model is important, some items are more critical than others. For evaluation of stability limitations, the axle motions are of particular importance. Thus, the key information pulled from the vehicle K&C testing was the axle motions relative to the body motions (bump steer, roll steer, axle displacement, etc.), and the suspension stiffness in jounce, bump stop locations, and roll stiffness.

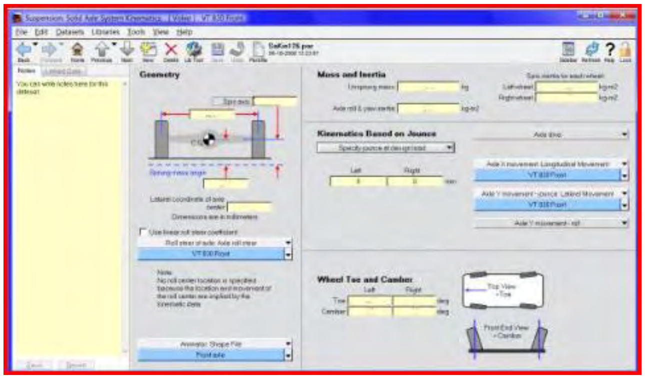


Figure 127. Screen Shot. Kinematics Table [13].

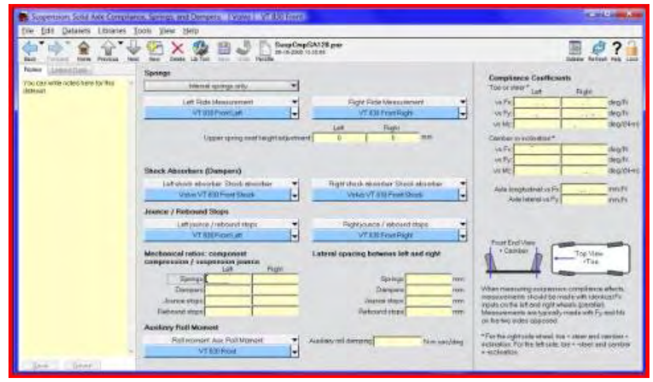


Figure 128. Screen Shot. Compliance Table [13].

Torsional Stiffness

By default, most models assume a perfectly rigid chassis. Since the vehicle used in this study was flexible in torsion, this was an important parameter to include in the model. The representation of the torsional stiffness in TruckSim® uses a different mathematical solver for torsionally compliant models with more degrees of freedom, which permits twisting of the vehicle frame. The flexing of the frame is described as a bending of a single rung ladder as shown below in figure 129.

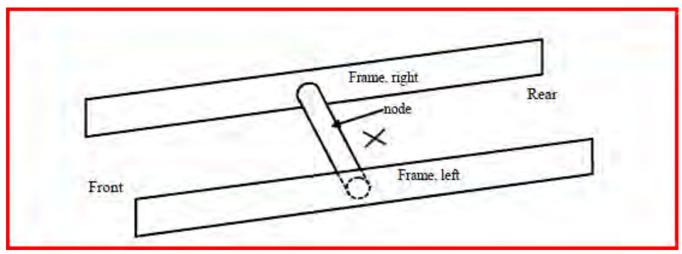


Figure 129. Illustration. Frame Torsion Model [13].

The torsional stiffness of the tractor-flatbed-trailer was documented in the sprung mass sections of the model.

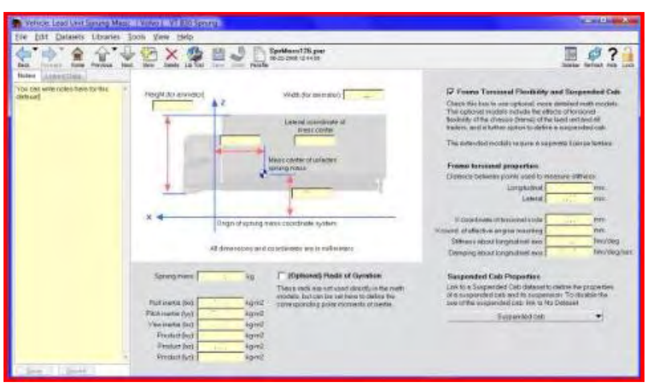


Figure 130. Screen Shot. Tractor Sprung Mass [13].

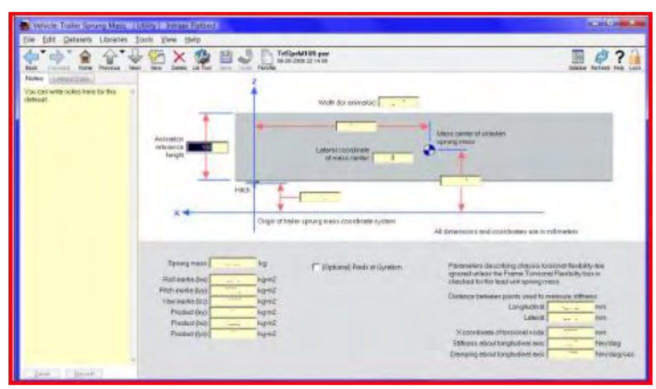


Figure 131. Screen Shot. Trailer Sprung Mass [13].

Constant Radius Test Modeling

Both the low- and high-speed constant radius maneuver tests were evaluated in TruckSim® to determine if TruckSim® could reproduce the vehicle performance observed during the field tests. Because the high- and low-speed tests were designed to evaluate differing vehicle responses, they are considered complementary information with one evaluating sub-limit responses and the other evaluating at-limit responses. Both test cases were therefore modeled to gage TruckSim®'s global accuracy

Low-Speed Constant Radius Test

The low-speed model was executed using a constant radius turn of 75 m and a speed increment of 1 km/h every 3 sec. This closely matches the field radius and is a slightly slower acceleration than the field test acceleration. It was not possible to match the test speed rate because the test speed was not linear due to the 1 percent slope in the test-track surface.

The typical way to view these results is by looking at the amount of steering needed relative to Ackerman steering (see Establishing the Equivalent Tractor Wheelbase). In this case, the model as executed here, has an Ackerman angle of 100 deg at the steering wheel. Steering amplitudes above 100 deg indicate understeering; steering amplitudes below 100 deg indicate oversteering.

As can be seen in figure 132 and figure 133, the TruckSim® model indicates either a slightly oversteering vehicle at the as-measured torsional stiffness or a slightly understeering vehicle for the case when the chassis is artificially stiffened by increasing the tractor and flatbed-trailer torsional stiffness by one order of magnitude each.

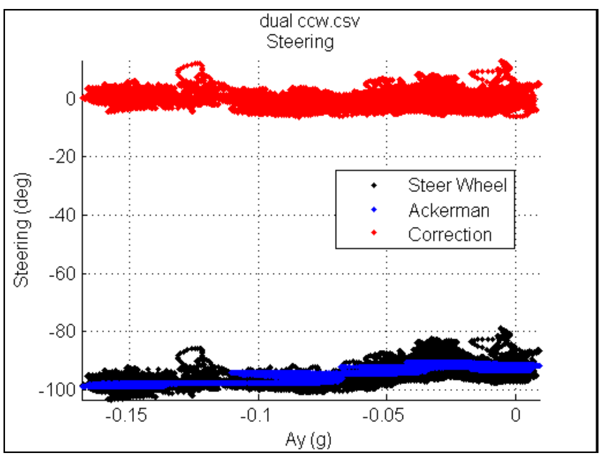


Figure 132. Graph. Field Test Data For Steady-State Testing.

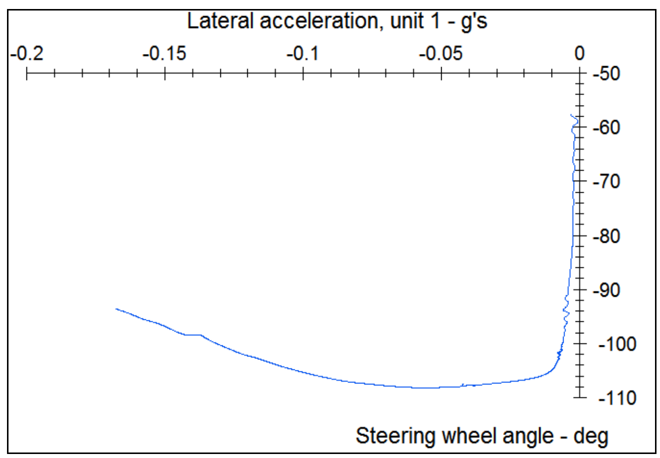


Figure 133. Graph. Steady-State Model Results (As Measured Chassis) [13].

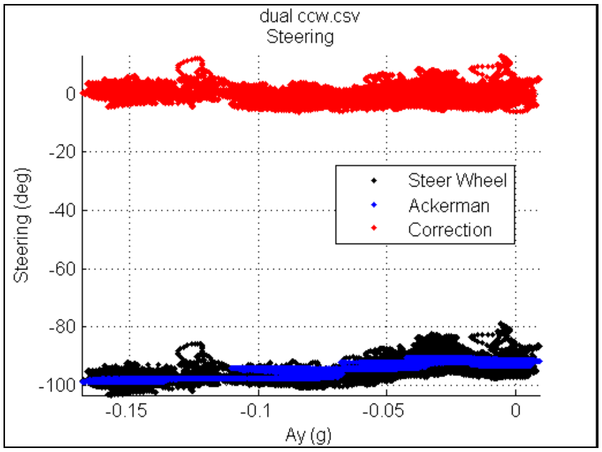


Figure 134. Graph. Field Test Data For Steady-State Testing.

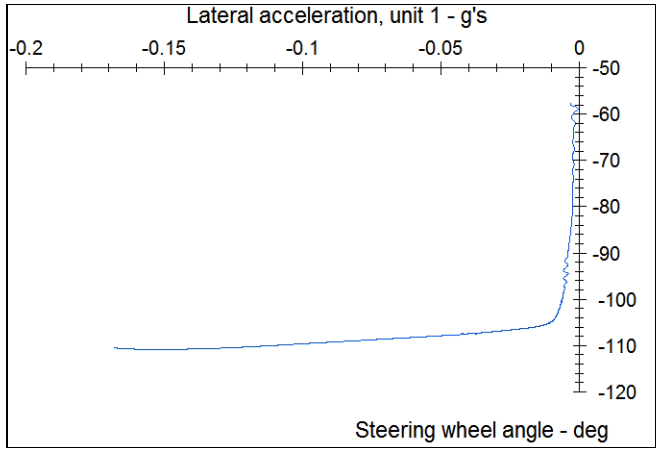


Figure 135. Graph. Steady-State Model Results (Stiffened Chassis) [13].

The indication here is that the model is sensitive to the torsional stiffness and that the response of the model with the true vehicle torsional stiffness is not correct. The artificial stiffening of the chassis does help to reproduce the correct slope in the steering vs. lateral acceleration plot, but it over predicts the amount of understeer by about 5 deg at the steering wheel.

Results of the analysis of the steady-state and transient steering modeling (to be addressed in the next sections of this report), indicate that the effect of torsional compliance has a significant impact on the model's response for all of the maneuvers defined in this test. Understanding and improving the modeling of torsional compliance will be a topic in Phase-B of this project.

High-Speed Circle Test

The high-speed model was evaluated using a slow constant steering rate of 5 deg/sec at the steering wheel with the truck traveling 61 km/h. This was done because the model would not permit full wheel lift using a steady speed increase (the open differential model limited the speed due to the loss of drive traction of the first lifting drive wheel). However, the analysis model is equivalent in terms of vehicle response to the high-speed constant radius field test with lift occurring for the model and the field data at around 61 km/h. Note: the field test was able to achieve wheel lift because the test surface had a 1 % slope which permitted gravity to accelerate the vehicle through the wheel lift event.

In table 33 and table 34 are the field data results for the high-CG and low-CG payload cases. Figure 136 and figure 137 show the results of the modeling where each line represents an axle (1 for steer through 5 for the last flatbed-trailer axle). The plots are of lateral acceleration vs. wheel load. When the lines reach zero wheel load, the wheel lifts.

Table 33. Field High-Payload Circle Test Drive Wheel Lift

Tractor Ay Lift			
Duncan Grouping	**Mean**	**N**	**Name**
A	0.37 g	4	Dual Tires Low-CG Payload
B	0.31 g	4	NGSWBT High-CG Payload
B	0.31 g	4	Dual Tires High-CG Payload

Table 34. Field High-Payload Circle Test Trailer Wheel Lift

Flatbed-Trailer Ay Lift			
Duncan Grouping	**Mean**	**N**	**Name**
A	0.37 g	4	Dual Tires Low-CG Payload
B	0.33 g	4	NGSWBT High-CG Payload
B	0.33 g	4	Dual Tires High-CG Payload

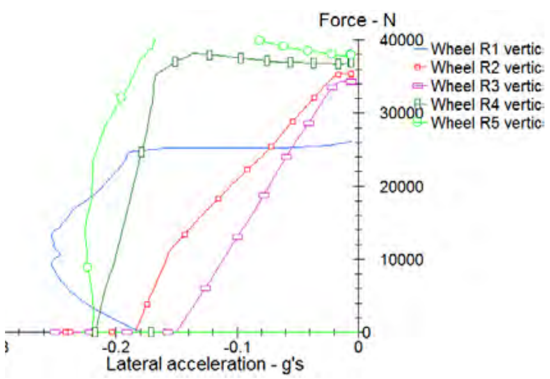


Figure 136. Graph. Wheel Lift By Position For As-Measured Chassis Model – Low-CG Payload [13].

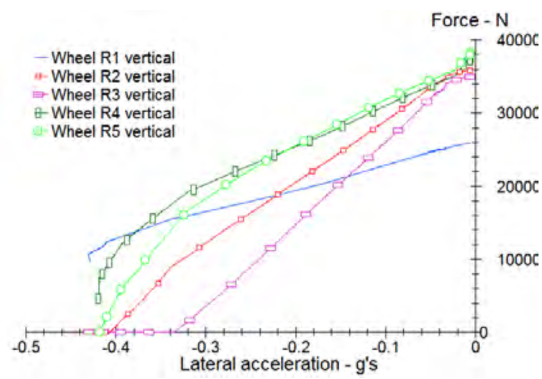


Figure 137. Graph. Wheel Lift By Position For Stiffened Chassis Model – Low-CG Payload [13].

As noted above in the field data summary (see 0), the field testing indicated that the high-CG payload cases had wheel lifts at around 0.31 g for the drive axle and 0.33 g for the flatbed-trailer. The model, using the measured torsional stiffness values for the vehicle, significantly under predicts the wheel lift threshold for both tractor and flatbed-trailer in both load configurations. Increasing the torsional stiffness of the model by an order of magnitude did accurately predict the drive axle lift threshold for the high-CG payload condition, but not the low-CG payload condition. The flatbed-trailer axle lift threshold was not correctly predicted for either load condition using the stiffened chassis.

As can be seen in table 35, the modeling was not able to replicate the absolute lift thresholds for most cases, and was not able to predict the correct relative lift thresholds for the drive and flatbed-trailer sets for any of the cases.

Table 35. High-Speed Constant Radius Maneuver Lift Threshold (Roll-Corrected Lateral Acceleration)

	Field Test	As-Measured Chassis Modeling	Stiffened Chassis Modeling
High-CG Payload Drive Axle Lift	0.31 g	0.14 g	0.31 g
High-CG Payload Flatbed-Trailer Axle Lift	0.33 g	0.22 g	0.38 g
Low-CG Payload Drive Axle Lift	0.37 g	0.15 g	0.34 g
Low-CG Payload Flatbed-Trailer Axle Lift	0.37 g	0.22 g	0.42 g

Step Steer Test

Like the constant radius test results, the modeling efforts for the step steer were also affected by torsional stiffness issues. In general, the same issues uncovered in the high-speed constant radius testing were observed here.

Modeling of the higher CG payload case gave the results shown in table 36 for the as-measured and stiffened chassis models with the high-CG payload.

Table 36. Estimated Step Steer Lift Threshold (Roll-Corrected Lateral Acceleration) – High-Payload Case

	Field Test	As-Measured Chassis Modeling	Stiffened Chassis Modeling
Drive Axle Lift	0.32 g	0.18 g	0.34 g
Flatbed-**Trailer Axle Lift**	0.30 g	0.30 g	0.36 g

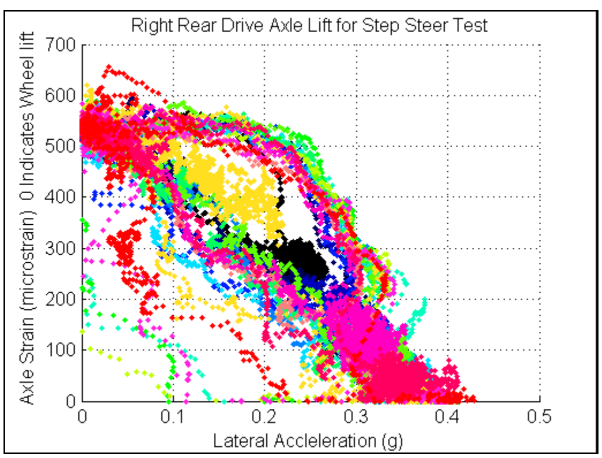


Figure 138. Graph. Step Steer Field Data Lift Point For Drive Axle High-CG Payload.

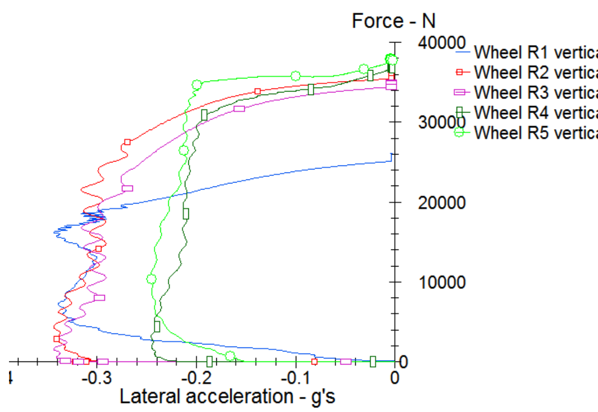


Figure 139. Graph. As-Measured Chassis High-CG Payload Step Steer Lift Point [13].

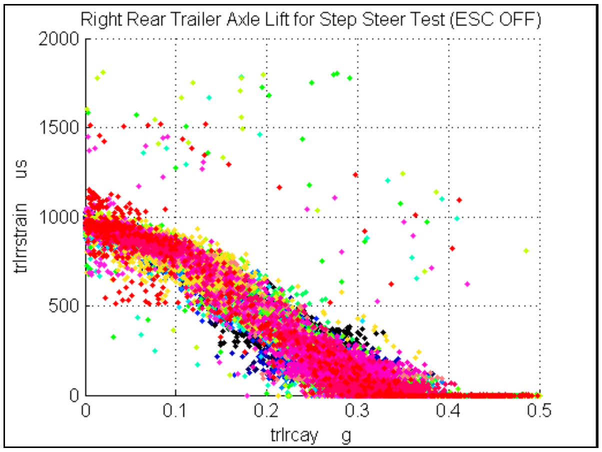


Figure 140. Graph. Steep Steer Field Data Lift Point For Flatbed-Trailer Axle High-CG Payload.

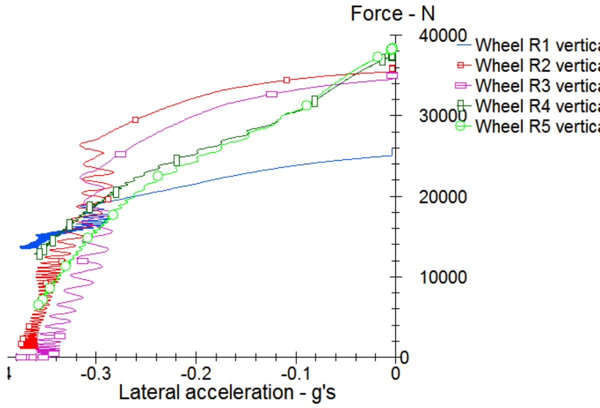


Figure 141. Graph. Stiffened Chassis High-CG Payload Step Steer Lift Point [13].

Figure 138 through figure 141 demonstrate that the model was fairly close to the lift threshold of the tractor for the stiffened chassis case. It was also fairly close to the flatbed-trailer lift point for the as-measured chassis case. However, the remaining lift points were not correctly estimated and neither chassis model correctly modeled the lift sequence of flatbed-trailer axle lift before drive axle lift.

The same evaluation of the model was preformed with the lower CG payload height field data with similar results.

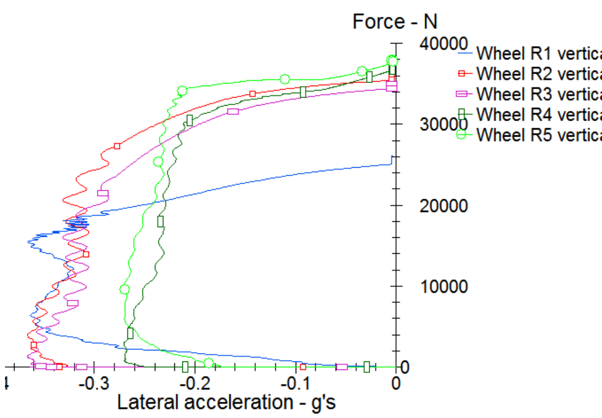


Figure 142. Graph. As-Measured Chassis Low-CG Payload Step Steer Lift Point [13].

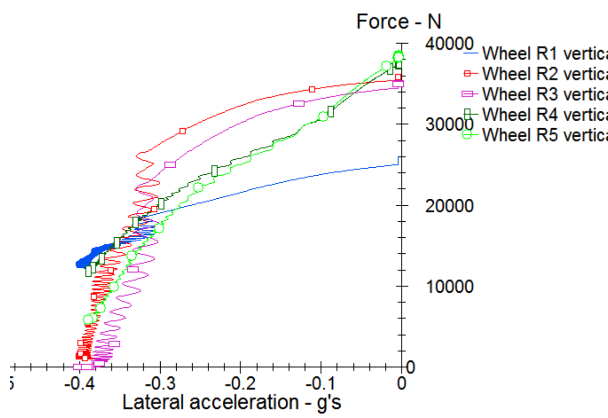


Figure 143. Graph. Stiffened Chassis Low-CG Payload Step Steer Lift Point [13].

Table 37. Estimated Step Steer Lift Threshold (Roll Corrected Lateral Acceleration) Low-CG Payload Case

	Field Test	As-Measured Chassis Modeling	Stiffened Chassis Modeling
Drive Axle Lift	0.39 g	0.18 g	0.38 g
Flatbed-Trailer Axle Lift	0.37 g	0.32 g	0.42 g

The nominal stiffness model predicted that the drive and flatbed-trailer axles would lift at a significantly lower lateral acceleration level than the field test results indicated. The stiffened model predicted more accurate drive axle lift values, but missed the correct lift point for the flatbed-trailer. Finally, both models missed the correct lifting sequence of flatbed-trailer axles lifting before the lifting of the drive axles.

Modeling Limitations and Issues to Be Addressed

According to the simulation results obtained to date, the torsional stiffness of the model has a significant impact on the modeled vehicle behavior. The effect of torsional stiffness is suspected as a reason for the model not accurately predicting the wheel lift points, but it is not known that this is the reason for the modeling errors. Attempts were made to determine why the model seemed to behave as it did, but no solution has yet been found. However, the following items were noted during the modeling process and may be contributing to the stability modeling issues:

1. The tractor and flatbed-trailer both have low torsional stiffness values. There may be limitations to modeling vehicles with this degree of flexibility in more traditional modeling programs.
2. The actual tractor-flatbed-trailer was so closely balanced in terms of roll stability in a steady-state test that both axle sets tended to roll at around the same lateral acceleration. The only other mechanism that could account for both axle sets rolling at the same threshold would be if the flatbed-trailer were passing significant roll moments from one axle to another. But because the flatbed-trailer has a low torsional stiffness, it cannot pass large roll moments. Therefore, the axle groups must be fairly well balanced in terms of lift threshold with respect to each other.
3. In cases where the two axle sets have similar roll thresholds, it would be expected that changing the torsional stiffness in the model would have a minimal effect in steady state analysis cases as there should not be a significant amount of torque transmitted through the chassis. However, the reality is that changing the torsional stiffness in the model had a significant effect on the model's response.
4. The model did not reflect the same magnitude of changes in the lift threshold for changes in the payload CG height as compared to the field data. Because increasing payload CG height increases the overturning moment, the association between applied moment to the chassis and the effect on chassis twist may account for the difference in the response of the model and the response of the test tractor-flatbed-trailer.
5. The lower speed portions of the steady-state simulations are closer to the physical data for both chassis stiffness models. Because the lower speeds have lower lateral accelerations and thus lower twisting moments, twisting of the frame and thus the effect of frame twist on the suspension loads, would be smaller.

The errors in the model are under investigation but are not understood. Future efforts will attempt to resolve the issues between test results and model results for tractor and flatbed-trailer chassis that are torsionally compliant.

TRAILER SOLID MODELING

The flatbed-trailer and its components were scanned using an ATOS II scanning system to accurately establish the dimensional characteristics. The scanning produces a point cloud which is then used to produce cross-sectional splines as the reference for making the solid model. The components, as shown in figure 144 through figure 149, are all reproducible in a dimensionally accurate process, and are shown as examples.

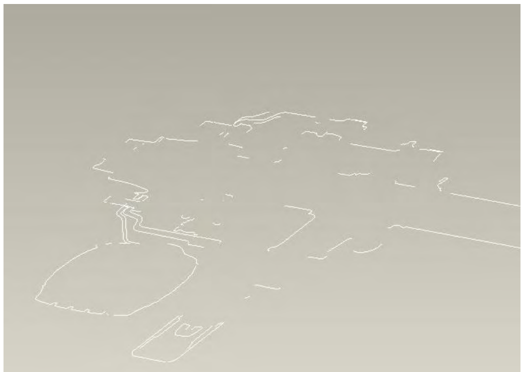


Figure 144. Illustration. Point Cloud Splines from 3D Scanner.

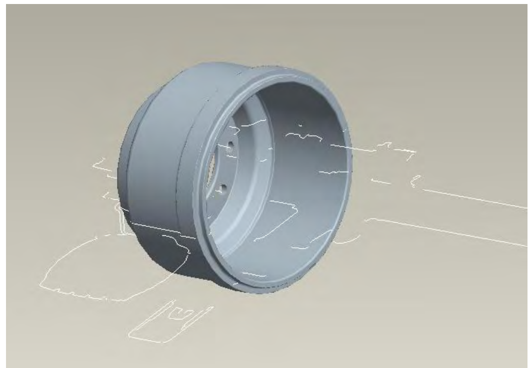


Figure 145. Illustration. Brake Drum and Splines.

Figure 146. Illustration. Wheel Hub.

Figure 147. Illustration. Tires.

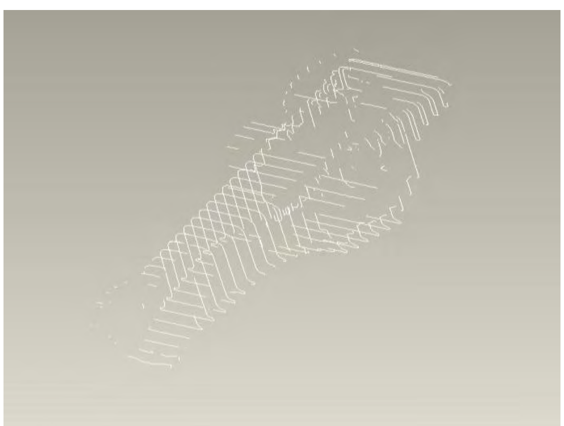


Figure 148. Illustration. Spline Cross Sections.

Figure 149. Illustration. Pro/E Solid Model.

The complete flatbed-trailer structure was also scanned. The model, which was subsequently constructed, is dimensionally accurate as well as material and construction methodology accurate. Several views of the flatbed-trailer are shown in figure 150 through figure 155.

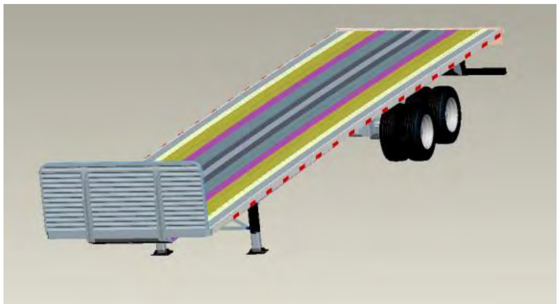


Figure 150. Illustration. Flatbed-Trailer Overview.

Figure 151. Illustration. Flatbed-Trailer Underside.

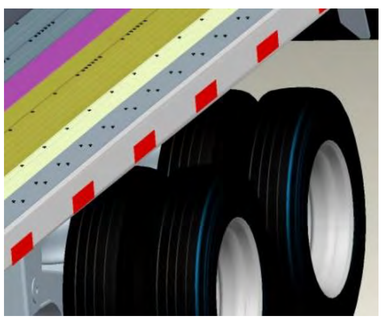


Figure 152. Illustration. Bed Details.

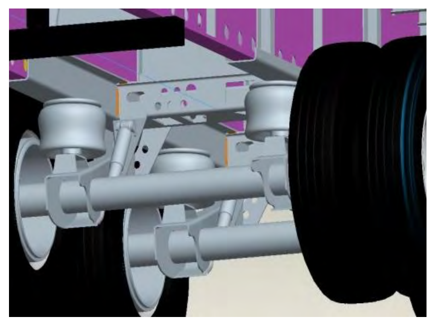


Figure 153. Illustration. Suspension Details.

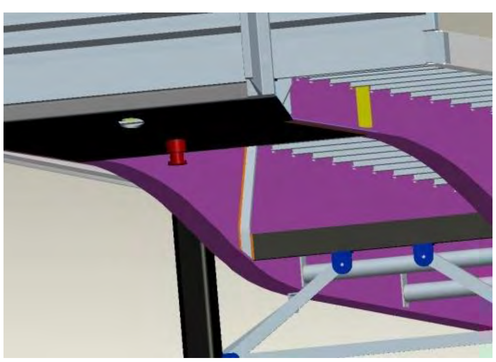


Figure 154. Illustration. Kingpin Area Details.

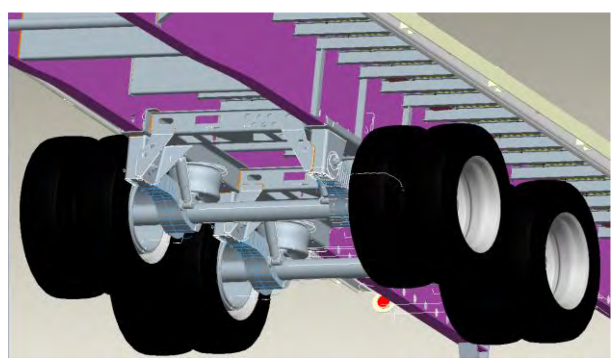


Figure 155. Illustration. Suspension Details.

The accurate solid models enabled further analysis which included Finite Element Analysis (FEA) as well as kinematic modeling. The flatbed-trailer as currently modeled will facilitate modifications and analysis of modifications which can then be fully analyzed in either TruckSim® or an alternative modeling package, saving the team considerable time and expense.

FINITE ELEMENT ANALYSIS (FEA)

The flatbed-trailer structure was analyzed to determine its match with the field testing and the testing performed at MARC. The FEA model of the flatbed-trailer and its performance are shown in figure 156 and figure 157.

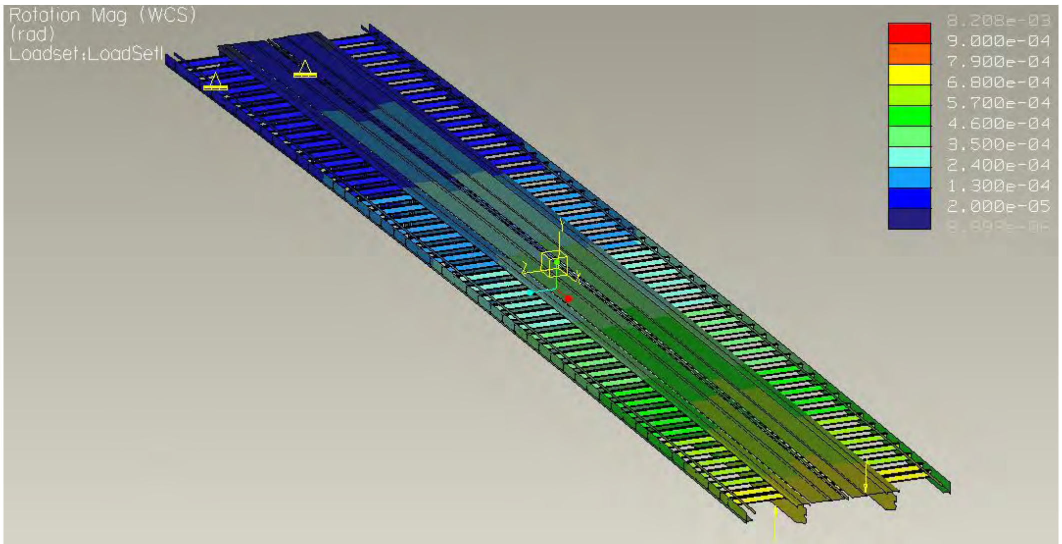
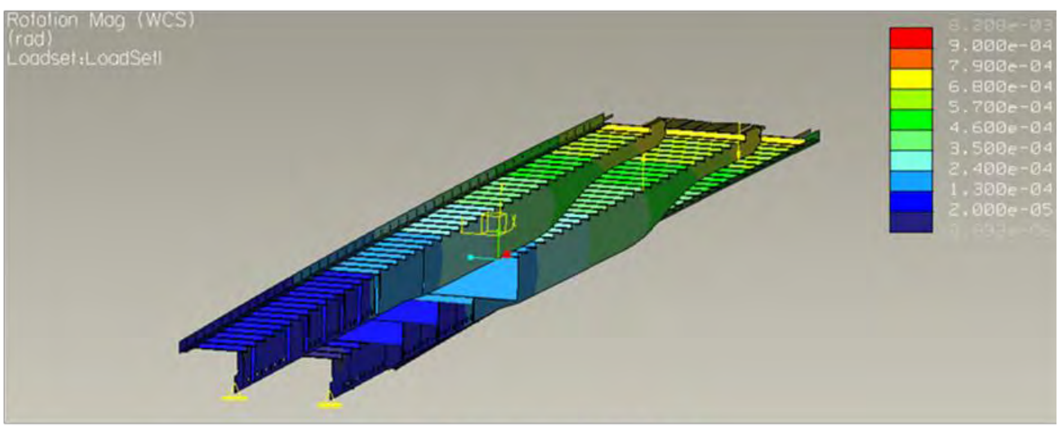


Figure 156. Illustration. Finite Element Analysis of the Flatbed-Trailer Structure.



Results = 1313 ft-lb/deg

Figure 157. Illustration. Finite Element Analysis of the Flatbed-Trailer Structure with the Torsional Stiffness Results.

The FEA results compared favorably with those determined experimentally in field testing as shown in table 38.

Table 38. Torsional Stiffness Comparisons

	Ft-lb/deg	N-m/deg
Field Testing	1388	1882
FEA Model	1313	1784

KINEMATIC MODELING AND ANALYSIS

An ADAMS model of the slider suspension was created using geometry from the flatbed-trailer solid model (see figure 158). The model accurately portrays the characteristics of the slider used in the test trailer and will allow for future development work and analysis. Some sample output is shown in figure 159 where the relationship between vertical suspension travel and shock displacement was studied. The model will be used in development of a library for future development aspects related to the suspension.

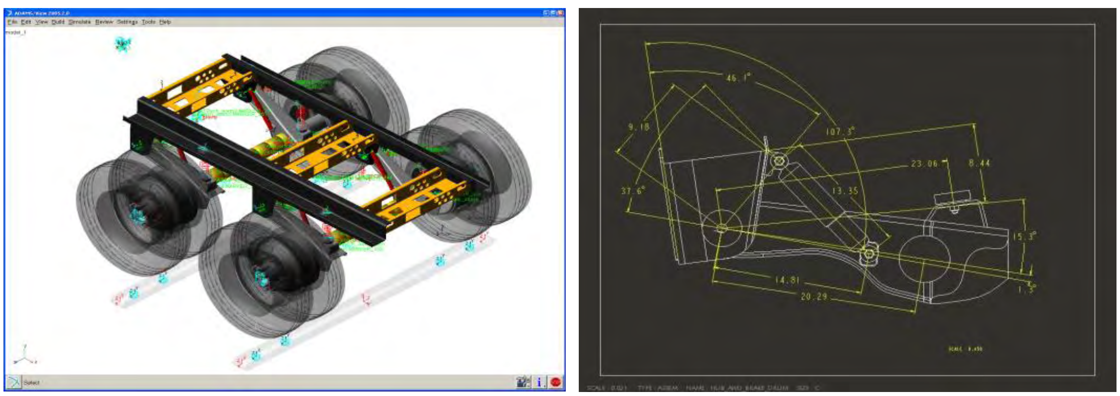


Figure 158. Illustration. ADAMS Flatbed-Trailer Suspension Model with Kinematic Dimensioning.

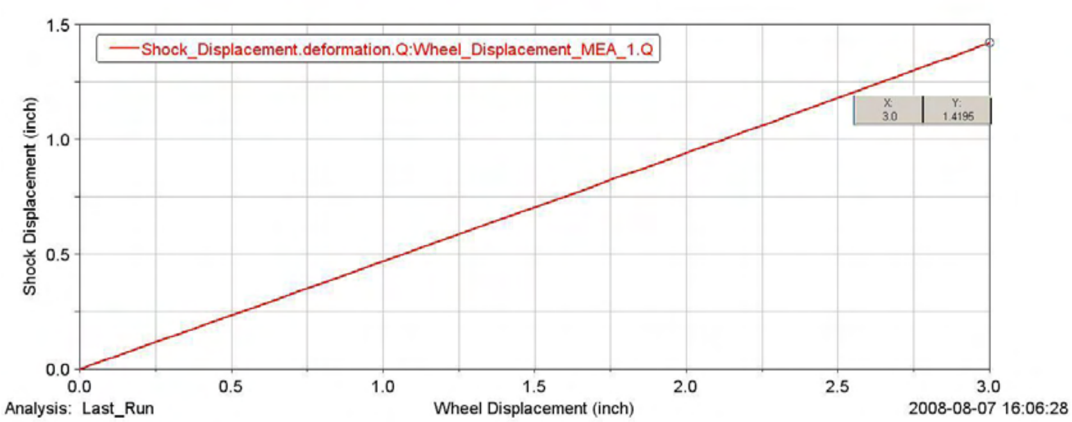


Figure 159. Graph. Trailing Arm Kinematic Analysis.

MODELING SUMMARY AND CONCLUSIONS

The modeling effort of this project was extensive and covered several aspects and approaches. Two separate functional models were developed using TruckSim® (a commercial application). The inputs for this simulation model were obtained from the K&C data that was gathered for the tractor and flatbed-trailer in this phase of the project, as well as manufactures' design/build data sheets. Particular attention was paid to inputs such as axle motions and suspension stiffness, as well as chassis rigidity. The former were essential for the evaluation of stability limits, while the latter was a very important parameter to include in the model since the vehicle used in this study was flexible in torsion.

The TruckSim® model was used to simulate the constant radius and step steer maneuvers. Both the low- and high-speed constant radius tests were evaluated in the model to determine if the model could reproduce the performance of the vehicle as observed during the field tests. The low-speed model was executed using a constant radius turn of 75 m (the same as that in the field test) and a speed increment of 1 km/h every 3 sec (slightly slower than that observed in the field tests). The model was found to be sensitive to the torsional stiffness parameters for the tractor and flatbed-trailer. When the measured vehicle torsional stiffness was used, the model did not correctly predict the understeer response of the vehicle. Artificially stiffening the chassis helped to reproduce the correct slope in the steering vs. lateral acceleration plot, but it over-predicted the amount of understeer by about 5 deg at the steering wheel.

The high-speed model was evaluated using a slow constant steering rate of 5 deg/s at the steering wheel, with the truck traveling at 61 km/h (this speed was selected because it was the lift speed that was realized in the field testing). Using the measured torsional stiffness values for the vehicle, the simulation model significantly under-predicted the wheel lift threshold for both the tractor and the flatbed-trailer in both the high-CG payload and low-CG payload configurations. Increasing the torsional stiffness of the model by an order of magnitude accurately emulated the same drive axle lift threshold for the high-CG load condition, but not for the low-CG load condition. The flatbed-trailer axle lift threshold was not correctly predicted for either load condition using the stiffened chassis. Although good agreement between the results of the test-track testing and the modeling results were not achieved, and the project modelers felt that the tractor-flatbed-trailer was faithfully modeled, further investigation is needed to determine if the lack of agreement stems from the modeling tools, or how chassis compliances affect the model of the tractor-flatbed-trailer. The project modelers noted that the models developed were sensitive to the torsional stiffness of the tractor and flatbed-trailer. Changes in torsional stiffness had significant effect in the model results even though the field data indicates that the front and rear of the flatbed-trailer have the same roll limit. If this is true, the torsional stiffness of the model should be of little importance

As in the previous case, the modeling efforts for the step steer were also affected by torsional stiffness issues which, in general, were the same ones uncovered in the high-speed constant radius testing. The simulation model was fairly close at predicting the lift threshold of the tractor for the artificially stiffened chassis case, as well as the flatbed-trailer lift point for the "as-measured" chassis case. However, the remaining lift points were not correctly estimated and neither chassis model correctly predicted the lift sequence of flatbed-trailer axles lifting before drive axles lifting.

In summary, the torsional stiffness of the model has a significant impact on the modeled vehicle behavior. The effect of this parameter is suspected as a reason for the model not accurately predicting the wheel lift points, but it is not known for certain that this is the reason for the modeling errors. Understanding how the chassis of a flexible vehicle behaves is important for understanding the vehicle's response to various inputs. Furthermore, modeling this torsional flexibility takes considerable care, is not trivial, and is a significant area for future research.

The lesson's learned and experience gained through the research conducted on the various tractor-trailer combinations (box-trailer, flatbed-trailer and tanker-trailer) provide a significant basis on which future class 8 roll stability design, development and testing can be based.

This study provided a wealth of information with regard to how low torsional stiffness vehicles respond to test conditions which included load and load position, tire combinations and specific maneuvers. The modeling efforts allowed the study within this phase to go farther than was possible in the on-track test environment alone, and provided some additional areas for examination as the effort moves toward future design phases. In the on-track testing, for the first test condition which was termed the "flat load," the flatbed-trailer wheel lift was relatively violent and quick (almost a snap lift). With the new load racks, the loads were raised, and discrete loads were placed over the tractor drive axles and flatbed-trailer axles. This produced a much more controlled wheel lift. The modeling effort, with an analysis of the behavior of the flatbed-trailer within the uniquely designed WMU analysis package, appears to also support this response. Further study into the time response of the flatbed-trailer structure may be needed if the torsional flexibility of the flatbed-trailer becomes part of any future recommendations.

The scanning effort, which resulted in the solid models, suspension kinematic models and the finite element models, provided a basis for further study into the tractor-flatbed-trailer as a system. The steady-state modeling effort provided knowledge gains into the tractor and flatbed-trailer suspension interactions and a clearer understanding into how to analyze the tractor-flatbed-trailer as a system with enough agility to assist in parametric investigations

Future Research Needs

Clearly the issues with the modeling need to be evaluated further and, if possible, the model needs to be corrected in future phases of this research. As part of the next phase of this project, a plan for improving the modeling capabilities needs to be included. This plan should also include the investigation of alternate solutions, including simpler models derived from classical vehicle dynamics models employing the physically measured parameters obtained during the Phase-A research. Further examination of the influence of the load position on the periodic behavior of the vehicle system, and its response to both steady-state and transient maneuvers may be relevant, at least from a modeling perspective.

CHAPTER 12. ADVANCED CONCEPTS AND TECHNIQUES

A model was developed at WMU using MATLAB/Simulink/SimMechanics to simulate roll-over tendencies of a tractor and a flatbed-trailer during transient maneuvers such as lane changes. The model can be used to predict impending roll-over by observing the conditions under which the vertical forces on the tires are reduced to where wheel lift could occur. The effects of geometry, mass and inertia, tire models, torsional structural stiffness, and severity of the maneuver on rollover propensity can be studied. The severity of the maneuver is defined by the steering angle profile (specified for the front wheels) and the speed of the tractor-flatbed-trailer.

BASIC ROLL MODEL FOR TRANSIENT MANEUVERS OF A TRACTOR-TRAILER

Introduction

Extensive modeling has been done of tractors and trailers undergoing steady-state turns [14, 15]. The results of these models have been used to predict truck rollover and to study the effects of design variable changes on rollover propensity. This section presents a basic, quasi-dynamic model for simulating the yaw-roll dynamics of a tractor and coupled trailer during transient maneuvers such as lane changes.

Lane Change Model Overview

To complement previous truck rollover modeling for steady-state turns, a model was developed in MATLAB/Simulink/SimMechanics to simulate rollover tendencies during transient maneuvers such as lane changes. The model focuses on the coupled yaw-roll dynamics of the tractor and trailer. All vertical motions and pitch rotations are ignored. The yaw motion of a tractor is initiated using a steering control function along with a quasi-static, lateral-force tire model. Lateral forces are generated using cornering stiffness data from Lawson [14]. The tractor maintains a constant speed using proportional speed control.

The sprung masses of the tractor and trailer are longitudinally segmented. They are connected to the unsprung masses (axles) with springs and dampers to model the suspension. They are connected to each other with springs to model the torsional stiffness of the frame.

As the tractor-trailer executes a maneuver, the sprung masses roll. As they roll, roll torques are transmitted to the unsprung masses (axles) through the suspension model and to adjoining sprung masses through the frame stiffnesses. The roll angles of the unsprung masses are used to redistribute the vertical loads on each side of an axle relative to their at-rest values. These changes in vertical loads produce changes in the lateral tire loads through the cornering stiffnesses and slip angles for those tires. Dual tires are assumed to contact the ground at a single point.

The at-rest vertical load on the fifth wheel F_{zKP} and rear trailer axles are calculated using a moment summation about the king pin and a vertical force summation on the trailer. All four loads F_{zrt} on the two rear axles are assumed identical. The total vertical load on the rear trailer tires is $4F_{zrt}$. The loads on the tractor axles are calculated using a moment summation about the front axle and a vertical force summation on the tractor. The loads on each side of the front axle F_{zf} are assumed identical, and all four loads F_{zr} on the two rear axles of the tractor are assumed identical. The total vertical load on the front axle is $2F_{zf}$, and the total vertical load on the rear axles is $4F_{zr}$.

The model utilizes data for the tractor, trailer, fifth wheel, suspension, and tires. The suspension roll stiffnesses, tire cornering stiffnesses, and fifth-wheel roll moment are all modeled as suggested in Lawson [14]. Suspension roll moments are modeled using two linear spring coefficients. The first coefficient is used until the suspension reaches a bump stop. After reaching the bump stop, an additional moment is produced using a second (larger) coefficient. A 4th order polynomial curve-fit is used to account for variations in cornering stiffness with normal load.

Lane Change Model Details

The lane change model is based on a multiple rigid-body transient dynamic analysis. All stiffness and damping characteristics are lumped at the joints that connect the bodies and between adjoining sprung masses. The lane change dynamics are excited by a quasi-static, lateral force tire model and steering function. The following sections provide more detail on each of these aspects of the model.

Multiple Rigid-Body Transient Dynamic Analysis

The rigid-body dynamic model of the tractor and trailer consists of fifteen rigid bodies. The tractor has eight bodies: a base, three axles (unsprung masses), and four sprung masses. The trailer has seven bodies: a base, two axles (unsprung masses), and four sprung masses. The sprung masses of the tractor and trailer are segmented longitudinally. The degrees of freedom (DOF) of the model are broken down as follows:

Table 39. Model Degrees of Freedom

Item	Degrees of Freedom
Tractor Base (or carriage)	3
Tractor Unsprung Masses (1 DOF each)	3
Tractor Sprung Masses (1 DOF each, masses 3 and 4 required to have same roll angle)	3
Flatbed-Trailer Base Relative to the Tractor Base	1
Flatbed-Trailer Unsprung Masses (1 DOF each)	2
Flatbed-Trailer Sprung Masses (1 DOF each)	4
Total	**16**

The base of the tractor has three degrees of freedom relative to the ground. Its mass center moves in the X-Y plane, and it yaws (turns) based on forces generated by the tire models. No other motion of this body is allowed. The tractor's axles (unsprung masses) are connected by simple roll joints at their mass centers to the tractor base. A sprung mass is connected above each of the unsprung masses also using a roll joint. A fourth sprung mass is located at the sprung-mass mass-center and is connected directly to the base using a roll joint. The mass centers of the sprung masses are generally located above the roll joint.

Torsional springs and dampers are located at the roll joints between the unsprung masses and the base to model tire stiffness and damping, and they are located at the joints between the unsprung and sprung masses to model suspension stiffness and damping. Torsional springs are also used between adjoining sprung masses to model frame stiffness.

The mass center of the trailer base also moves in the X-Y plane, and it yaws (turns) as it is pulled by the tractor. The connecting joint restricts the X-Y motion of the king pin relative to the rear sprung masses of the tractor. As such, it transmits dynamic horizontal loads between the tractor and trailer, but not dynamic vertical loads. However, the at-rest vertical load at the fifth wheel is accounted for in the static force and moment summations.

The trailer's two axles (unsprung masses) are connected by simple roll joints at their mass centers to the trailer base. A sprung mass is connected above each of the unsprung masses also using a roll joint. The third and fourth sprung masses, located at the king pin and the sprung-mass mass-center, are connected directly to the trailer base using a roll joint. As with the tractor, the mass centers of the sprung masses are generally located above the roll joint. Torsional springs and dampers are used as with the tractor to model the suspension roll stiffness and frame torsional stiffness.

The roll moment generated at the fifth wheel is modeled as suggested by Lawson [14]. The moment is applied directly between the sprung mass over the forward rear axle of the tractor and the forward-most sprung mass of the trailer (located over the fifth wheel). It is calculated as a function of the normal force on the fifth wheel, the half-width of the fifth wheel plate, the deflection of the fifth wheel at separation, and the fifth wheel lash. A plot of a typical fifth wheel roll moment versus relative roll angle is shown in figure 160.

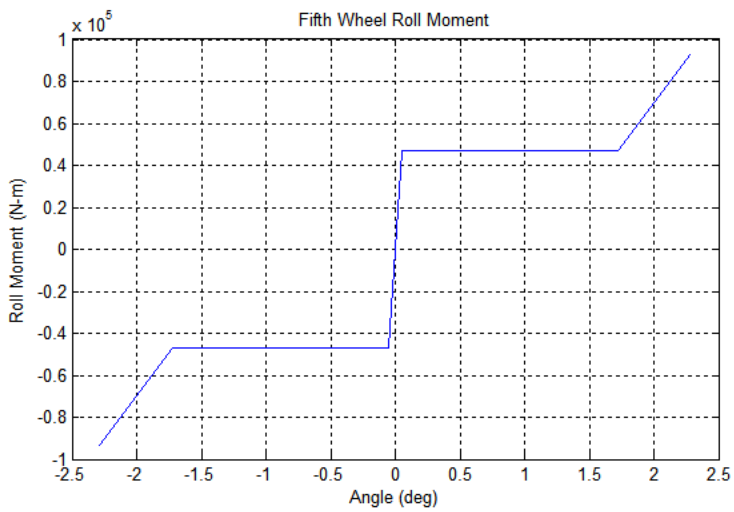


Figure 160. Graph. Typical Fifth Wheel Roll Moment.

Quasi-static Tire Force Model

The tires of the tractor and trailer are connected directly to the base bodies. The front tires of the tractor can be yawed relative to the base of the tractor, whereas the rear tires of the tractor and trailer are always aligned with the adjoining base body. The yaw angle of the front tires relative to the base of the tractor are specified using a steering control function described below.

As suggested in reference [16], the lateral forces on the tires are calculated using equation 24:

$$F_y = C_\alpha \alpha \qquad \textbf{24}$$

Figure 161. Equation 24.

Here, α represents the slip angle and C_α the cornering stiffness of the tire. The cornering stiffness is calculated as a function of the vertical load using a 4th-order polynomial curve-fit to data presented by Lawson [14]. The cornering stiffness curve-fit data for the tires on the two rear tractor axles is shown in figure 162. The slip angle, cornering stiffness and lateral force are calculated individually for each tire.

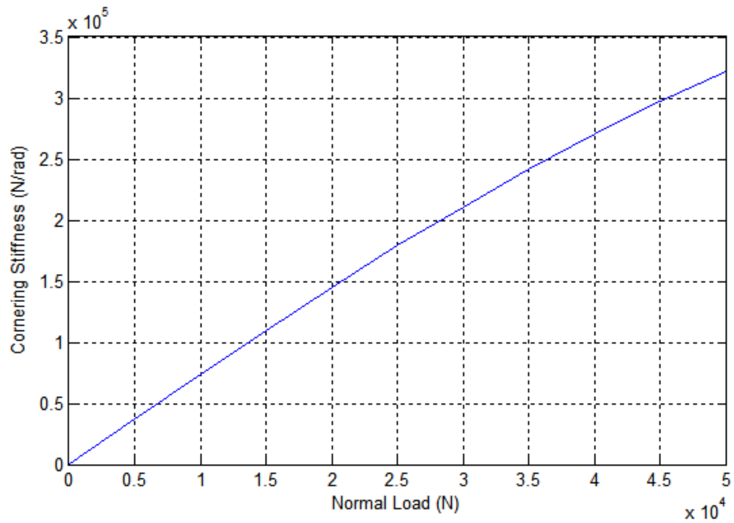


Figure 162. Graph. Cornering Stiffness for Rear Tractor Axles.

Steering Control Function

To provide smooth steering, an angular acceleration profile is used to prescribe physically consistent data for the angle, angular velocity and angular acceleration of the front tires relative to the base of the tractor. The relative angular acceleration is assumed to be the sum of a series of positive and negative square pulses, and the angular velocity and angular position are found by integrating that signal. The magnitude and duration of the individual square pulses can be varied to increase or decrease the severity of the maneuver for a given truck speed. A sample lane change steering angle input for a series of pulses over an 8 sec period is shown in figure 163.

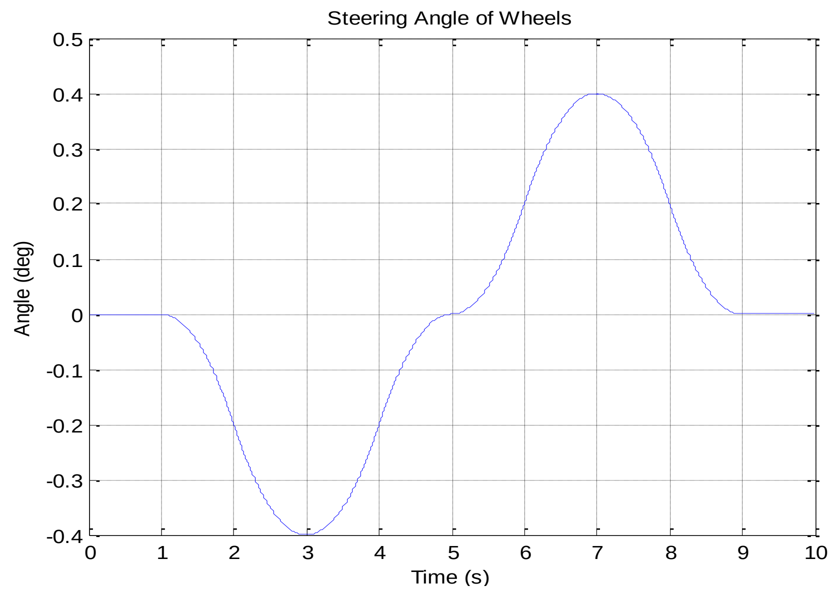


Figure 163. Graph. Typical Steering Control Function.

Summary

A model has been developed at WMU using MATLAB/Simulink/SimMechanics to simulate roll-over tendencies of a tractor and trailer during transient maneuvers, such as lane changes. The model can be used to predict impending roll-over by observing the conditions under which the vertical forces on the tires become small. The effects of geometry, mass and inertia, tire models, torsional structural stiffness, and severity of maneuver on roll-over propensity can be studied. The severity of the maneuver is defined by the steering angle profile (specified for the front wheels) and the truck speed.

ANTI-ROLLOVER DEPLOYABLE AXLE

The purpose of this research is to design a system, method or mechanism that can help improve a tractor-trailer's resistance to rollover. Over the years, from 1994 to 2002, more than one-fifth of all heavy truck crashes involve heavy truck rollover, with roughly 115 cases in 2002 alone, as reported by Evaluation of Heavy Truck Rollover Accidents compiled by Evans, Batzer and Andrews. From these statistics, there is an average of 122 incapacitated and fatally injured victims per year. Therefore in future truck design, it is crucial to design it with an efficient anti-rollover system or device.

Literary reviews show that modern trucks are using computer controlled anti-rollover systems generically called Intelligent Truck Rollover Advisor Systems (e.g., the Bendix ABS-6 Advanced with ESP). The Bendix system is known as the only ABS- based truck stability system capable of recognizing and assisting with rollover, jackknifes and loss of control on a variety of road conditions through advanced sensing and automatic brake system.

Figure 164. Photograph. Bendix ABS-6 Advanced With ESP: A Brake System Through Innovation in Electronics Stability.

There are other patents made by numerous other companies and private researchers in this area, but they use computer programmed systems, and only a few mechanical systems are available. One of the most promising mechanical anti-rollover device is Patent No. 6,588,799 B1 which is the Vehicle Anti-Rollover Device shown in figure 165.

The Vehicle Anti-Rollover Device is a mechanical device that extends the wheel base of the vehicle so as to prevent vehicle rollover when the vehicle tilts. The device comprises a sliding steel bar or rod with a wheel or cylindrical ball joint at the terminal end that will slide out from a vehicle when the vehicle begins to tilt. The bar or rod extends to the side of the front and rear of the vehicle to a point where it touches the road to prevent further tilt or rollover. The sliding bar or rod is enclosed in a sleeve that is attached to the frame, bumper or body of the vehicle, so that the load of the vehicle can be absorbed, preventing tilting or rollover.

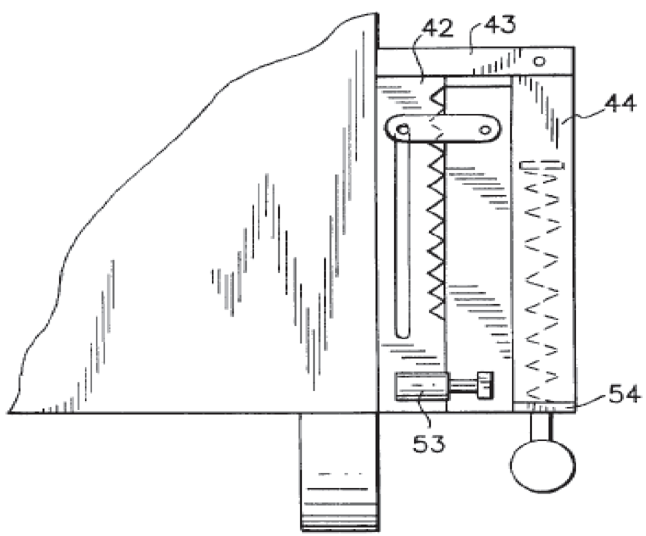


Figure 165. Illustration. A Partial View of the Vertical Bar.

However, the problem with this design is that it is rather impractical. The sliding bar or rod that slides out from the side of the vehicle (figure 166) will be extended more than the width of the vehicle, which can be hazardous to passing vehicles or people. Therefore, the design of a device that uses the same concept but for which there will be no extension past the width of the vehicle was designed.

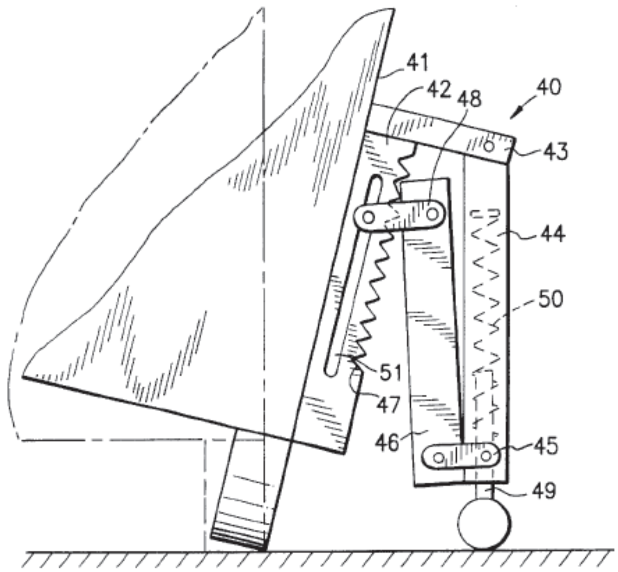


Figure 166. Illustration. A Vertical Mounted Bar Prevents Rollover.

This system is basically a modification of Hendrickson lift axle as shown in figure 167. When the truck tilts, the air suspensions, or airbags in this case, will be deflected, giving way to the weight of the truck. But when the truck is experiencing a sharp turn, the truck will usually undergo a larger tilt than a typical turn and as a result, this will cause more deflection on the airbags. This will usually result in a truck rollover. To prevent this from happening, a pendulum, as shown in figure 168 is added on the bottom of the truck. The end of the pendulum is encased in a steel casing with a slot for the pendulum to slide in. The steel case is welded under the axle. When the truck is tilting, meaning the weight of the truck is tilting, the pendulum will swing so the end of the pendulum will slide along the slot until it is fully extended. The steel case will stop the pendulum from swinging out further, preventing the truck from tilting any further, preventing the rollover of the truck. For simplicity the landing gear could be attached together as part of the feature in the lift axle as well, minimizing the number and weight of additional components.

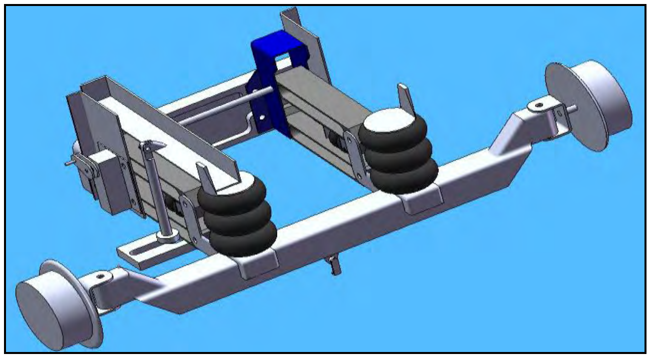


Figure 167. Illustration. Modified Lift Axle.

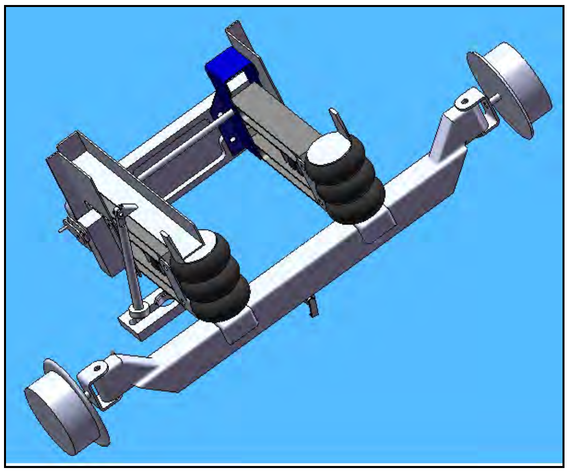


Figure 168. Illustration. Modified Lift Axle (Alternative View).

REPOSITIONABLE 5TH WHEEL

A concept that might reposition the 5th wheel within the tractor frame, as shown in figure 169, has potential merit. The objective would be to optimally position the weight on the truck chassis to add stability to the tractor itself. This concept has to be explored more fully in the future to understand all its implications, and should be modeled within a dynamic system to understand its potential for further development.

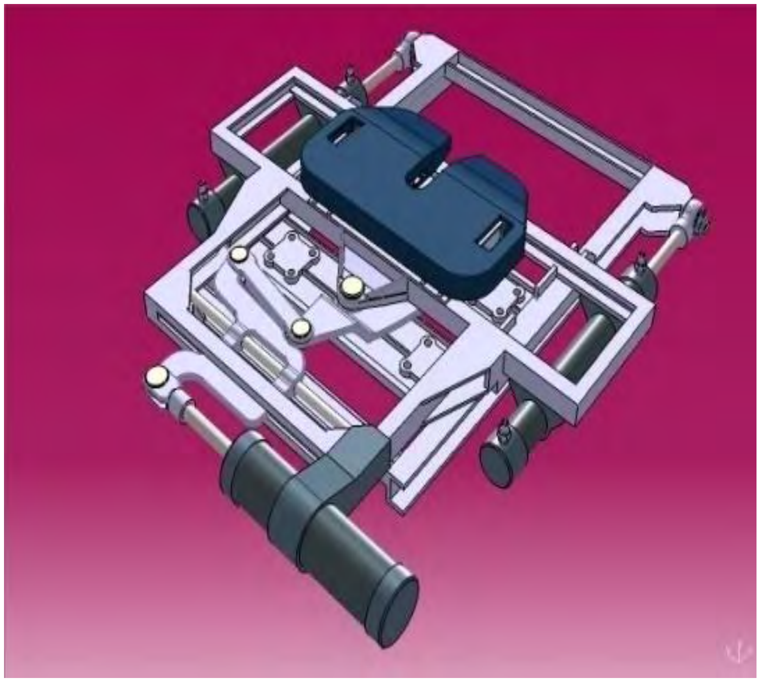


Figure 169. Illustration. Moveable 5th Wheel (Courtesy of Ohio State University).

ACTIVE SUSPENSION CONTROL

An active suspension model has been developed for the purpose of increasing heavy truck rollover thresholds. To function, it utilizes measures of roll angle, speed, and steering wheel angle data, along with all of the spring and damper forces. It uses the roll angle, speed, and steering wheel angle to develop coefficients to multiply the original spring and damper forces in order to better control the roll angle of the truck.

The subsystem to develop the coefficients is the most intricate portion of the model. It uses the roll angle, speed, and steering wheel angle to compute two separate coefficients for the springs and the dampers. It uses subsystems to actuate the system when the roll angle is greater than 0.7 deg, otherwise the system will output coefficients equal to zero. This will allow for a passive suspension when the truck is not experiencing excessive roll in order to maximize the ride quality when the truck is not in danger of rollover. For each coefficient, three parts are developed separately and added to formulate the coefficients.

The first part takes the absolute value of the roll angle and multiplies it by a coefficient for the spring coefficient. The damper coefficient is somewhat more involved. It first checks to see if the speed of the roll angle is low enough, then checks to see if the roll angle is high enough before it multiplies the roll angle by a coefficient. This is designed to check if the truck is in a constant turn at speeds high enough to roll or tip the truck excessively. If either of these conditions are not satisfied, the subsystem passes through the roll angle unchanged. The reason for increasing damping when the truck is in a constant higher speed turn is to decrease the time to steady-state of the roll angle by minimizing any oscillatory behavior after turn entry. It is undesirable for the truck to be leaning back and forth during cornering and this attempts to alleviate such a phenomenon.

Next, the subsystem looks at the speed of the vehicle. The system takes the speed and multiplies it by a coefficient for the springs and a coefficient for the dampers. Finally the subsystem takes the speed of the steering wheel angle. It also uses two different coefficients for the springs and dampers.

The subsystem then adds the three parts of each coefficient and rounds them to three decimal places before passing the coefficients on to the parent system. The coefficients are then multiplied by the spring and damper forces respectively for each axle. The force for each spring and damper is then run though a subsystem to limit the maximum force it can exert before the block diagram (see figure 170) sends the forces back to TruckSim®.

What makes this system an active suspension and not a reactive suspension is the way the system attempts to predict when roll will occur, and prepares for it. It does this by taking into account the steering wheel velocity. If the vehicle is moving down the road with little to no roll, the system will effectively be off. When the steering wheel is turned rapidly, the system stiffens the suspension, preparing for a roll before the roll occurs.

Application of the Active Suspension Model

The active suspension model operates by increasing the force applied by the springs and the dampers of the vehicle. This can be accomplished in two ways. The spring force can be achieved by using an air bag suspension with a compressor. Because the system requires more spring force, the compressor increases the pressure in the air bags and increases the effective spring constant for the suspension. To increase the force from the dampers, standard passive dampers can be replaced with new dampers outfitted with magneto-rheological fluid and electro magnets. The magneto-rheological fluid can change its viscosity according to the magnetic field that is affecting it. This type of a system has already been developed by Delphi for use in cars but could be applied for use in heavy trucks. Magneto-rheological fluid works quite well for an active suspension system. Very little power is required to run it, and response times are on the order of milliseconds.

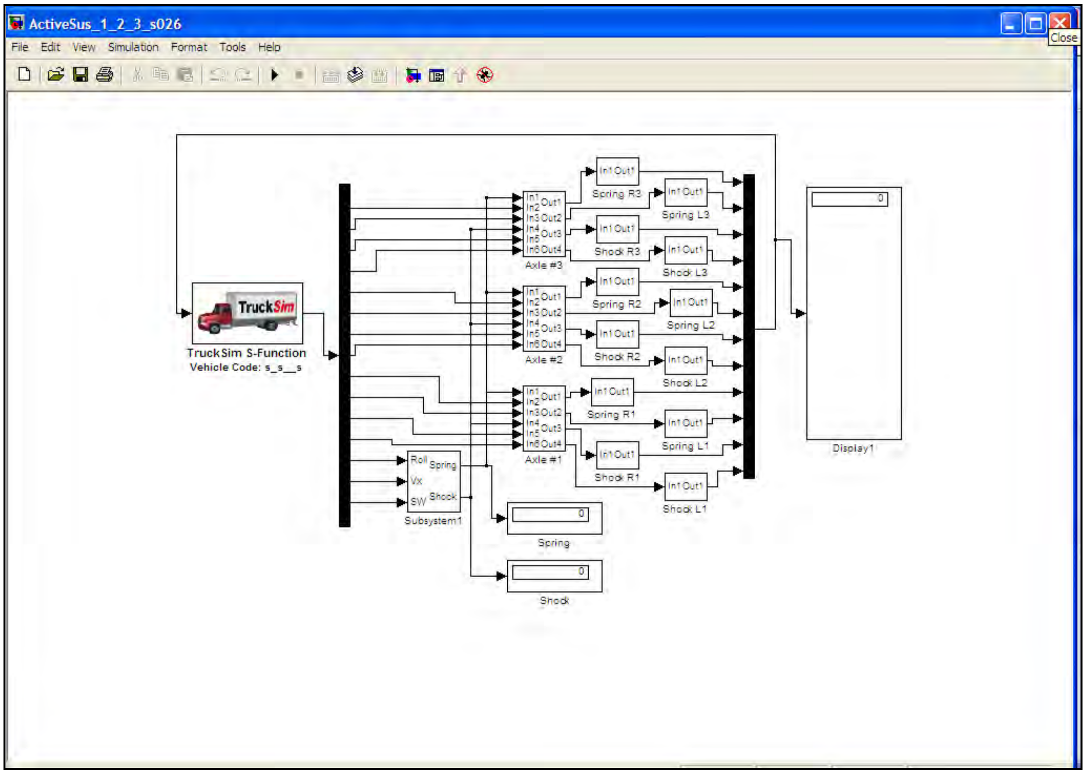


Figure 170. Screen Shot. Simulink Block Diagram.

Shown below are some of the sub-systems of the active suspension model.

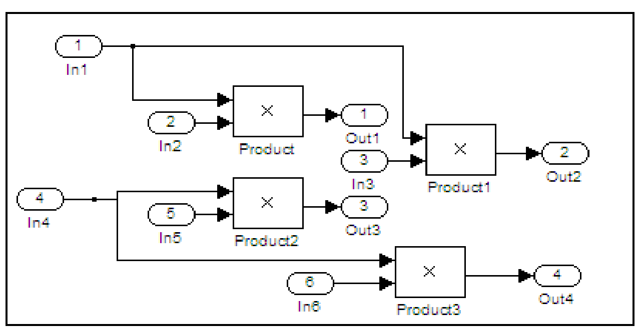


Figure 171. Diagram. Subsystem a.

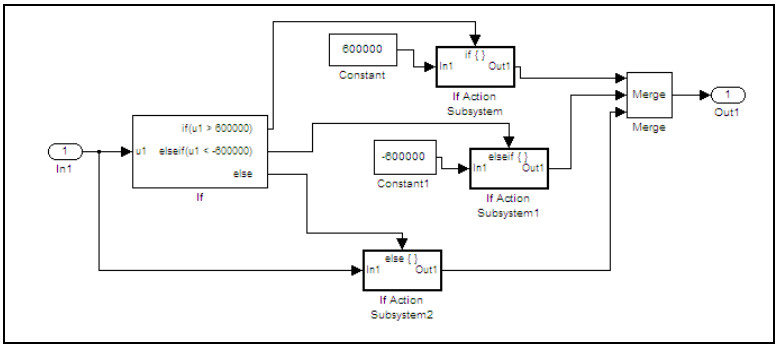


Figure 172. Diagram. Subsystem b.

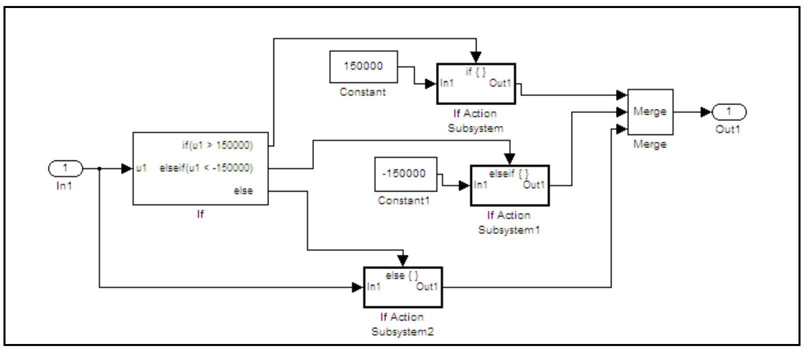


Figure 173. Diagram. Subsystem c.

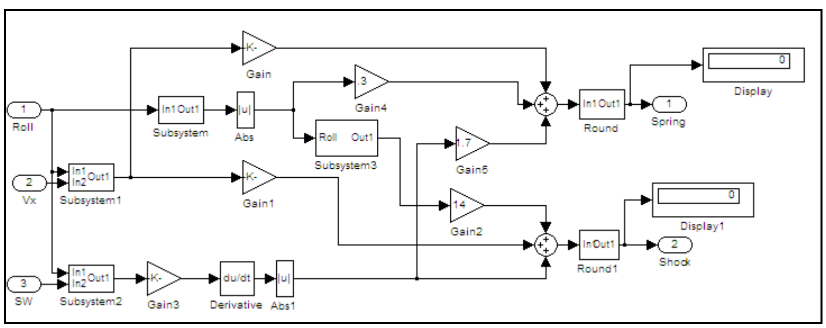


Figure 174. Diagram. Subsystem d.

Figure 175 and figure 176 show the graphs of the roll angles for the cornering analysis, while figure 177 and figure 178 reflect the double lane change maneuver with and without the active suspension control.

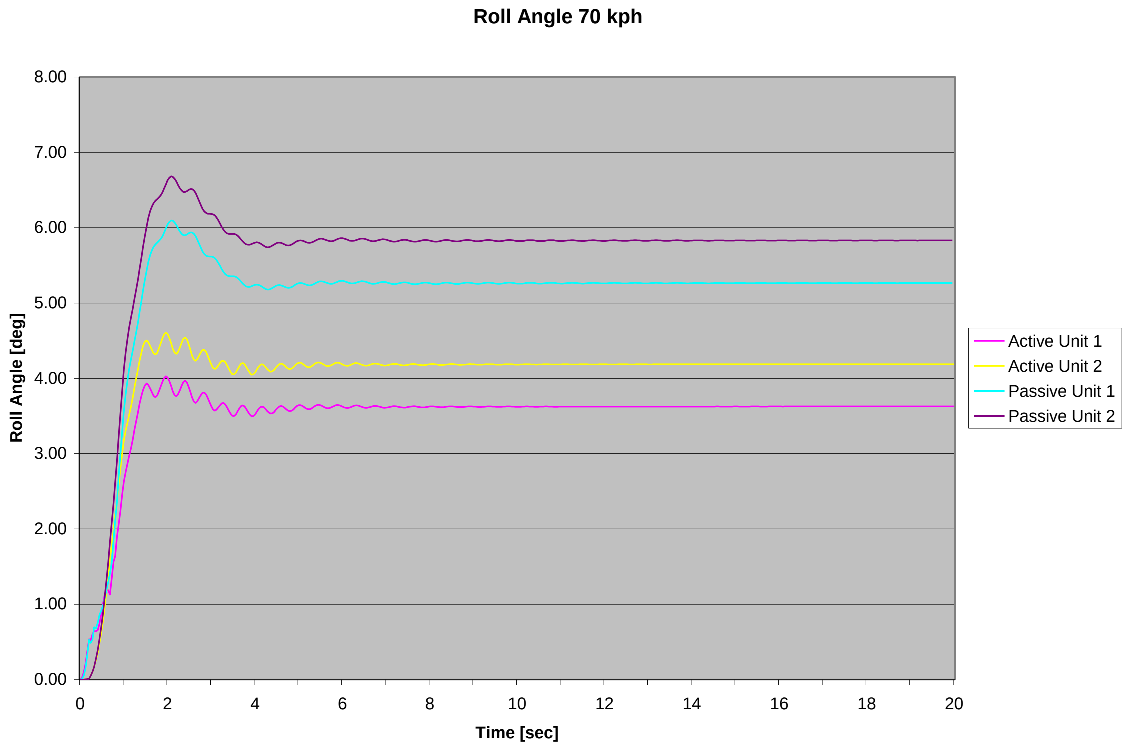


Figure 175. Graph. Cornering Analysis (70 km/h).

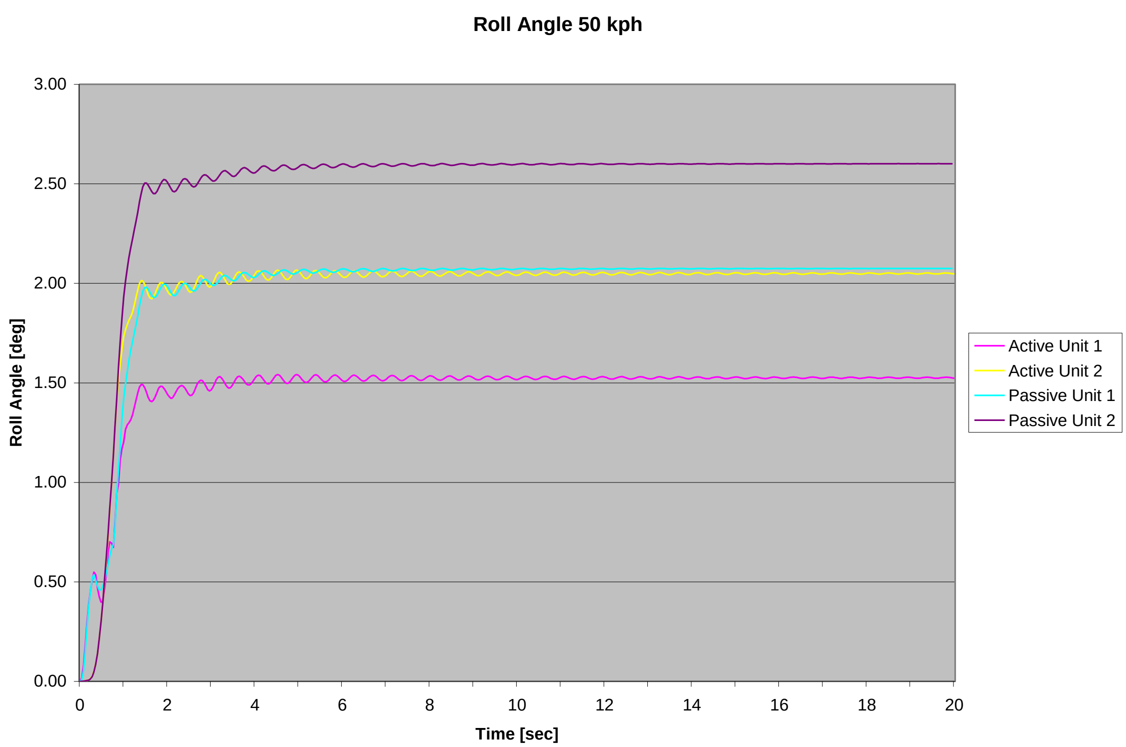


Figure 176. Graph. Cornering Analysis (50 km/h).

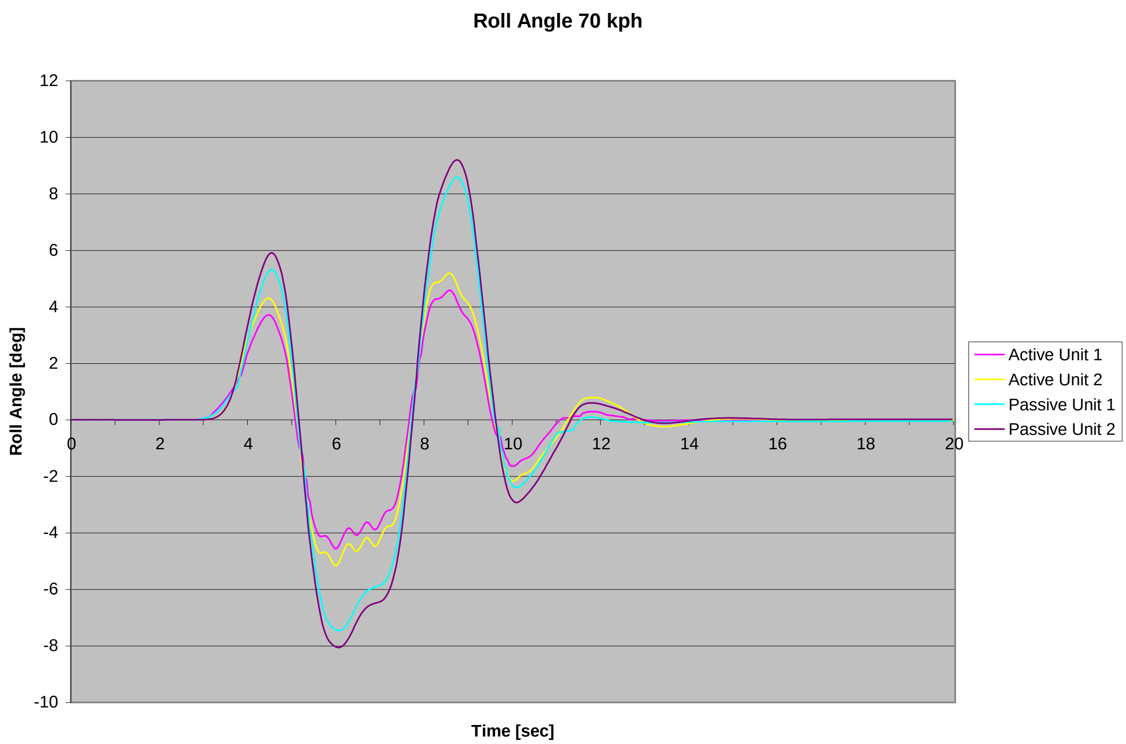


Figure 177. Graph. Double Lane Change (70 km/h).

Figure 178. Graph. Double Lane Change (50 km/h).

Figure 179. Illustration. Maximum Truck Roll Angle for a Passive and Active Suspension at 8.68 Seconds into the 70 km/h Simulation.

<u>Results and Conclusions for the Active Suspension Model</u>

The final results of the developed model demonstrated the feasibility of an active truck suspension and parameters that might be included in its control. The maximum rollover speed for a double lane change maneuver was increased 5 km/h and the rollover speed for constant cornering was increased 4 km/h as well. The major achievement of this active suspension is the decrease in roll angle as compared to the passive suspension under the same conditions as shown in figure 179. As the graphs show, the active suspension does not affect the roll at lower speeds, but as the speed increases and the roll angle of the vehicle increases, the effectiveness of the system increases.

CHAPTER 13. SUMMARY OF RESULTS

<u>REVIEW OF PROGRAM THEME AND PHASE-A EFFORTS</u>

Phase-A efforts were the third in a set of six phases that make up the Heavy Truck Rollover Characterization Research Program. The work was started in 2004 and will culminate in Phase-D, which is currently envisioned to be conducted in the 2011 timeframe. The overarching theme of this research has been to understand and characterize the dynamics associated with a class 8 combination tractor-trailer, including various trailer types (box, flatbed and tanker), as they engage in maneuvers that can elicit vehicle rollover behavior. In order to be able to more fully understand the behavior of the test vehicle, a second theme of this research is to effectively characterize the test vehicle of interest within various commercially available and specially developed software, and to use these models to: a) seek emulation of test-track behavior (a test of the fidelity/depth of the characterization and/or the robustness of the modeling environment, and b) if good agreement is achieved between the models and the performance on the test-track, to suggest enhancements in the vehicle system design that could improve the roll stability of future tractor-trailer designs. The selected models required a significant amount of data about the tractor-flatbed-trailer. Some of this data was collected from the K&C testing conducted by MARC, and the torsional stiffness testing conducted by MARC and WMU.

Suggested enhancements would be based on the experiences gained by the HTRC through this Program, and by working closely with the class 8 tractor and trailer industry to seek out those enhancements which could be practically tested, evaluated, tested and demonstrated.

SUMMARY OF TEST-TRACK TESTING RESULTS

Phase-A on-track testing was difficult to successfully achieve. Three attempts at testing (October, 2007, November, 2007 and January, 2008) were engaged in until a fourth attempt was successfully carried out in May, 2008. Concerns over the safety of a tractor-flatbed-trailer that was felt to be torsionally compliant led to: a) movement of a set of outriggers to a different position, b) the addition of a second set of outriggers, and c) the construction of a customized load rack that distributed the payload into two separate loads (one positioned over the drive axle and the other positioned over the flatbed-trailer axles), and allowed the payload CG height to be raised to allow wheel lift to occur at slower speeds. With regard to item c); slower speeds were desirable because the outriggers were “clamped” on, and the functionality/integrity of the outriggers were suspect at higher speeds.

The testing in May, 2008 involved three maneuvers and produced a significant amount of data that was analyzed by ORNL and MARC. In addition to the data collected on the effects of dual tires vs. NGSWBTs, tractor ESC on and off, and high-CG and low-CG payload heights, other data, such as that related to tractor-trailer characterization (e.g., its over/understeer tendencies) were also derived.

The ORNL and MARC analysis of the data related to the effects of the tires, tractor-ESC and CG height utilized different statistical analysis approaches, but the results of both confirmed the overall results. They were also very interesting because some were contrary to what the team had expected. With regard to the tires: for this tractor-flatbed-trailer, the performance of the vehicle during the test maneuvers (with tractor ESC on and off, and with high-CG and low-CG payload heights) was effectively the same for dual tires and NGSWBTs. For the testing conducted with a tractor-van-trailer in Phases 1 and 2, the performance of the test vehicle with dual tires and NGSWBTs seemed to be different. It should be noted that in Phases 1 and 2, insufficient data was collected to make statistically significant conclusions, however, analyses conducted for each case showed an improvement in roll stability when NGSWBTs were mounted on the tractor, on the box-trailer, or both. A primary difference between the test tractor-trailer combination used in Phase 1 and 2, and the test tractor-trailer used in Phase-A was that the Phase-A tractor-flatbed-trailer was much more torsionally compliant than the tractor-van-trailer used in Phases 1 and 2. Studies in Phase-B to address the performance of a tractor-tanker-trailer, a significantly more torsionally stiff configuration, will provide additional insight into the benefits of tires as a function of torsional stiffness, for roll stability.

The performance of the tractor-flatbed-trailer with the tractor ESC on and off was also studied. For the constant radius maneuver (dual tires and NGSWBTs, and high-CG and low-CG payload height), the tractor ESC functioned as expected, and prevented wheel lift, making the event less severe and safer. The ESC system did not prevent wheel lift in all step steer cases but did appear to reduce the lift duration when a lift occurred. The HTRC was interested in determining the speed margin that might have been gained by having the tractor ESC on; however, this would have required the driver to drive at relatively high speeds in order to attempt to achieve wheel lift with the ESC on for some cases. Funding constraints and safety concerns prevailed and those tests were not performed.

With regard to the lane change maneuver; this is a transient maneuver that involved the driver turning the steering wheel left, then right, very quickly. The data analysis showed that in the first half of the maneuver (quick left turn, and just before the driver makes a quick right turn), the time was so short that the tractor ESC was not able to activate. During the second half of the maneuver, the tractor ESC was able to activate, but wheel lift was experienced for both the tractor ESC on and off situations. The difference between the wheel lift between the tractor ESC on vs. the tractor ESC off situation was that the duration of the wheel lift for the former case was of much shorter duration. Also, for the ESC on cases, the tractor-flatbed-trailer came to a quick stop. An area of uncertainty involved the quick stop of the test vehicle. In Phase B, this issue may be further studied via data from additional data channels (e.g., brake-pedal pressure as a function of time).

The effects of high-CG and low-CG payload height was as expected; however, it was somewhat surprising that a 1 ft change in CG height for the tractor-flatbed-trailer would provide such a large difference in the severity of the maneuvers. In Phase-B efforts, variation in the payload CG may not be possible because of the potential introduction of slosh, and because typically in the tanker cargo industry, driving a tanker-trailer with partially-filled compartments is, if at all possible, avoided, and is therefore not characteristic of what might be found in industry.

SUMMARY OF MODELING RESULTS

Efforts in the modeling area were initiated to build tools that would allow the conduct of "what if" studies in the future related to roll stability design enhancements, or to conduct model-based evaluations of new roll stability enhancing technologies. In order to develop these tools, the tractor-flatbed-trailer has to be faithfully represented within the modeling environment. Some of the critical data required for the models came from the K&C and torsional stiffness testing. These two testing efforts led to: a) a new torsional stiffness procedure developed by MARC for the flatbed trailer on their K&C rig, and b) a WMU-developed field-based torsional stiffness procedure. Such procedures were not publicly available prior to their development in Phase-A.

Once faithfully represented within the selected modeling environments, the testing maneuvers engaged in during the on-track testing could be executed within the model. The idea was to see if the performance of the modeled tractor-flatbed-trailer was similar to the performance of the tractor-flatbed-trailer during the on-track testing. The HTRC modelers chose TruckSim® as the commercially-available vehicle dynamics model to use within Phase-A. Unfortunately, the similarity between modeled performance and test-track performance was not strong. Further investigation into these differences are planned for Phase-B

Other modeling efforts included the development of ADAMS solid-body models of various components of the tractor-flatbed-trailer that were subsequently used to develop finite element models to be used in finite element analyses. These solid-body models will provide considerable flexibility in building software-based future tractor-trailer designs. Also of value will be the MATLAB model of the flatbed-trailer suspension. The use of a more sophisticated flatbed-trailer suspension model, especially in conjunction with a torsionally compliant tractor-flatbed-trailer could help in understanding the complex relationship between torsional stiffness and the flatbed-trailer suspension, and could contribute to understanding the TruckSim® modeling difficulties that were encountered in Phase-A.

CHAPTER 14. CONCLUSIONS AND RECOMMENDATIONS

Phase-A efforts contributed significantly to the knowledge-base that the HTRC has acquired regarding heavy truck rollover. With the completion of Phase-A, the data collected during the on-track testing of a tractor-van-trailer and a tractor-flatbed-trailer is large and provides a strong basis for future modeling efforts. The modeling suite that has resulted from the efforts of Phases 1, 2 and A are also significant, and sets the stage for the advanced concept development efforts to be addressed in Phase-C.

The experiential information gained through the research conducted in Phases 1, 2 and A provides a strong basis for conducting the Phase-B efforts that will be addressing a tractor-tanker-trailer. Experience has been gained with regard to the effects of standard dual tires, NGSWBTs, wider-slider box-trailer suspensions, tractor ESC, and payload CG on roll stability. Additional experience is anticipated through the work with a tractor-tanker-trailer, especially with regard to the combined use of the tractor ESC and the tanker-trailer ESC.

Based on the results of the research conducted in prior phases, the following recommendations are made:

1. Seek the participation in the HTRC of industry representatives (tractor-OEMs, trailer OEMs, first tier suppliers and trade organizations) to reflect real-world experience into the design of an advanced and integrated tractor-trailer that reflects high levels of roll stability.
2. Conduct test-track testing for a tractor-tanker-trailer in Phase-B that is complementary to the research conducted in prior phases.
3. Utilize the same Volvo tractor for conducting Phase-B test-track testing as was used in Phase-A testing.
4. Conduct K&C testing on the tanker-trailer to support faithful representation in a chosen vehicle dynamics model.
5. Investigate the differences in the performance of the TruckSim®-modeled tractor-flatbed-trailer and the tractor-flatbed-trailer performance during the on-track testing with the goal of recommending the future use of TruckSim® or the selection of an alternative vehicle dynamics model.
6. Continue to develop the modeling suite that includes TruckSim® (or an alternative vehicle dynamics model), ADAMS, FEA models, and MATLAB models.

7. Continue the exploration of novel designs and technologies that can enhance roll stability, and to evaluate such concepts within a modeling environment and/or on-track testing.
8. Work in close collaboration with other, complementary heavy truck safety projects (e.g., an NTRCI-sponsored heavy truck run-off-the-road project).
9. Seek the participation in the HTRC of federal agencies that have a complementary interest in HTRC research. Some candidates include the Department of Energy's 21st Century Truck Program (which has safety goals along with energy efficiency and emissions goals), various agencies within DOT that have truck safety interests (i.e., FHWA, FMCSA (Federal Motor Carrier Safety Administration), NHTSA (National Highway Traffic Safety Administration), and RITA), and perhaps TSA (Transportation Security Administration) which has an interest in secure commodity transport.
10. As the HTRC research moves toward an advanced design concept, consider a system's approach that includes elements of energy efficiency, environmental friendliness, and security. Such an approach will garnish more broad-based interest and will not unduly advance heavy truck safety at the expense of energy efficiency, environmental friendliness or security.
11. Conduct cost-benefit analyses for any advanced design concept that HTRC moves forward with.
12. Conduct a Heavy Truck Rollover Workshop that will allow: a) dissemination of information and results from the research conducted by the HTRC, b) a communing of ideas related to heavy truck rollover, c) a compilation of suggested heavy truck design enhancements and technologies for high roll stability, d) the crafting of a move-ahead strategy for an advanced, roll-stable heavy truck, and e) elicitation of interest in future partners for the HTRC.
13. Publish and present NTRCI-approved papers and presentations at well-attended national conferences and symposia to raise awareness of the HTRC research, and to seek partnering opportunities.

FUTURE EFFORTS

Future efforts for the Heavy Truck Rollover Characterization Program will involve:

- The conduct of on-track testing and modeling for a tractor-tanker-trailer, the K&C and torsional stiffness testing of a tanker-trailer, and development of a set of suggested advanced and roll-stable designs and technologies for future considerations; to be accomplished in Phase-B.
- Seeking new HTRC partners from industry, academia and governmental agencies willing to actively participate in HTRC future research phases (Phase-C and Phase-D).
- From suggested concepts from prior HTRC research phases and from other sources, congeal an advanced, integrated and roll-stable tractor-trailer concept in Phase-C.
- From the Phase-C concept; build, evaluate, test and demonstrate an advanced, integrated and roll-stable tractor-trailer in Phase-D.
- Engaging in significant outreach efforts in order to gain visibility for the HTRC advanced design concepts, and to engage with potential partners.

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APPENDIX A: HEAVY TRUCK ROLLOVER LITERATURE REVIEW

INTRODUCTION

Tractor semi-trailer rollovers are usually serious events, often with fatalities occurring during the event [1, 2, 3, 4, 5]. Wang and Council [2] determined that there are approximately 4,500 to 5,000 truck rollovers on ramps (highway or other entrance / exit paths) per year in the U.S. with an estimated cost of around $450 million measured in 1995 dollars. On a per-accident basis, the cost of a rollover event is estimated to be somewhere between $120,000 and $160,000 [28].

Furthermore, the US Department of Transportation [3] noted that 64% of all single vehicle (events other than vehicle to vehicle collisions) tractor semi-trailer fatalities occurred in rollover events. In fact, rollover events were the second most harmful event type with vehicle to vehicle collisions being first [5]. Given the high human and monetary costs associated with heavy truck rollovers, the study of heavy truck stability and operational environment seems to be a prudent activity, especially if such work can be used to improve the rollover sensitivity of such vehicles.

Rollovers are generally divided into two categories: tripped and un-tripped. Because of the complexity of defining and evaluating a “generic” tripped event, this study was limited to un-tripped events. But, since statistical accident analysis shows a correlation between static roll stability and real world rollover accidents [9], it seems reasonable that improvements in tractor semi-trailer roll stability should reduce overall rollover accident rates, both tripped and un-tripped. Based on this premise, this study focused on tractor and semi-trailer design performance evaluation using objective field testing, kinematics testing, and modeling in hopes of identifying ways to improve the rollover sensitivity of tractor semi-trailers.

BACKGROUND AND LITERATURE REVIEW

There has been a lot of research into tractor semi-trailer stability during the last two decades. While some of the research used analytical models to study fundamental stability principles of cg (center of gravity) height, mass, track width, etc. [11, 12, 13, 14, 15, 18, 19], others have used the increasing capabilities of computers to model and simulate tractor semi-trailers under a variety of maneuvers [23, 24, 25, 27]. However, regardless of the tools used for the analysis, all of the studies point to the same stability limiting factors in tractor semi-trailer design and usage. In short, tractor semi-trailers are usually significantly less stable than typical passenger cars, and the probability of a rollover event increases by approximately 23% for every 10% increase in cargo weight carried by the vehicle [3].

Basic Design and Stability-Basic Vehicle Design Constraints

The first, and possibly the most critical, observation with the design of tractors and semi-trailers has to do with the many constraints placed on the design of the vehicles. For trailers these design constraints include dock heights of shipping and receiving centers, flat trailer flooring needs for drive on / drive off of loading equipment, clearance of relatively tall tires required for load carrying capacity, and more. Tractor design limitations include physical constraints such as the size and weight of engines strong enough to pull the heavy loads, attachment requirements for trailer connections, chassis flexibility requirements for managing ground elevation changes (tractors use open differentials), plus the common manufacturing and usage constraints such as component packaging and driver needs. Additional overall design constraints such as weight, width, length, height, and more are defined by the government. Moreover, many of the governmental regulations are based on protection of the road infrastructure and traffic considerations [9, 31]. With the majority of the parameters governing the design of the vehicles concentrated on issues other than stability, it is not surprising that the result is a vehicle with relatively low stability levels.

Rollover Frequency and Lethality

The fact that tractor semi-trailer rollovers are not terribly frequent is a result of the training of the drivers and the general care with which they are operated. But given that some 70% of all commercial goods in the US are transported by truck [31], there are a lot of commercial tractor semi-trailers operating in the US resulting in a significant number of rollover accidents. Further compounding the rollover probability issue is the fact that the likelihood of a rollover increases approximately 23% for every 10% increase in cargo load and 49% for every 10 mph increase [3]. Given that the commercial trucking industry is so competitive, the operators tend to maximize the load carried and minimize the time to transit from pick-up to delivery making rollovers even more likely.

The load effect on stability comes from two parameters: raising the cg height and increasing the roll moment for a given lateral acceleration [18]. This load effect was studied by Robert Ervin and Arvind Mathew [18] for typical van trailers (box trailers) from which they developed the following trends:

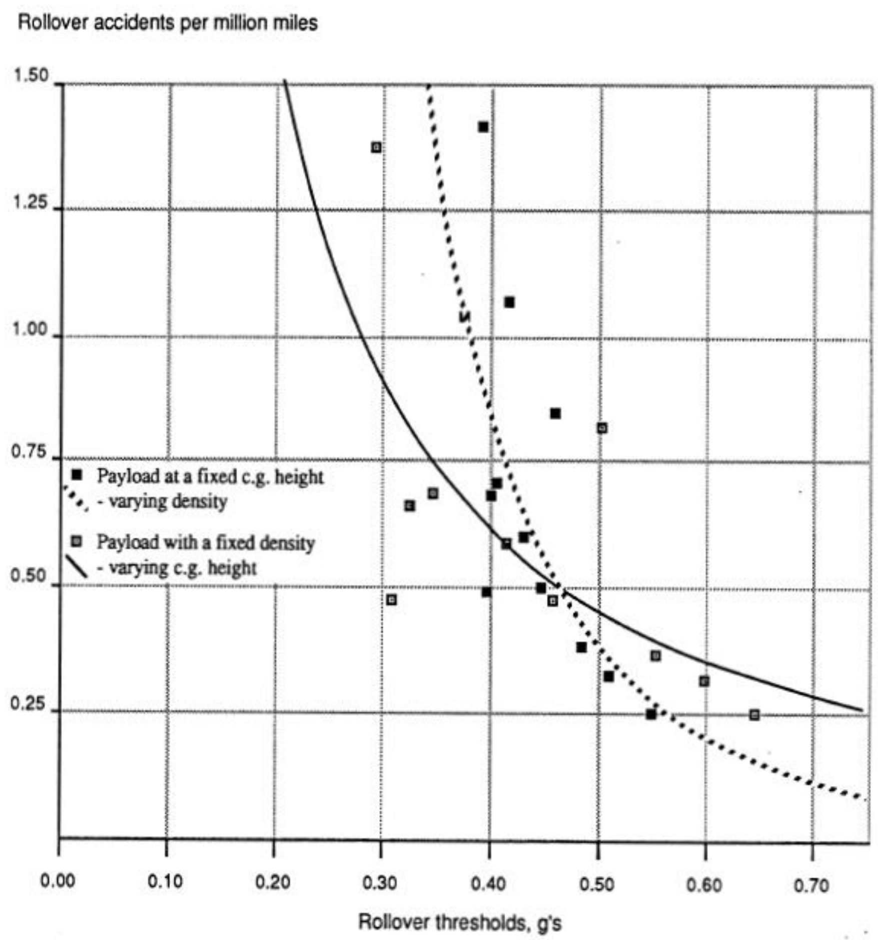

Figure 180. Graph. Rollover Rates vs. Rollover Threshold Estimation [19].

Note: The rollover threshold is essential a parameter which indicates the vehicle's propensity to rollover in a turn. It is described in detail below. Also, this data is largely based on tractors and trailers with a 96" width. Most trucks today use a 102" width so the typical dispersion has probably shifted down and to the right.

Unfortunately, due to the severe nature of rollover events, tractor semi-trailer rollovers tend to be more lethal than other types of accidents. NHTSA data shows that rollovers occur in 3 percent of all crashes involving combination trucks [1]. However, a rollover occurs in 13 percent of all fatal crashes involving heavy trucks [1]. So, though the frequency of rollover occurrences is lower than that of other accident types, the severity of the injuries is typically higher.

Figure 181. Photograph. Post Rollover Tractor [6].

Rollover Propensity and Rollover Metrics

Rollover Threshold

With the multiple design constraints on the configuration of tractors and semi-trailers that do not consider vehicle stability needs, it is not surprising that tractor semi-trailers are less stable than typical passenger vehicles. The most common metric used to evaluate the stability of a tractor semi-trailer is the determination of the vehicle's rollover threshold. The rollover threshold is simply an indication of how difficult it is to overturn a vehicle and is typically reported as the lateral acceleration level needed for a vehicle to tip over [8, 10, 12, 13, 14, 15, 18, 19]. The lower stability levels of tractor semi-trailers are largely due to the fact that they usually have much higher centers of gravity (cg) as compared to passenger vehicles. For reference, passenger cars typically have a rollover threshold of more than 1 g (1 gravitational acceleration unit or 9.81 m/s^2, 32.3 ft/s^2), SUVs and trucks have rollover thresholds of 0.8g to 1.2 g, while tractor semi-trailers typically have rollover thresholds between 0.25 and 0.5 g [14].

Other Grading Criteria

While the rollover threshold is the most used stability metric, it is not the only vehicle stability grading criteria. Other metrics for measuring the stability of tractor-trailers include off-tracking, rear amplification, overshoot, lateral acceleration gain, and yaw rate gain [46, 47, 48, 49, 50, 51]. Of these measurements, off-tracking, rollover threshold, yaw rate gain, and lateral acceleration gain can be measured using a steady-state or a dynamic maneuver. Rear amplification and overshoot are transient response measures. As these parameters are key metrics in evaluating vehicle stability, they warrant clear definitions:

1) Rollover threshold: The level of lateral acceleration (square of velocity divided by the radius of the turn) beyond which vehicle overturn occurs in a steady turn [13]. It should be noted that this definition does not include yaw divergence (jackknife) scenarios [13]. Rollover incidents are more typical for heavily loaded vehicles and jackknife incidents are more typical for lightly loaded vehicles [3].
2) Yaw rate gain: Ratio of yaw rate change to road wheel or, more commonly, steering wheel angle.
3) Lateral acceleration gain: Ratio of lateral acceleration (in g) to road wheel or, more commonly, steering wheel angle.
4) Off-tracking: "Lateral deviation between the path of the centerline point of the front axle of the vehicle and the path of the centerline point of some other part of the vehicle" [47]. Typically this is the first trailer axle compared to the steer axle.
5) Rearward Amplification: "Ratio of the maximum value of the motion variable of interest of a following vehicle unit to that of the first vehicle unit during a specified maneuver" [47]. This can be lateral acceleration, yaw rate, or other vehicle response measurement.
6) Over shoot: Ratio of peak yaw rate or peak lateral acceleration to steady-state yaw rate or lateral acceleration for a step input [46].

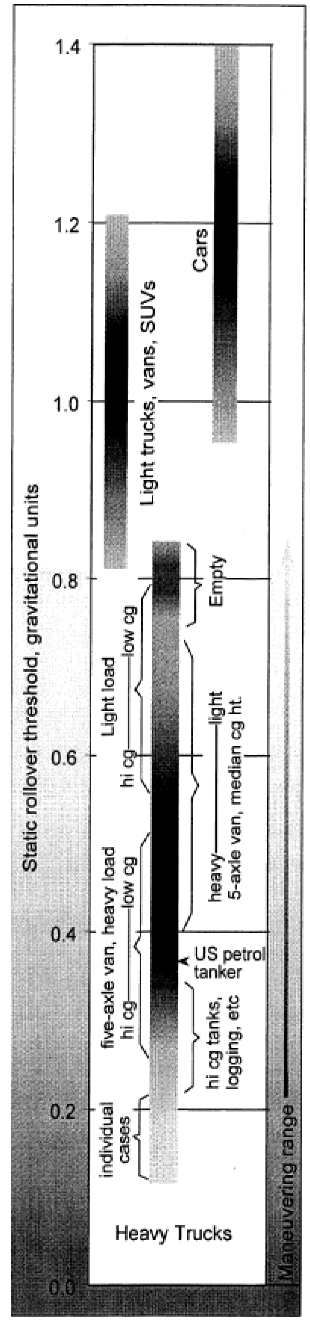

Figure 182. Diagram. Typical Rollover Thresholds [14].

As rearward amplification and over shoot require a dynamic response, they are usually reported for field testing only. Conversely, as the rollover threshold is fairly straight forward to determine, it is reported for almost all stability evaluation methods [8, 9, 10, 11, 12, 13, 15, 17, 18, 23, 25, 26, 27, 30, 31, 44].

Sources of Instability

As noted above, the basic stability of a tractor semi-trailer is limited by its high center of gravity (cg). However, the higher cg is not the only parameter that influences the stability of the vehicle. Other parameters such as vehicle compliances, connection tolerances, and suspension clearances reduce the stability of the vehicle even further. The typical effects of the suspension compliances and component clearances on basic vehicle stability are shown to the right.

This stacking of the compliances and clearances helps to demonstrate why the rollover threshold is so low for many tractor semi-trailers. These vehicles are limited by both functional design constraints and realistic component operational limits. The result is that the typical tractor-trailer rollover threshold is typically in the 0.3 to 0.4 g range.

This reduction in rollover threshold is significant as the resulting vehicle stability begins to approach the field usage conditions of the vehicle. The AASHTO (American Association of State Highway and Transportation Officials) guidelines for highway curve design result in lateral accelerations as high as 0.17g at the advised speed limits [15]. Furthermore, it has also been observed that drivers maneuver their vehicles at well over 0.2 g fairly regularly [15]. This means that in many cases, the rollover margin of the vehicle for the given road conditions is very small.

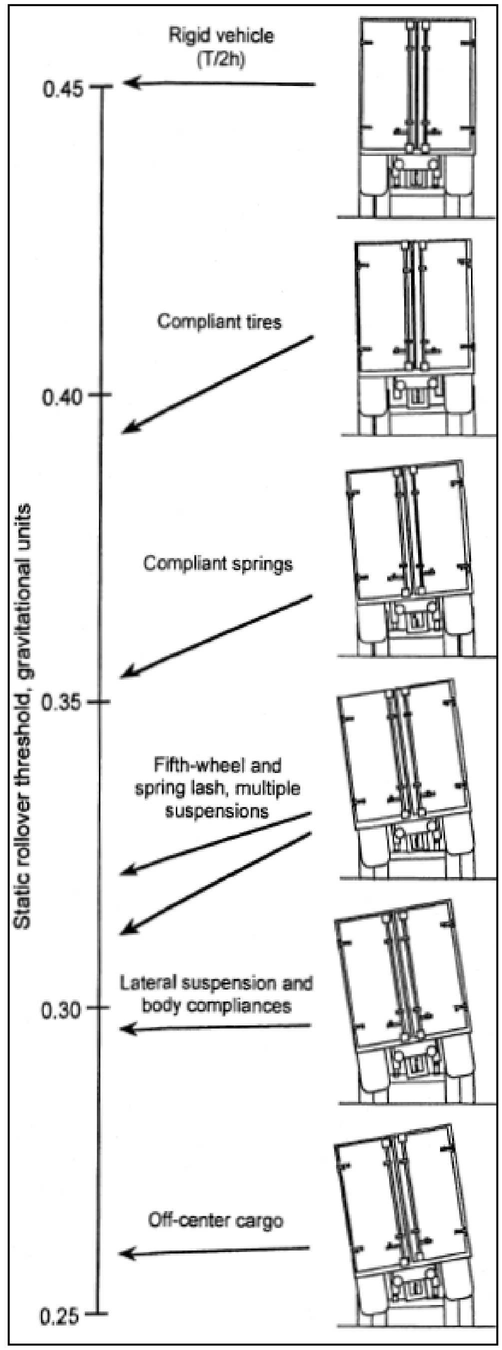

Figure 183. Illustration. Stability Limiting Sources [14].

Roll Moments and Roll Axes

Centripetal Acceleration

When a tractor semi-trailer maneuvers in a steady turn, several sets of reaction forces and moments are developed. The first are the generation of the centripetal accelerations ($A_y = \frac{V^2}{R}$), and resulting forces when multiplied by the vehicle mass as the vehicle traverses the turn and the reacting lateral forces of the tires. This centripetal acceleration multiplied by the mass and cg height from the ground also generates an overturning moment. So, for stability, the two lateral forces (centripetal and tire reaction) must balance and the vehicle must be able to produce a restoring moment to counter the overturning moment. If the lateral force cannot be countered, the vehicle slides off the road (typical passenger car stability limit). If the restoring moment cannot be generated, the vehicle rolls over (typical truck stability limit).

Segmented Vehicle

For simplicity, it helps to think of the tractor semi-trailer as a segmented vehicle where the reaction forces and moments of the differing axles (or axle groups) do not interact. While this assumption is not technically correct; it makes the analysis simpler and the axle interactions will be addressed later. Using this approach, each axle (or axle group) can be treated as an independent system.

If the vehicle were rigid, then the only reaction moment needed to prevent the vehicle from rolling over would have to come from the compliant tire reaction forces. As the vehicle's cg is pulled outward, the cg moves laterally shifting weight from the inside wheel to the outside wheel. This is accomplished by the vehicle rolling about the ground plane so that the cg moves toward the outer wheel. Note that there are two destabilizing forces generated here [13]. The first is the centripetal force; the second is the gravitational force of the vehicle acting on a vertical plane that is closer to the outer wheel.

The overturning moment is resisted by the imbalance of tire normal forces which generates a moment in the opposite sense as the overturning moment. Thus, up to the point of rollover, the restoring moment is a self-induced stability reaction where the vehicle's destabilizing loads are compensated by tire load re-distribution. Of course, once the entire static axle load is transferred to the outside tire, the peak restoring moment is generated and further roll of the vehicle will not produce a higher restoring moment (rollover threshold).

Unfortunately, the vehicle is not rigid, and the sprung mass (mass above the suspension) is supported by springs. Just as the tires had to resist the roll moment by re-distributing load, the springs have to re-distribute load from the inside to the outside of the vehicle. So, similar to the cg rotation about the ground plane, the sprung mass rotates about the suspension roll center (generally at the axle). Finally, as the suspension springs are closer together than the tires, the springs have to generate a greater load transfer to compensate for the reduced lever arm of the spring spacing.

This sprung mass rotation produces two new reductions in overall vehicle stability. As the sprung mass rolls about the suspension center, the cg moves even further outboard which further increases the destabilizing moments at the ground plane as the cg moves further from the axle center. Second, as the springs transfer load to compensate for the overturning moment, they reduce the normal load that the tires can transfer.

From this it is clear that any source of compliance requires the generation of a reaction moment and all of the reaction moments add to generate the total reaction moment needed for stability. Additional compliances include such items as the fifth wheel and spring lash. This final reaction moment is the moment generated by the load fraction of the tires at the ground plane. See "Rollover of Heavy Commercial Vehicles" [15] for a good description of the mechanics.

Axle Rollover Threshold

Simplifying the model a little so that sprung mass (mass above the suspension) and un-sprung mass (the suspension mass) are combined, a simplified moment balance equation can be developed to determine the maximum amount of lateral acceleration the vehicle can respond to. Here M is the vehicle mass, h_{cg} is the cg height, g is gravity, T is the track width, and Δy is the cg offset defined in figure 184.

$$A_{y\_\max} * M * h_{cg} = M * g * \left(\frac{T}{2} - \Delta y\right)$$

Noting that the mass can be canceled gives:

$$A_{y\_\max} * h_{cg} = g * \left(\frac{T}{2} - \Delta y \right)$$

This indicates that the maximum lateral acceleration the segmented vehicle can resist is a function of the cg height (h_{cg}) and the system compliance (Δy). Here A_{y_max} is, in effect, the rollover threshold.

Based on this, the effect of track is easy to gage: Increase the track width and the rollover threshold goes up. Likewise reducing the cg height makes the rollover threshold go up. The effect that is a little harder to see is that increasing the stiffness of the vehicle or raising the sprung mass rotation point increases the rollover threshold by reducing the Δy term [13].

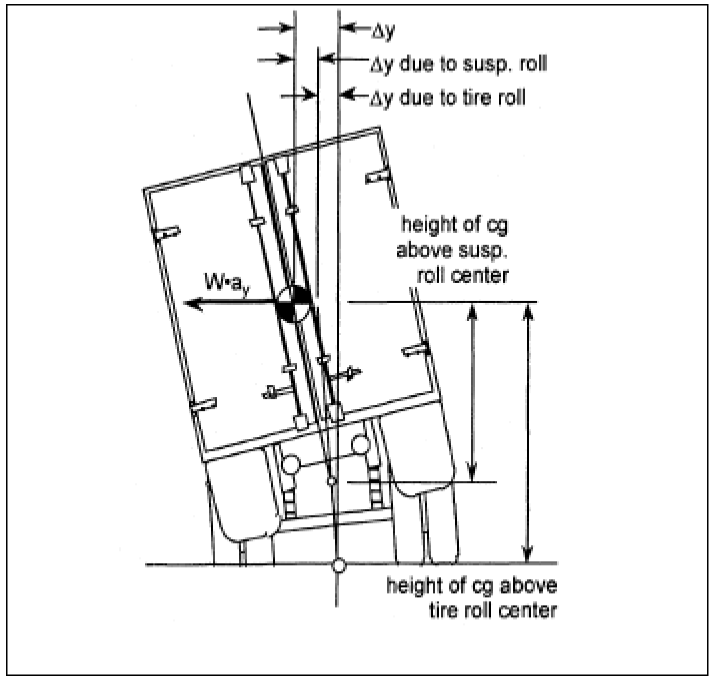

Figure 184. Illustration. Roll Displacements and Reactions [14].

Roll Sequence

Typically, the cg height of tractor semi-trailer increases as one moves from the steer axle towards the rear of the vehicle [38, 43, 50]. The front axle load is dominated by a relatively low height engine which makes the front axle cg fairly low. The drive axles are of significant mass (with a low cg height) so that the total drive axle load cg height is lower than the cargo load cg height. Finally, the trailer axle load is essentially the cargo load and the cg height is nearly the cargo cg height.

This rise in cg height would indicate that the trailer tires would have the lowest rollover threshold, followed by the drive axles, and then the steer axles. To help improve the rollover threshold of the drive and trailer axles, the stiffness of the axles tends to increase from the steer axle toward the trailer axles [13, 14, 20, 30]. Along with the suspension stiffness variation, the sprung mass rotation height also increases from front to rear on the truck [20, 43, 50] to help improve stability. The roll center variation comes from the fact that the front suspension is low (below the engine), the drive axle suspension is relatively low to fit under the fifth wheel, and the trailer suspension is under the trailer bed and above the axles.

Both the increased rear suspension rates and higher rear axle roll axes reduce the Δy term in the stability equation for the higher cg height cases which raises the rollover threshold for the drive and trailer axles. However, despite the attempts to improve the rear axle rollover thresholds, most tractor semi-trailers rollover starting at the rear of the vehicle and move forward to the steer axle [8, 14].

Coupled Vehicle

As mentioned above, the axles do not act independently of each other. As the body of the tractor semi-trailer rolls about an axle, it influences the nearby axles through the torsional rigidity of the vehicle. However, relative to the suspension stiffness, the effect of the torsional rigidities is small [30]. But this does not mean that the vehicle's torsional rigidity is not important [14, 21, 29, 30].

For a static or steady-state lateral acceleration, the roll of the vehicle will generate a restoring moment at each axle. If the entire vehicle is modeled as a single mass with a single cg height, the global overturning moment can be evaluated as a simple one axle model. In this analysis, the restoring moment from each "axle" is assumed to be additive with the other axles. This is accomplished by the vehicle's torsional stiffness carrying restoring moments from axle to axle. So the maximum roll possible is defined by the sum of all the restoring moments. This maximum roll is usually not reached as the chassis is not very efficient at transmitting the torque from axle to axle.

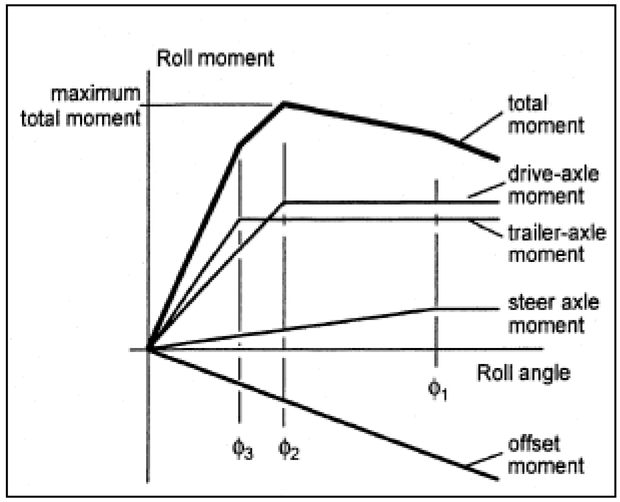

Figure 185. Illustration. Roll Moment and Wheel Lift [14].

In general, the vehicle does not have to carry much of a torsional load under static conditions as each axle responds to the overturning moment of the mass about that axle. What will be observed is that the vehicle leans a bit further where the cg is higher to compensate for the higher overturning moment. The chassis torsional stiffness converts the difference in local roll angle of the vehicle into a torque. In this way the moments from each axle effectively sum to produce the total restoring moment of the vehicle. This additive restoring moment progression is show in the figure above.

As the lateral acceleration increases the vehicle rolls further as it reacts to the increased overturning moment. However, at some point, an axle will reach its restoring limit (usually the trailer axle) and the inside tire will lift off the ground. At this point, the remaining axles must account for the portion of the overturning moment above the peak restoring moment achievable by the lifting axle. This "extra" restoring moment must be first generated by a nearby axle and then transmitted through the torsional "spring" of the vehicle.

It should also be noted that when an axle lift occurs, the effective restoring moment slope (moment generated per degree of roll) decreases as one (or more) of the generation sources (axles) is no longer contributing. Thus an axle lift not only reduces the effective total vehicle roll stiffness, but also requires the chassis to play a much more significant part in transmitting the vehicle stability torques. The same stability reducing process holds with the next axle to lift and so on until the remaining axles (usually the front axle) cannot resist the overturning moment and the vehicle rolls over.

The last important item to note is that this carrying of torsional stiffness from axle to axle can be significantly more important in transient modes. For static modes, each axle is subjected to similar overturning moments as the vehicle has as a uniform lateral acceleration. In transient modes, one part of the vehicle may experience much higher overturning moments from uneven lateral accelerations requiring the other axles to contribute to overall stability. But regardless of the nature of the acceleration (static or dynamic) if the other axles cannot accommodate the stability needs of the limiting (lifting) axle or the chassis cannot transmit the needed torque from axle to axle, a rollover occurs.

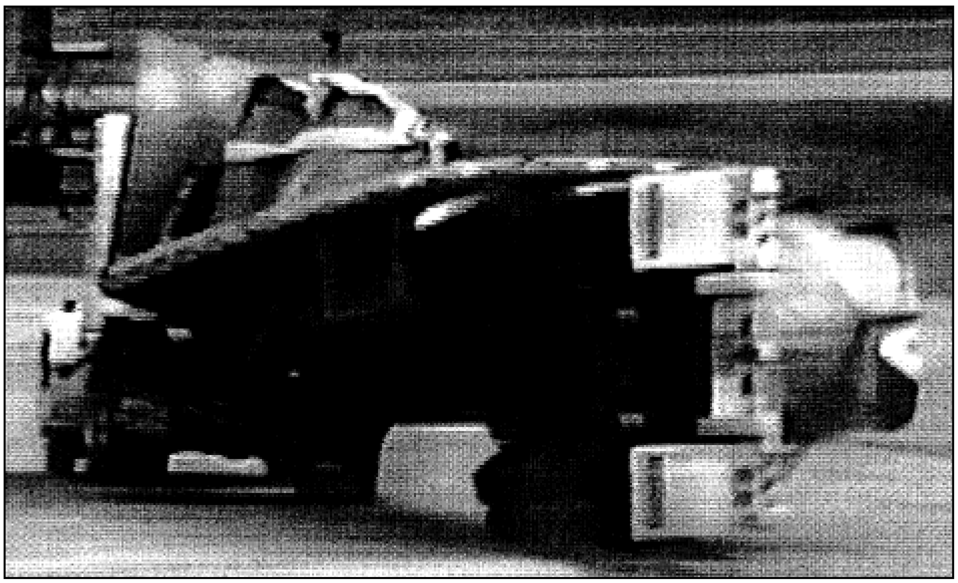

Figure 186. Photograph. Example of a Tractor Semi-Trailer Rollover Scenario [14].

TRACTOR AND TRAILER RIGIDITY EFFECTS

As noted above, the rigidity of the tractor and trailer are important [14, 23, 29, 30] in correctly evaluating the vehicle's response to both static and dynamic maneuvers. This is especially true of transient maneuvers [14] which require additional stability metrics for evaluation [46, 49]. However, the true rigidity of the tractor or trailer is difficult to determine as it is a combination of multiple compliances, some of which are difficult to measure.

Suspension Rigidity

The most obvious source of compliance in the tractor or trailer is the suspension. After all, the suspension's purpose is to isolate the sprung mass from the large transient input forces from the road. But, as noted above, the suspension does reduce roll stability by permitting the sprung mass to rotate about the axle. Furthermore the distribution of axle compliance is not uniform on the vehicle with the axle suspension stiffness generally increasing going from the steer axle to the trailer axles [13, 14, 30] and thus for a given lateral acceleration, the front of the vehicle will tend to roll further about its axle [21, 30].

Chassis Lateral Compliance

As the tractor is significantly shorter than the trailer the effect of lateral compliance of the tractor is not as great as that of the trailer. While the primary stability issue with the trailer is the relatively high cg location of the loaded system, low trailer stiffness may allow the trailer to shift laterally during a maneuver which has the effect of reducing track width. Track width reductions further reduce the trailer's restoring moment potential as the load cg moves toward the outboard tire. Hence the lack of rigidity of the trailer can reduce the rollover threshold of the trailer by changing the effective load placement.

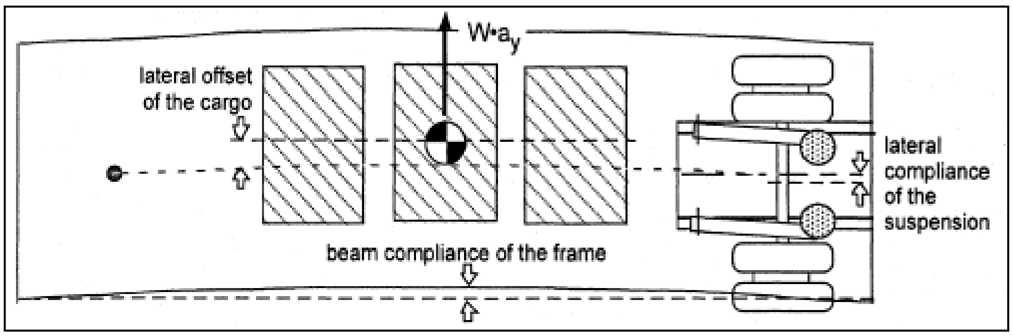

Figure 187. Illustration. Lateral Trailer Flexibility [15].

Chassis Torsional Compliance

The compliance of the tractor or trailer can also come into play in the torsional sense. The torsional compliance of the tractor means that as lateral loads and roll moments are applied to the tractor, the frame twists like a ladder. The frame twist then acts to rotate the axles attached to the frame changing the load distribution of the vehicle. This twisting effect must be accounted for to get accurate estimates of the wheel loads of the tractor [21].

As noted above, if all wheels remain on the ground, the torsional rigidity of the trailer primarily acts as a torsional compliance between the trailer cg and the kingpin or rear axles. This essentially results in a slight movement of the cg location as the trailer twists about the rear axles and the kingpin. But when one set of tires lifts off the ground, the vehicle cannot transfer any more roll moment through that axle [13, 14, 15, 30], then the lateral acceleration induced roll moment must be transferred to the remaining wheels on the ground via trailer twist. As the trailer axles usually lift first [13, 14, 15, 30], this means the trailer continues to roll with only the drive axles resisting the roll via the torsional rigidity of the trailer.

or a static or a steady-state test, the rollover moment increases slowly. Once an axle set lifts, the additional roll moment is carried by the remaining axles with a small amount of twist of the trailer since the carried roll moment is relatively small. If the lateral acceleration increases slowly, the tractor axles will lift resulting in a rollover before the relative twist of the trailer is too large. However, this is may not be true for transient maneuvers or cases where the lateral acceleration increases quickly. Here there may be significant roll moment difference between the axles. Either case generally results in a progressive rollover starting at the rear of the trailer with the difference being the relative angle between the trailer and drive axles.

It is for this reason that "Dynamic rollover threshold is the worst case measure of roll instability" [14].

Torsional Rigidity Values

It is interesting to note that while many sources list the torsional rigidity of both the tractor and semi-trailer as important [14, 23, 29, 30] few attempted to provide an estimate of the torsional rigidity [20, 30]. Sampson and Cebon [30] provided an estimated torsional rigidity of 3,000,000 N-m/rad or approximately 40,000 lb-ft/deg for a water trailer while Winkler, Bogard, and Karamihas [20] estimate a dry van trailer's torsional rigidity at 48,000 lb-ft/deg. Oddly, for a parameter that is considered important, there is essentially no information on typical rigidity ranges of differing tractors or trailers. Furthermore, there does not seem to be even a defined testing method for evaluating the rigidity of a tractor or trailer.

As the trailer rigidity is noted as being important, this project included both the development and execution of a trailer torsional test as well as the incorporation of the results in the modeling of the system. It is hoped that future work can focus on additional trailer combinations to aid in the development of a better understanding of trailer rigidities and the implications of the relative rigidities.

Physical Vehicle Rollover Threshold Evaluation

There are three main methods for evaluating a tractor semi-trailer for rollover potential: tilt testing, field testing, and modeling [8]. Each method has benefits and limitations associated with the process. Tilt testing is the simplest to execute and possibly the cheapest, but it requires a physical vehicle, a special test rig, and it is a static measurement of a dynamic vehicle. Field testing is the "true" answer where the full scale dynamic response of the vehicle is observed, but it is expensive, sensitive to noise, and also requires a physical vehicle. Modeling is the most flexible method, but the solution accuracy is very dependant on the accuracy of the vehicle model and highly accurate vehicle models are difficult to develop. As these three methods are so commonly used, they are detailed below. For reference, this study used field testing and modeling techniques for the evaluation of the tractor semi-trailer stability.

Tilt Table Testing

A tilt table test is, as the name implies, a test where the vehicle is placed on a table and tilted over until it "rolls over". In reality, the vehicle is constrained so as to not roll over completely. Probably the best known test tilt test facility is the one at the University of Michigan's Transportation Research Institute (UMTRI) [13, 14, 15].

Figure 188. Photograph. UMTRI Tilt Table [13].

- ***Basic Concept***

The tilt table works by using gravity to simulate lateral acceleration. As the vehicle is tilted over, part of the acceleration generated by gravity is proportioned such that it acts in the lateral direction of the trailer. As a result, a slight reduction in "vertical" load is seen as the "lateral" load is applied. "For moderate angles of tilt, the component of gravity perpendicular to the table remains sufficiently near unity so that accurate representations of 'vertical' tire and suspension loadings are maintained" [13]. For instance, a tilt angle simulating 0.3 g of lateral acceleration reduces the vertical acceleration to 0.96 g [13]. The effective lateral acceleration applied to the vehicle is then:

$$Ay = \tan(\Phi) = (W * \sin(\Phi)) / (W * \cos(\Phi)).$$

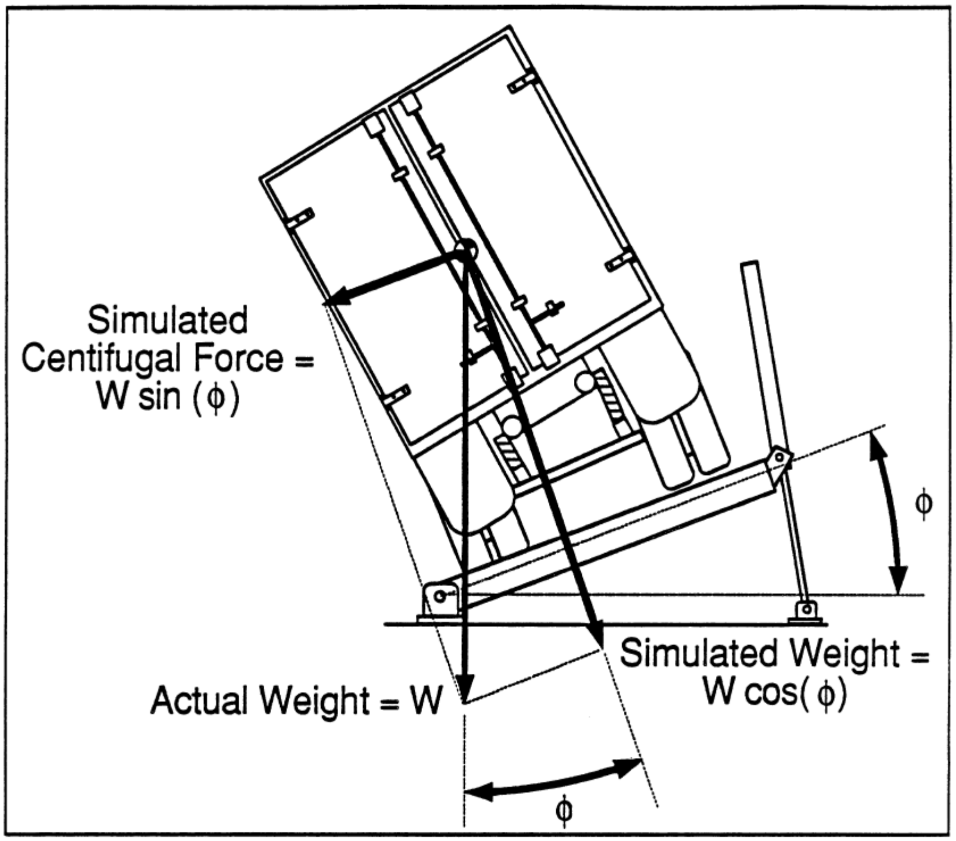

Figure 189. Illustration. UMTRI Tilt Table Diagram [13].

- ***Usage and Results of Tilt Table Analysis***

The advantages of the tilt table are that the fidelity of the test is much better than a typical field test [13] and the test is significantly shorter as no vehicle instrumentation is needed. Of course, as the vehicle is tilted, the normal load to the suspension drops and the suspension unloads slightly, artificially raising the cg height. However, the error generated by this effect is typically small. The test is generally better at evaluating large lateral acceleration levels such as rollover events as the vehicle's hysteretic influences are not as great (the hysteresis level is relatively small compared to the reaction forces [13]). The higher lateral load accuracy is due to the fact that though lower lateral acceleration levels keep the "vertical" suspension loads closer to the real usage conditions, the hysteretic effects make low acceleration measurement less accurate [13]. A final note of importance is that the test can only evaluate one articulation, i.e. 0°.

One extremely valuable result of the tilt test is that the various suspensions can be evaluated individually or as a system. A typical test will produce a roll vs. lateral acceleration curve similar to the one below. Note that the vehicle does not have a zero roll for zero lateral acceleration. This is a result of Coulomb friction from the repetitive raising and lowering of the vehicle. [13]

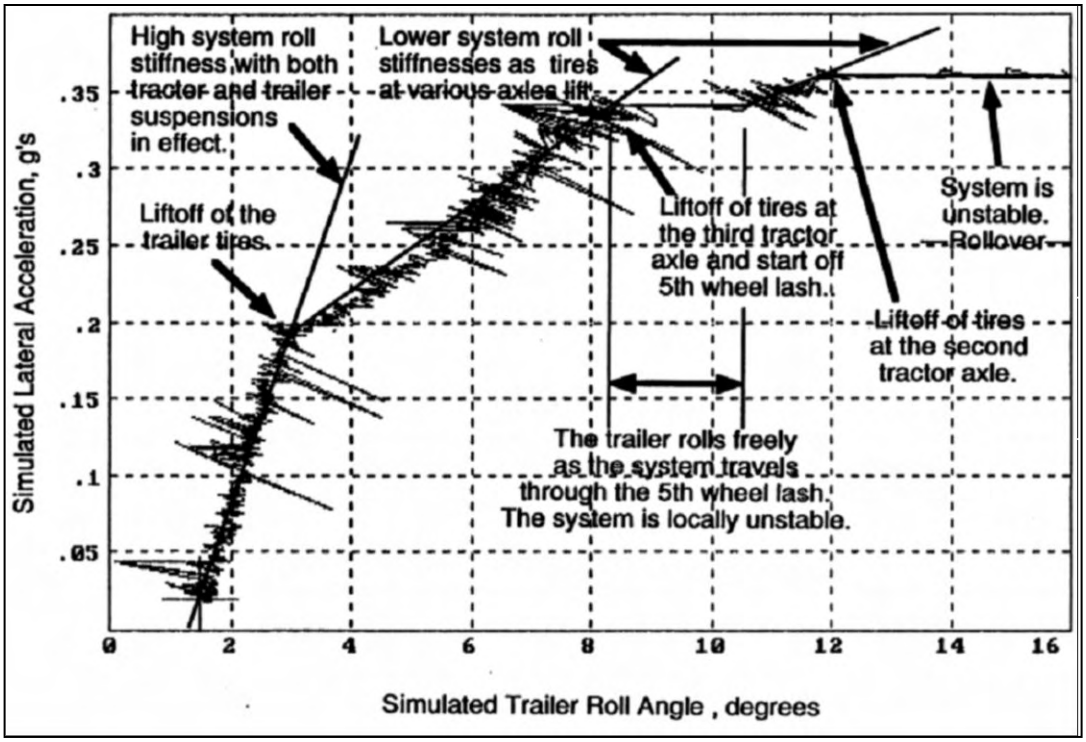

Figure 190. Graph. Roll vs. Lateral Acceleration [13].

- ***Vehicle Evaluation***

Here the various suspension effects can be observed as first the trailer tires lift off, then the drive axles lift off. After the trailer tires lift, the trailer roll is being resisted by the tractor until the tractor starts to roll. When this happens, the trailer to tractor coupling, the fifth wheel, travels through its clearance (lash) and the trailer rolls and additional few degrees with essentially no restoring moment increase.

At low restoration moment levels, tractor to trailer torsional moment can be passed by the lateral shift of the center of compressive load on the surface of the fifth wheel. When the compressive load point moves off the surface of the fifth wheel, the moment cannot be transmitted in compression and the trailer roll moment changes from compression to tension and is passed from the trailer body down to the tractor through tension in the fifth wheel kingpin [13]. Because of clearances designed into the coupling mechanism, the kingpin must move upwards to be placed in tension, and thus, a relative motion occurs across the coupling [13].

This axle lift order (trailer, drive axles, and then front axles) is typical as the trailer generally has the highest CG, followed by the drive axles, and then the steer axle. The front axle is usually so compliant that it cannot generate a high enough restoring moment to prevent rollover [13, 14, 30]. Collecting the suspension reactions and the vehicle roll moment shows the combining effect of the suspensions (below). Note that the roll moment has a negative slope by the time the front axle lifts off indicating that the vehicle "rolled over" when the drive axles left the ground.

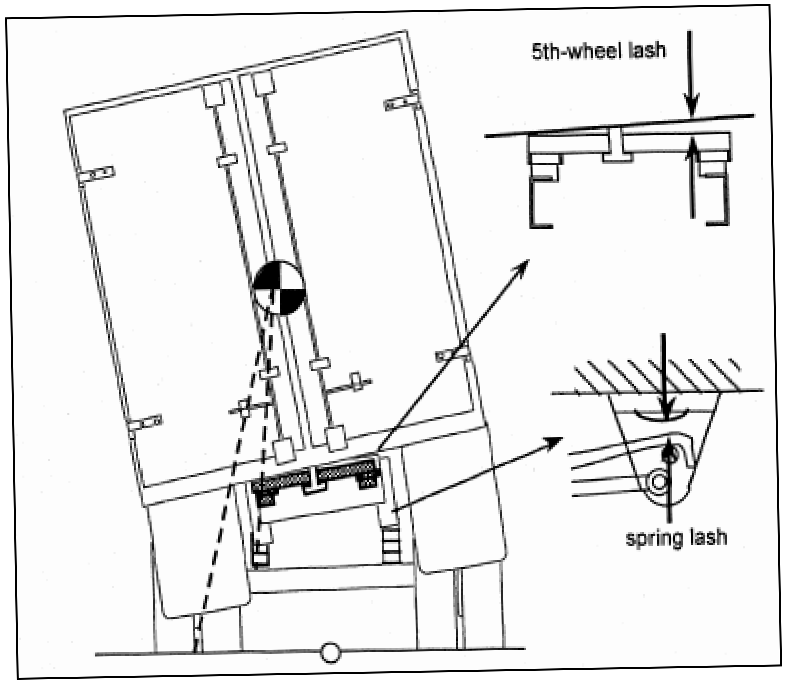

Figure 191. Illustration. Trailer Lash [14].

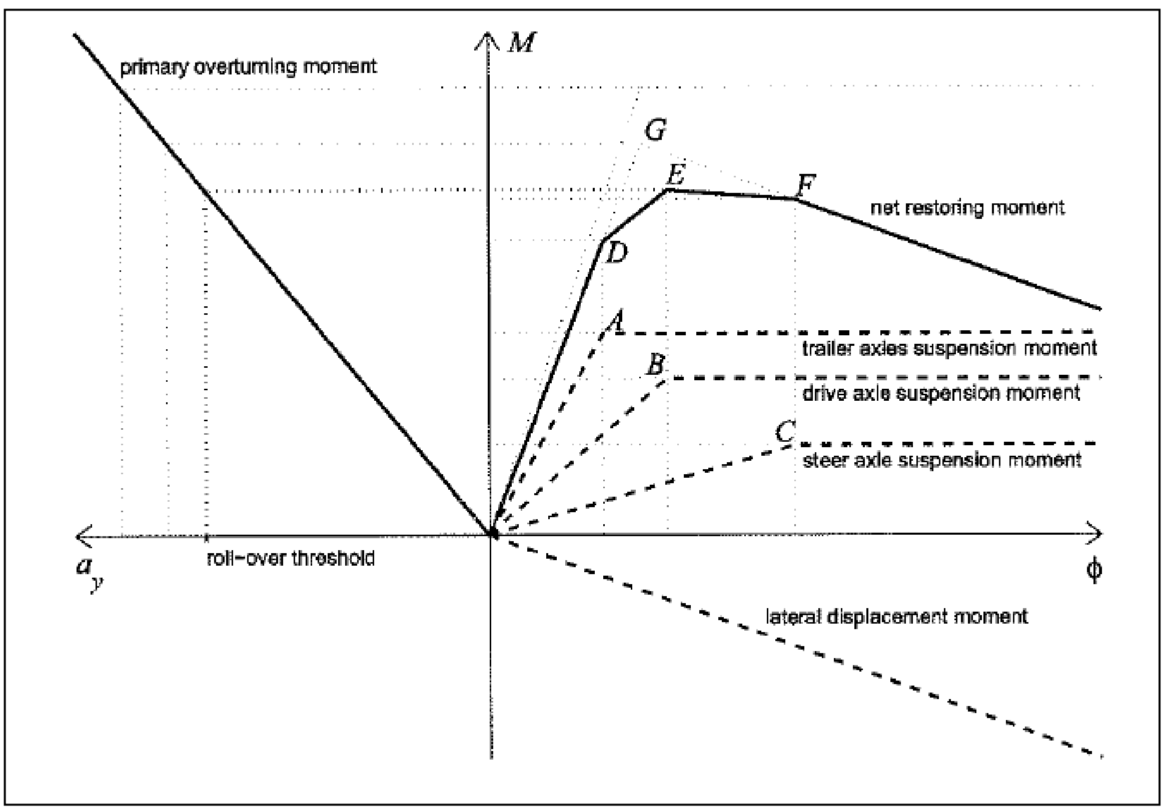

Figure 192. Diagram. Net Roll Moment from Tilt Table Test [30].

The one major fact to remember here is that the roll angle on the bottom axis is for the ground plane and is not the angle of any given location on the vehicle. There are actually many vehicle roll angles as the truck is torsionally flexible and the overturning moments vary by axle. Looking at the tilt table picture above, one can see that the rear of the trailer is tilting further than the tractor. Most of this difference in roll angle between the tractor and the trailer axle locations is the fifth wheel lash. But some of the difference is due to the torsional compliance of the trailer. This shows that while the vehicle rigidity is not as significant here, it does have a significant impact in vehicle roll response.

Field Testing-Test Cost and Description

Field testing is not as frequently preformed relative to the other methods as the costs can be quite high. Testing involves the selection of a tractor and trailer(s) along with the instrumentation and safety equipment (out riggers, load restraints, etc.), setting up the vehicle, and executing the test. But field testing produces a true vehicle response characteristic, and is especially valuable for evaluation of transient vehicle modes.

Field testing typically involves two distinctly different tests. The first is a steady-state test where the vehicle is driven in a large circle. This test provides similar results to the tilt test where the vehicle is evaluated at increasing levels of lateral acceleration. The obvious differences being that the lateral acceleration is true acceleration (not a fraction of gravity) and that the articulation angle is un-constrained. The preferred method to perform this test is with a constant velocity and varying radius [48]. However, the test can be run with constant radius and increasing speed or constant steer angle and increasing speed [51].

The second test is a transient test where the vehicle's response to changing inputs over time is evaluated. There are many transient tests including pulse steer, step steer, lane change, swept sine, and more. Both the Society of Automotive Engineers and the International Standards Organization have published test procedures for transient response evaluation [46, 47, 49]. The objectives are to measure the peak responses or over shoot responses of yaw rate, lateral acceleration, roll angle, etc., as well as the time lag for trailer response, and the peak yaw rate and lateral acceleration gains for each vehicle unit. Additionally, the dynamic load transfer is measured to evaluate the sensitivity to rollover (transfer of all load to outer wheels indicates rollover) [10, 11, 47].

Generally, it is desired that the trailer not amplify any of the steering inputs. This means that the trailer has the same roll angle (or less) as the lead unit, the trailer's lateral acceleration is the same as the lead unit's, and the yaw rate of the trailer does not exceed the lead unit's yaw rate. However, because of the physics of the vehicle's response, some amplification is expected. While some vehicles are more sensitive to rear amplification and need to maintain very low rear amplification levels due to high cg locations or extreme rearward cg locations, the general recommended limit for rearward amplification is about 2.2 [10].
The second main dynamic stability metric is the dynamic load transfer ratio. The amount of dynamic load transfer from the inner wheel to the outer wheel depends on the roll angle of the vehicle, the load cg location, and many other factors. Defining the transfer of the entire inside wheel load from the inside wheel to the outside wheel as 100%, the suggested dynamic load transfer limit for stability is 60% [10]. Of course, the amount of transfer is also related to the

maneuver so what is essentially being evaluated is the severity of the maneuver that the tractor semi-trailer can accomplish.

Modeling of Tractor Semi-trailer Rollover Propensity

Often it is more convenient to model a tractor and trailer combination rather than conduct actual tests. This is especially true if the goal is to evaluate generalized vehicle parameters or other non-vehicle specific variables such as component stiffness or control inputs. For these cases, theoretical model may be employed with acceptable accuracy levels.

Even for cases where there is a specific vehicle of interest, there may be other factors such as test costs that dictate the use of modeling for the vehicle analysis. In these cases, full vehicle models describing the actual vehicle can be developed. Full vehicle models generally require more time to complete as there are more parameters to deal with and the analysis result scope (output parameter of interest) is generally larger.

The result of the differing modeling needs is the use of differing modeling approaches. The generalized modeling needs are usually met with theoretical models and the vehicle specific models with multi-body (component) or kinematics models.

- ***Theoretical Models***

  Theoretical models used for heavy truck analysis are generally either degree of freedom based or classical vehicle dynamics based models. Degree of freedom models are derived from component degrees of freedom and the equations of motion needed to describe the motion of the degrees of freedom. Vehicle dynamics based models use the vehicle parameters such as track width, wheel base, cg location, etc. to define the response of a vehicle to inputs.

- ***Degree of Freedom Models***

  Degree of freedom models are useful for evaluation of generalized vehicle parameters and vehicle operational controls in a limited environment (i.e. most parameters are fixed). For a tractor semi-trailer analytical model to match reality, the number of degrees of freedom in the model would have to be very large. The most basic model would have to have five degrees of freedom for each sprung mass and two degrees of freedom for each un-sprung mass [25]. So a minimal two drive axle tractor and two axle semi-trailer model would have to have at least 20 degrees of freedom to just to describe gross vehicle motion for a rigid chassis.

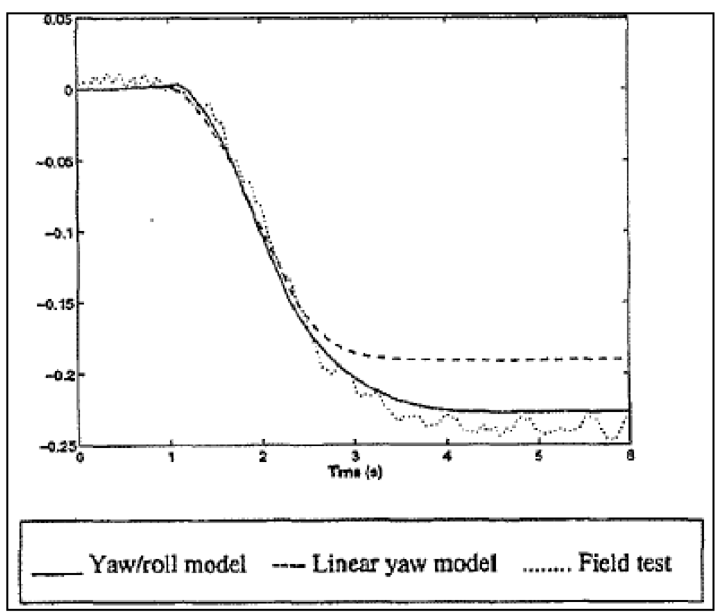

Figure 193. Graph. Simple Degree of Freedom Model Accuracy [25].

Obviously, building a highly accurate analytical model with compliances and non-linear components would be difficult. But analytical vehicle models do not have to be overly complicated to be useful for evaluation of a particular region of interest. Region focused models need only to be accurate in the region of interest. These types of models are generally used for fundamental system analysis, input response sensitivity, or control evaluations such as the evaluation of roll threshold for tipped trucks [24] or rigid body yaw and roll estimation for shifting cg position [25].

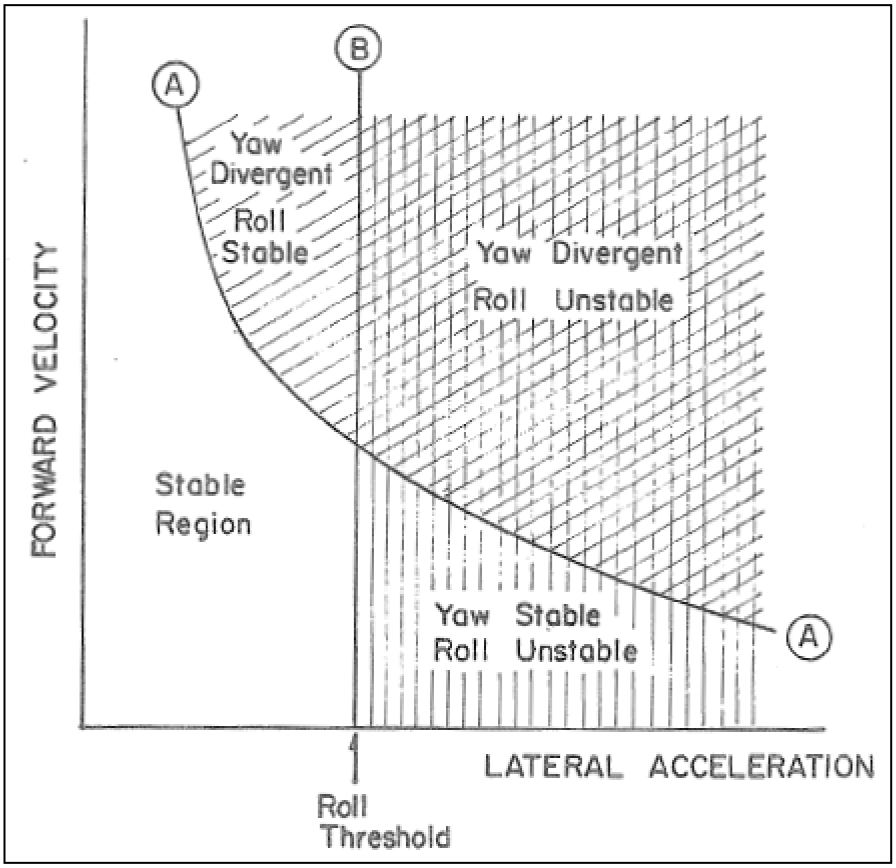

Figure 194. Graph. Stability Regions [23].

While a highly accurate model is not required for accurate results, the relevant model parameters do need to be accurate and the model's approximation of the vehicle motion sufficiently close to reality. For example, the yaw / roll model from Tong et. all [25] works well, but the linearized model does not.

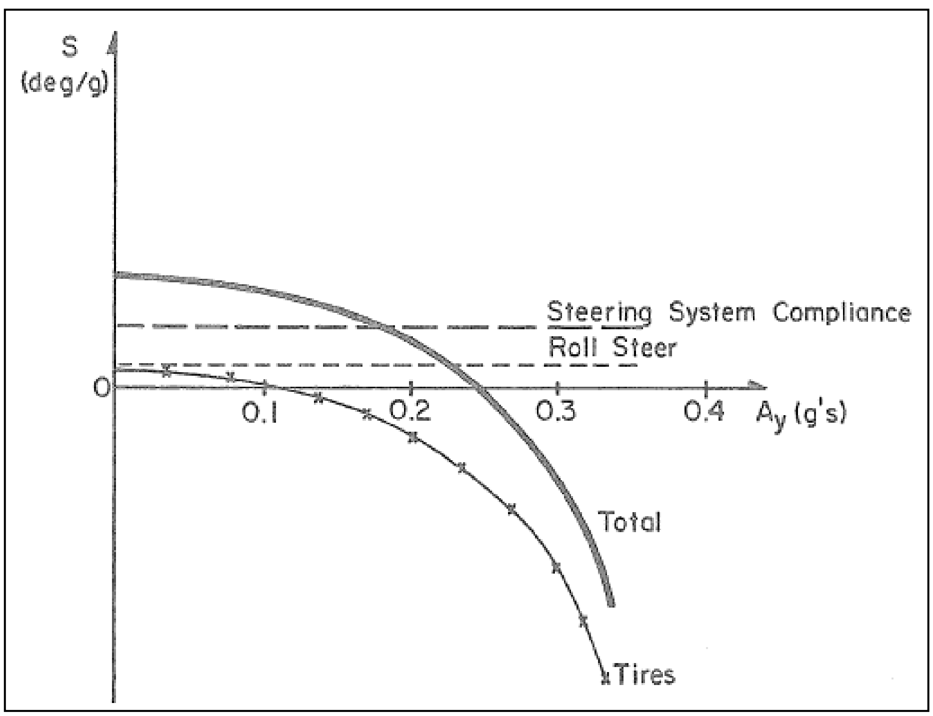

Figure 195. Graph. Tractor Semi-Trailer Understeer Gradient [23].

- ***Classical Models***

Classical vehicle dynamics models like that used by MacAdam [23] are generally used to equate basic vehicle responses to fundamental vehicle responses. For example, the sensitivity of a tractor semi-trailer to yaw instability or rollover is fundamentally related to lateral acceleration and forward velocity.

A major advantage of this approach is that generic grading criteria can be developed to gage how well a vehicle will perform under a variety of basic conditions. The static rollover threshold is an example of this type of criteria as is the understeer gradient of the vehicle as listed above.

- ***Full Vehicle Models***

While analytical models are good for investigation of vehicle response theories and design parameter impacts, they are not generally well suited for evaluation of an actual (or proposed) vehicle's amplitude responses to differing inputs. Specific vehicle responses can be obtained through testing or the vehicle can be modeled and evaluated using simulations. Reasons for modeling rather than testing a vehicle's response include, but are not limited to, not having a real vehicle (development or availability), test expense, or concerns about damaging the vehicle.

Obviously, if the objective is to estimate how a specific vehicle will respond to steering inputs, the vehicle must be modeled with significant care. In this endeavor, the most important step in developing a vehicle model is the determination of what parameters need to be collected and how to measure the parameters. Critical parameters include, but are not limited to, masses and cg locations, suspension chassis compliance, component lash, and tire properties.

Of these model parameters, one merits particular interests. Most of the historical tractor semi-trailer case studies used simple lumped masses with torsional springs and dampers [17, 21, 22] to model chassis torsional stiffness. However, as noted by Aurell [21], this approach is not ideal as the actual chassis does not rotate about a pivot point but rather the frame twists as a ladder along its length. The result of the simple lumped mass approach is incorrect load transfer estimations for the axles.

  - *Model Classifications*

  There are two major types of full vehicle response models: Multi-body and Kinematics based. Multi-body models have the benefit of not requiring an actual vehicle as the model is assembled from component properties. However, the multi-body model can be quite difficult to assembly accurately. The kinematics model approach requires a real vehicle to test but is simpler to model as the force / response curves are the inputs to the model.

  - *Multi-body Model*

  Multi-body (component) based models such as ADAMS are assembly type models were each component is measured, modeled, and assembled virtually. The assembly is then exercised virtually to evaluate responses [37]. The advantages are that the vehicle does not have to exist; only the components or the component properties need to be known. The disadvantage is that it can be difficult to accurately characterize and combine all of the components. "The difficulty in creating the model comes from determining the appropriate level of detail that will accurately represent the dynamics of interest [31]".

Once the model is developed, it can be used to simulate the vehicle's response to inputs. However, this often requires tuning to keep the components in proper alignment as geometric and reaction constraints are broken. The resulting vehicle motion is the sum of the part reactions which provides flexibility to tune the model. However, it should be noted that the error I the model is the stacking of the errors in each part's design and the interaction rules.

- *Kinematics Model*

A kinematics and compliance model is based on the measurement of a real vehicle as it is displaced and deformed under various loads. Here the individual components are not of interest, just the total vehicle's response. The advantage is that the true vehicle response from a given force, induced roll angle, or torque is directly measured. The disadvantages are that the data collection requires a real vehicle and the data is quasi-static in nature. This study used kinematics and compliance data as we had an actual vehicle, a testing facility was available, the kinematics and compliance data is easier to use to develop a model, and we did not have access to the engineering specifications for the vehicle needed for a component based model.

The first step in creating a kinematics model is to evaluate the vehicle kinematiclly. The next phase of creating a kinematics model involves developing the mathematical equations describing the vehicle's reaction forces and vehicle responses. The model needs to be able to determine reaction forces and moments generated by vehicle inputs, determine velocity and position by integration of accelerations, and other similar system dynamics parameters. The development of these types of models can be quite intensive in terms of time and resources.

To save time and resources, it was decided that this project would use a commercial application called TruckSim developed by Mechanical Simulation to handle the simulation mathematics. TruckSim has all of the dynamic equations developed and the data management formats defined. Al that is needed here is to import the needed truck kinematics data, the appropriate tire data, and the test conditions.

Stability Controls

As electronics and mechatronics have become less expensive, the use of active stability control systems in tractor semi-trailers has increased. As with passenger vehicles, the goal of stability control systems is to reduce the number of accidents by keeping or returning a vehicle to a stable operational environment. While passenger car stability is primarily concerned with yaw control and path deviation, heavy truck stability systems must handle these same issues plus roll stability issues and multi-unit articulation issues. As a result, the complexity of a stability system in tractor semi-trailers is considerably higher than their passenger car equivalents.

Generally, stability control systems can be divided into two classes. The first uses active suspensions or other mechatronics devices to alter the vehicle's dynamic characteristics. The second class uses the braking system to induce additional dynamic forces to counter the externally applied dynamic forces generated by the maneuver the vehicle is attempting. Brake based systems are usually an advancement of anti-lock brake systems where the wheels are individually controlled. Common names for these types of systems include ESC (Electronic Stability Control), RSC (Roll Control System), and EBS (Electronic Braking System) [36]. The vehicle used in this study was equipped with a stability control system of this type.

Active Suspensions

Active suspensions are not in general use but are currently being studied [28, 29, 30]. Active suspensions usually are either systems with active controls of existing suspension components or systems with additional mechanical suspension components such as active roll bars [28, 35] or active fifth wheels [29].

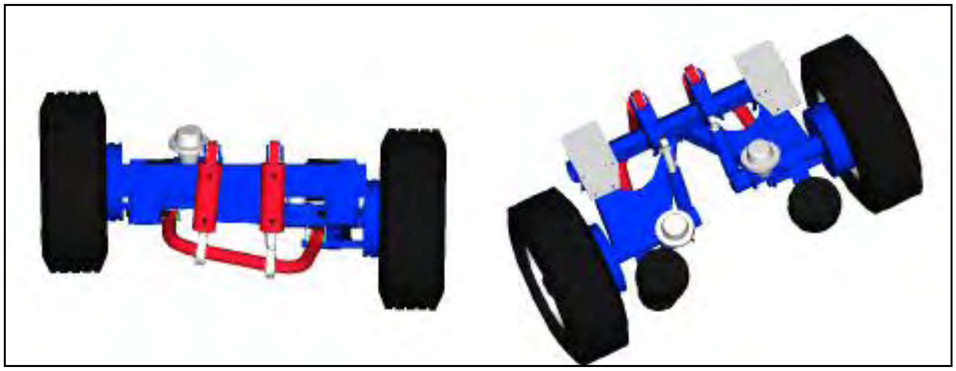

Figure 196. Illustration. Active Roll Suspension [28].

- ***Effectiveness***

  Systems with active control of existing suspension components, air bags for example, are not typically efficient as the amount of energy (usually compressed air) needed to adjust the suspension is too great to produce on a truck. Systems with auxiliary driven suspension components are feasible as they can be controlled reliably and efficiently. The issues with driven suspension components is in the control methodology [30] as the suspension is always acting in a time delayed manner after the vehicle initiates a maneuver. It is estimated that these systems could improve rollover potential between 30% and 60% [30].

- ***Applications***

  Active suspensions are being investigated for stability limited vehicles as these vehicle offer the best opportunity for demonstrating improvements [30]. Generally the idea is to force the suspension to deform generating a moment counter to the overturning moment the vehicle is experiencing. A large part of this effect is generated by simply pushing the suspension through the vehicle’s lash as this does not require much power and increases stability by keeping the vehicle’s cg from shifting as much.

Electronic Stability Control Systems (ESC)

Electronic stability systems are making inroads into the trucking fleets. This is primarily due to the relatively low cost of the system compared to the potential cost savings from a vehicle accident. As the systems work through the brake systems, most of the needed hardware is already on the tractor as part of the anti-lock brake system (ABS). A few extra sensors (yaw, lateral acceleration, roll) and some software changes are all that is needed to turn the ABS system into an ESC system.

- ***Yaw and Roll Control***

ESC systems provide overturning stability by inducing yaw moments into the vehicle to counter the overturning moments. This is done by selectively braking the wheels on the tractor and/or trailer to introduce the desired yaw moment.

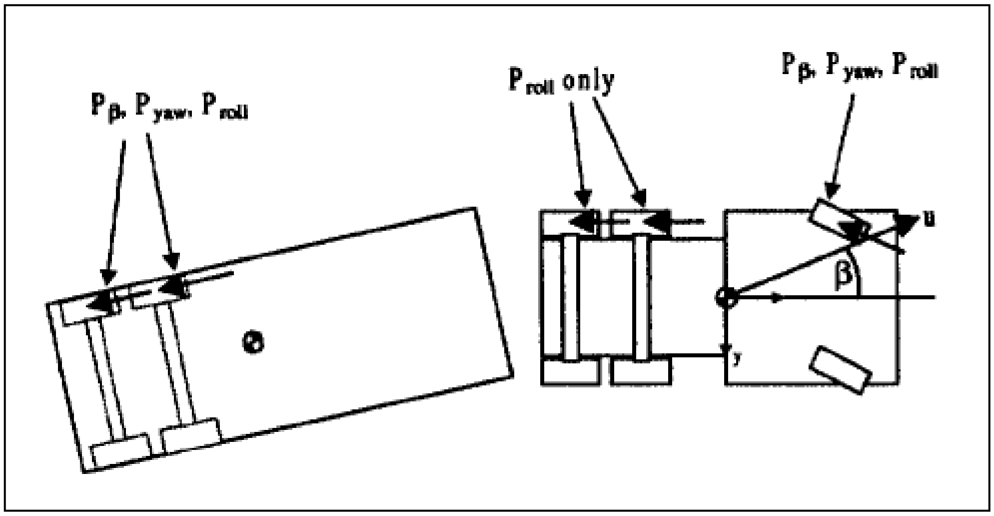

Figure 197. Illustration. ESC Braking [33].

Unlike active suspensions which are primarily designed to reduce rollovers, ESC systems can also help reduce jackknife (yaw divergence) and path deviation incidents by automatically slowing the vehicle as instability is detected. Given that yaw divergence accident rates increase with lightly loaded vehicles [3], the ability to reduce both loaded condition stability conditions is a clear advantage for ESC type stability systems.

- ***Combination Vehicle Issues***

The main difficulty with ESC type stability systems is that semi-trailers are rarely tuned to the tractor in real world applications. As a result, the ESC system may or may not (usually not) have input from the trailer as to what stability state it is in [40]. Similarly, the system may or may not (usually not) be able to selectively control the trailer wheels. However, even without the trailer's participation, the current state of the art systems can infer the trailer response and apply restoring moments at the tractor to stabilize the entire vehicle.

- ***Effectiveness***

While ESC systems are becoming more and more prevalent, their real world effectiveness has not been well documented. But, given that many US fleets, which are self insured, are buying the systems, it is clear that the prevailing opinion is that the systems prevent enough accidents to justify the cost of the system. Lastly, it is obvious that such systems cannot prevent tripped rollovers or vehicle to vehicle accidents so ESC systems will never be a panacea for all truck stability issues.

Since ESC is quickly gaining ground in the market and there is not a lot of field data on the effectiveness of the systems, a major part of this study was designed to provide some insight into how well the systems perform in various maneuvers. While this study is by no means comprehensive, it does provide the opportunity to gage the relative merits of stability control as compared to conventional, and well understood, stability influencing parameters.

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APPENDIX B: TEST PLAN FOR TEST TRACK TESTING

HEAVY VEHICLE SAFETY RESEARCH CENTER
Heavy Truck Rollover Characterization – Phase-A
Test Plan for Test Track Testing – Attempt-4
Prepared by:
Oak Ridge National Laboratory (ORNL)
Michelin Americas Research & Development Corporation (MARC)
National Transportation Research Center, Inc. (NTRCI)

BACKGROUND

The National Transportation Research Center Inc. (NTRCI) in partnership with the Oak Ridge National Laboratory (ORNL), Michelin Americas Research and Development Corporation (MARC), Western Michigan University (WMU), Battelle, and Clemson University, will conduct controlled combination vehicle (tractor with flatbed trailer) rollover testing in support of the Federal Highway Administration's (FHWA's) goal of continued improvement of highway safety.

Understanding the interactions of vehicle load, tires, suspensions, vehicle types, vehicle stiffness (tractor and trailer), and roadway surfaces/tire interface on truck rollover events can contribute significantly to improving heavy truck safety. Such understanding can be applied to support the design and evaluation of new technologies such as wider axles, new generation single wide-based tires (NGSWBTs), adaptive suspension systems, rollover warning systems, etc. It can also contribute to improving roadway design to minimize the potential for truck rollover stemming from vehicle-highway interactions, and can contribute to more effective regulation aimed at reducing truck rollovers. Additionally, such understanding is essential for development of a next generation heavy vehicle dynamics model that would be a valuable tool for evaluating new product innovation, and for the assessment of the interactions associated with complex vehicle dynamics. Such a model would provide the capabilities of a virtual test ground; i.e., computer-based experiments could be conducted in lieu of expensive test-track or field operational tests. Such a model could include a detailed and integrated braking system model that addresses not only the brakes, but also its suspension, axles, drive-train, tires, tire-road interface, etc. Additionally, such a model could include modules reflecting driver performance and behavior. It should be noted that at the current time, the modeling efforts in this project are not addressing coupled inputs (e.g., steering and braking) or human behavior, but such elements should be addressed in a "next generation" model. Although there have been a number of heavy truck rollover tests conducted in the past, there are almost no existing publicly available databases on heavy truck rollover data of sufficient granularity to support development of a robust vehicle dynamics model.

Understanding the dynamics and characteristics of heavy truck rollovers is complex, and cannot be achieved thoroughly without the conduct of vehicle testing, the collection of vehicle dynamics data and information on truck rollover performance, etc. In addition to being complex, limited resources for truck rollover research necessitates that it be conducted with a longer-term perspective.

PURPOSE

The intent of this document is to provide guidance for conducting a heavy truck rollover test and to describe the related truck configuration, data acquisition configuration, equipment, and test events.

The purpose of the rollover testing, i.e., what we want to do or achieve, is to:

7) Gain unique knowledge of suspension roll stability and roll characteristics. Of specific interest are the tractor's drive suspension and the trailer suspension.
8) Understand the effects that suspension roll characteristics have on overall vehicle roll stability.
9) Understand the roll stability and handling differences between dual tires and NGSWBTs.
10) Understand and verify the benefits of using Electronic Stability Control (ESC).
11) Acquire actual rollover data that can be used for correlation to modeling and the simulation of dynamic roll events in modeling.
12) Determine (if possible) predictive indicators of roll that can be used to alert drivers and prevent rollover accidents.
13) Gain insight of the effects of increased torsional stiffness in an integrated suspension.
14) Gain insight of the effects of increased torsional stiffness in tractor and trailer frames.
15) Learn about the rollover propensity and characteristics of different trailer center of gravity heights.

APPROACH

This Test Plan calls for rollover testing of a class 8 over-the-road tractor with flatbed trailer at the Transportation Research Center (TRC) in East Liberty, Ohio. Prior to the initial test-track testing (i.e., November, 2007), the test vehicle was characterized. MARC conducted torsional stiffness and Kinematics and Compliance (K&C) testing on the tractor. The flatbed trailer was also sent to WMU for further characterization where a solid model of the trailer was created (using Pro-E), and an ADAMS model of the suspension was created. WMU also subjected the trailer to a torsional stiffness test without the ballast load. After the test-track testing is complete, MARC will conduct a torsional stiffness test and roll testing of the trailer. Just prior to the tests to be conducted in late-April/early-May, WMU will travel to TRC to gather torsional stiffness data on the fully-loaded trailer with the outriggers in place. If possible, and resources allow, WMU testing will be done with the current ballast as well as the new ballast configuration.

The rollover testing of the test vehicle will be accomplished with two different tire configurations (one with NGSWBTs on the tractor-flatbed, the other with standard dual tires on the tractor-flatbed). Four different test maneuvers will be performed with each tire configuration; two transient and two steady-state. The two transient tests are: 1) a step steer maneuver (SSM) and 2) a highway evasive maneuver (HEM). The steady-state tests are: 1) a constant radius turn maneuver (CRTM) and 2) a ramp steer maneuver (RSM). These four maneuvers will produce data that will give insight into the interaction of the tractor, flatbed trailer, and tires relative to rollover.

Tires used for the testing will be as follows:

- Steer Tires: 275/80R22.5 XZA3
- Standard Drive Axle Tires: 275/80R22.5 XDA Energy
- NGSWBT Drive Axle Tires: 445/50R22.5 X One XDA (used with 2.25" offset rims)
- Standard Trailer Tires: 275/80R22.5 XTA Energy
- NGSWBT Trailer Tires: 445/50R22.5 X One XTA (used with 2.25" offset rims)

Tire configuration 1 – NGSWBTs mounted on the tractor drive axles and trailer. The steer axle, drive axle, and trailer axle tires for tire configuration 1 will be: XZA3, X One XDA, X One XTA, respectively.

Tire configuration 2 – Conventional dual tires mounted on the tractor drive axles and trailer. The steer axle, drive axle, and trailer axle tires for tire configuration 2 will be: XZA3, XDA Energy, XTA Energy, respectively.

While this type of testing with this tire configuration has been done by other research teams, it will be conducted in this project in order to generate accurate baseline data that can be used for comparison with tire configuration 1.

The tire configuration and test maneuvers to be conducted are listed in table 40.

INITIAL TESTING EFFORTS

The full regimen of test maneuvers with the specified vehicle tire/rim configurations called out in table 40 shall be conducted using one tractor and one standard suspension flatbed trailer. After completing rollover testing at TRC, the flatbed trailer shall be delivered to MARC where it will undergo torsional stiffness testing.

The tractor-trailer was delivered to TRC for testing on October 4, 2007. Setup, including installation of the test instrumentation on the tractor-flatbed required three days. Also, as part of the setup, the tractor-flatbed was fitted with outriggers and anti-jackknifing chains. A detailed schedule of the tractor-trailer setup and rollover testing was provided by TRC and will be incorporated into the final version of this document when available.

Table 40. Rollover Test Matrix.

Test Series	Tire Configuration	Tractor Drive and Trailer Tires	Test Maneuver
1	1	NGSWBTs2	CRTM
2	1	NGSWBTs2	RSM1
3	1	NGSWBTs2	SSM
4	1	NGSWBTs2	HEM
5	2	Standard Duals3	CRTM
6	2	Standard Duals3	RSM1
7	2	Standard Duals3	SSM
8	2	Standard Duals3	HEM

1If wheel lift is achieved during CRTM, the RSM will not be performed.
2Rims used with the NGSWBTs will be 14X22.5 with a 2.25 in offset.
3Rims used with the Standard Duals will be 8.25X22.5 with a standard offset.

An original Test Plan was developed in September, 2007, and initial testing was attempted in November, 2007. Because of the unexpected behavior of the tractor-flatbed combination, the testing in November was halted. The unexpected behavior involved the sudden and dramatic lift-off and subsequent slamming-down of the drive axle during a transient maneuver. The research team felt that moving the outriggers forward, toward the tractor, would provide protection for the tractor. The Test Plan was modified, the outriggers were move forward, and the testing was again attempted in December, 2007. For the second attempt, the sudden and dramatic drive axle lift-off during a transient maneuver was again experienced, and the outriggers were successful in protecting the tractor. However, the trailer axles also subsequently lifted-off. With the outriggers mounted in a more forward position, the trailer was left unprotected against rollover. As a result, testing was terminated. The research team felt that mounting two sets of outriggers on the trailer; one to protect the tractor and one to protect the flatbed would allow a third attempt at testing to be successful.

The Test Plan was again modified, MARC shipped its outriggers to TRC, the MARC outriggers were mounted, and testing was again attempted in January, 2008. For the third attempt at testing, the tractor-flatbed with two sets of outriggers attempted a transient maneuver at ever increasing speeds in order to achieve wheel lift-off. When no wheel lift-off was experienced at a relatively high speed, TRC terminated the test. The reason cited was that with the high speeds, if a sudden and dramatic lift-off was achieved, the outriggers may not be able to protect the tractor-flatbed. The research team agreed that the only way to achieve lift-off at a slower speed without a tractor-trailer re-design was to increase the height of the payload center-of-gravity (CG). The current payload configuration involved ballast with a low-CG distributed homogeneously along the spine of the flatbed. The team also felt that the configuration of the ballast might add stiffness to the flatbed, and as a result, it was recommended that the ballast would be "lumped" over the drive and trailer axles. This Test Plan will address the fourth attempt at testing utilizing two load racks; one mounted over the drive axle, and one mounted over the trailer axle that will allow various CG heights to be achieved. With such flexibility in the manipulation of the payload, the fourth testing attempt provides high assurance that good experimental results will be achieved.

All project-related test-track testing equipment will be provided by TRC. TRC has, and shall continue to provide data throughout the testing for verification that all sensors are functioning properly (see Sections 5.1.4, 5.3 and 5.4 for details).

TEST TRACK TESTING

The fourth test-track testing effort of this project will be completed at TRC in East Liberty Ohio. NTRCI, in conjunction with ORNL and MARC, successfully performed Phase I and II truck rollover testing with a van trailer at TRC's facility in 2004. Figure 198 provides a layout of TRC's test-track facility.

WORK INSTRUCTION STEPS

Vehicle Configurations and Specialized Equipment

1) Volvo Model: VT64T830 class 8 (33,001 lb GVWR and over) heavy conventional tractor (leased by MARC). See figure 199.
2) Utility 48 ft Flatbed trailer (leased by MARC). See figure 200.
    a. Aluminum deck
    b. Intraax air suspension
3) Two sets of outriggers (mounted near the front and rear of flatbed trailer) and anti-jackknifing chains for combination vehicle. One set of outriggers will be provided by MARC, and one set of outriggers, along with the anti-jackknifing chains, will be provided by TRC.
4) Ballasting (provided by MARC).
5) Ballast Rack System (developed by TRC for NTRCI)
6) Sensors and Data Acquisition Systems (DASs) (provided by TRC).

Ballasting

The ballast will consist of cast iron blocks loaded onto the flatbed trailer. Vehicle/tire configurations will be tested in a "fully-loaded" condition (i.e., max GVWR). The trailer will be loaded with ballast to bring the GVW as close to 80,000 lbs as possible (including the TRC and MARC outriggers and anti-jackknife chains). No additional ballast shall be attached to the tractor for any part of this testing. All ballast shall be secured by chains before testing.

Ballast Rack

Based on specifications provided by the research team, TRC will design, build, and install a rack system to elevate the ballast blocks above the deck of the flatbed trailer. The rack will be designed in such a way that the combined CG of the ballast can be raised in several increments to a maximum of approximately 12 feet above the road surface when mounted on the flatbed trailer. The rack will also be designed such that the ballast can be divided into two loads; one over the drive axles of the tractor and one over the trailer axles.

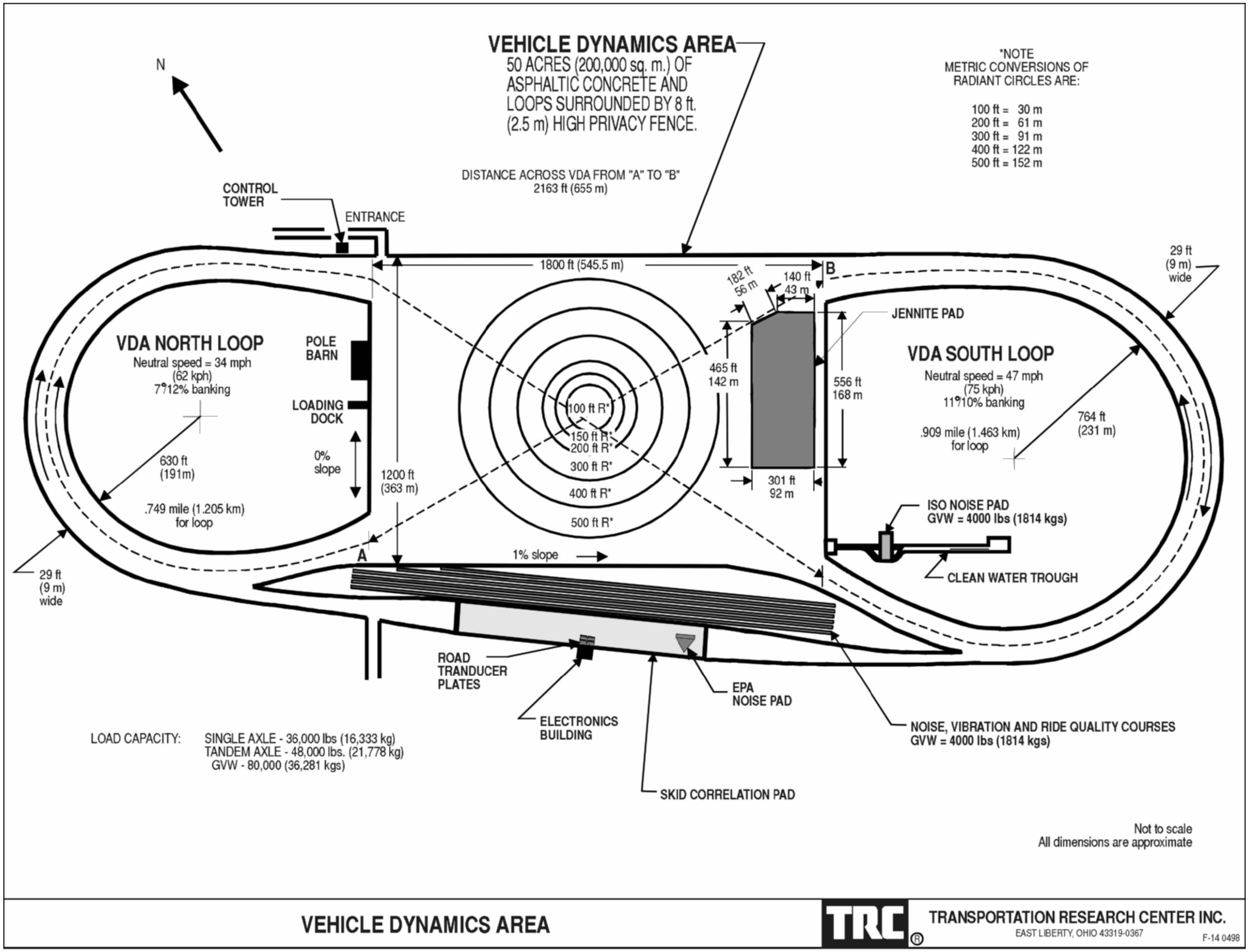

Figure 198. Diagram. TRC, Inc..

Figure 199. Photograph. Volvo VT64T830.

Figure 200. Photograph. Utility Flatbed Trailer.

Data Acquisition

All project related test-track testing sensors and DAQs shall be provided by TRC. TRC will provide ORNL and MARC staff with data after the first test-run of the day and at the end of each test maneuver set in order to verify that all sensors and related equipment are functioning properly.

The same sample frequency, 100 Hz, will be used for all data collection channels. Specific channels along with the data voltage range, equipment/sensor used, and sensor location for each data type shall be provided by TRC and will be listed in a table for reference by the research team. An example of the format for this table is presented in table 41.

Tires

The truck was delivered to TRC with NGSWBTs on the tractor and trailer. MARC will have shipped a set of mounted dual tires for the tractor and trailer to TRC prior to the initiation of the testing. MARC will also have shipped two additional NGSWBTs for the tractor, two additional NGSWBTs for the trailer, and three additional conventional steering tires to TRC as well. The test maneuvers are scheduled to be performed in order of least-aggressive to most-aggressive, relative to tire wear and damage. The test maneuvers and tire change order are displayed in table 42. See table 43 for tire sizes and model numbers.

A warm-up period will be required before each series of testing (after every significant downtime). The vehicle under test shall be operated for a period of 30-minutes prior to the start of the test. The warm-up operation is performed to allow the tire temperature and pressure to reach normal operating levels before the test maneuvers are initiated. The tire pressure shall be monitored and maintained at 7.9 bar (hot) for the steer axle and NGSWBTs, and 7.1 bar (hot) for the dual tires. Tire pressure, tire temperature and road surface temperature shall be recorded before each Test Series using an optical pyrometer.

All of the tires shipped to TRC will be used as part of the normal test plan. If additional tires are needed at TRC (e.g., in the event of a damaged or flat tire), a spare tire will need to be shipped from MARC. The current budget does not provide funding for any spares at the test-track.

Video Cameras

Due to budget and time constraints, the actual vehicle rollover testing will not be professionally video recorded. The research team will video-record the test maneuvers as the opportunity allows. After the testing is completed, a professional video-recording will be made for public relations purposes (not data analysis). This video will highlight the tractor, trailer and will repeat some of the test maneuvers.

Table 41. Data Acquisition Channels Details (TBD).

Megadac Channel Number	Measurement	Units	Sensor	Recommended Location	Voltage Range	Sample Frequency
0	Vehicle Speed	mph	Datron DLS Optical			
1	Lateral Acceleration	G's	Watson DMS-E604	Close to Tractor CG Inside Cab		
2	Roll Angle	Deg				
3	Yaw Angle (Sideslip Angle?)	Deg				
4	Roll Rate	deg/s				
5	Pitch Rate	deg/s				
6	Yaw Rate	deg/s				
7	X Acceleration	G's				

Example, Not Actual Data

Table 42. Tire Change Order.

Test Series	Tractor Steer	Tractor Drive/Trailer	Test Maneuver
1	Standard	NGSWBTs	CRTM
2	Standard	NGSWBTs	RSM
3	Standard	NGSWBTs	SSM
4	Standard	NGSWBTs	HEM
Tire Change, Two New Steering Tires[1], Replace NGSWBTs with Duals			
5	Standard	Standard duals	CRTM
6	Standard	Standard duals	RSM
7	Standard	Standard duals	SSM
6	Standard	Standard duals	HEM

[1]Tire change will require mounting new tire(s) to rims.

Table 43. Tire Size and Tread Type.

Tire Mfg	Tire Size	Model Number	Tire Pressure (Bar)	Tire Pressure (PSI)
Michelin	Steer 275/80R22.5	XZA3	7.9	116
Michelin	Drive 275/80R22.5 (dual)	XDA Energy	7.1	104
Michelin	Trailer 275/80R22.5 (dual)	XTA Energy	7.1	104
Michelin	Drive 445/50 R22.5 (NGSWBTs)	X One XDA	7.9	116
Michelin	Trailer 445/50 R22.5 (NGSWBTs)	X One XTA	7.9	116

VEHICLE TEST EVENTS (MANEUVERS)

A series of four test events have been selected to maximize the desired data while minimizing testing time and resources. The four test events are described in detail, below. The data acquired from the selected events will ultimately be correlated with virtual simulation model results. Due to the amount of open space required to perform the selected maneuvers, all four maneuvers will be performed in the Vehicle Dynamics Area of the facility near the center of the test-track (see figure 198).

The speeds, steering input angles, and articulation angles proposed in this test plan are estimates based on the response of the vehicle observed during previous testing attempts. The actual speed and angles used at the track may be adjusted for the new load configuration, if necessary.

Step Steer Maneuver

Background**:** The Step Steer test will provide information regarding sensitivity between the steering wheel input and the resulting yaw rate, lateral acceleration and roll of the vehicle as a function of steering input. This information will better facilitate the understanding of the total vehicle system response. The Step Steer maneuver used for this test set is based on ISO 7401.

Setup: This is a straight-line test with a single quick turn to a specified steering wheel angle. Sufficient test area is required to operate the test vehicle at a constant speed of 35 mph for approximately 30 seconds, with room to attain the test speed and to decelerate at the end of the test.

Ballasting:	GVWR:	approximately (but not exceeding) 80,000 lbs
	UVW:	approximately 38,950 lbs

Vehicle Control**:** Human driver

Procedure: A human driver will provide the steering input angle required to achieve no more than (and likely less than) a 0.4 g peak lateral acceleration (estimated to be the lateral acceleration required to achieve wheel lift-off) at 30 mph (slow speed). Some adjustment may be necessary when this procedure is done using different tires. The steering wheel input rate will be in 150 deg/sec range (actual value TBD).

The test shall be repeated six times at 30 mph.

Using a bracketing method, the speed corresponding to wheel lift-off without the ESC engaged shall be determined. The test shall be repeated six times at this speed. The ESC shall then be turned "ON," and the test shall be repeated until a consistent ESC activation pattern is observed (up to a maximum of six times).

Two additional runs with the ESC "ON" will be made at a speed above the wheel lift-off speed (TBD, mostly wheel lift-off speed plus two mph).

It shall be verified that sufficient data has been obtained at the end of this set of test maneuver.

Highway Evasive (Single Lane Change) Maneuver

Background: This test is conducted to determine the transient response of a vehicle when subjected to a sudden lane change. The resulting data will be used to study the transient roll control characteristics of vehicle attitude and rollover.

Setup: The Single Lane Change Course will be set up with pylons as shown in figure 201. It will be set up on a level asphalt area of sufficient size to accommodate safe entry and exit of the course for the size and weight of the vehicle being tested.

Ballasting:	GVWR	approximately (but not exceeding) 80,000 lbs
	UVW	approximately 38,950 lbs

Vehicle Control: Human driver

Procedure: The test consists of negotiating a left-hand lane change (course shown in sketch). The human driver will attempt to maintain a constant speed throughout this test maneuver (i.e., upon entering Gate 1 and until exiting Gate 2). Gate 1 will be entered from the left at constant speed. Upon exiting Gate 1, the human driver will make a quick turn to the left and then back to the right in order to line up the vehicle to enter Gate 2. The driver will initially make a total of six low-speed runs, negotiating the course at 25 mph.

After the six low-speed runs, the speed will be increased in increments of 2 mph until the rollover threshold is met (outrigger contact with the ground is likely). Once the rollover threshold speed has been determined, the test will be repeated six times.

After six runs at the rollover threshold speed, the human driver will repeat the runs with the ESC "ON" until a consistent ESC activation pattern is observed (up to a maximum of six times).

Two additional runs will be made with the ESC "ON" at a speed above the wheel lift-off speed (TBD, mostly wheel lift-off speed plus two mph).

It shall be verified that sufficient data has been obtained at the end of this set of test maneuvers.

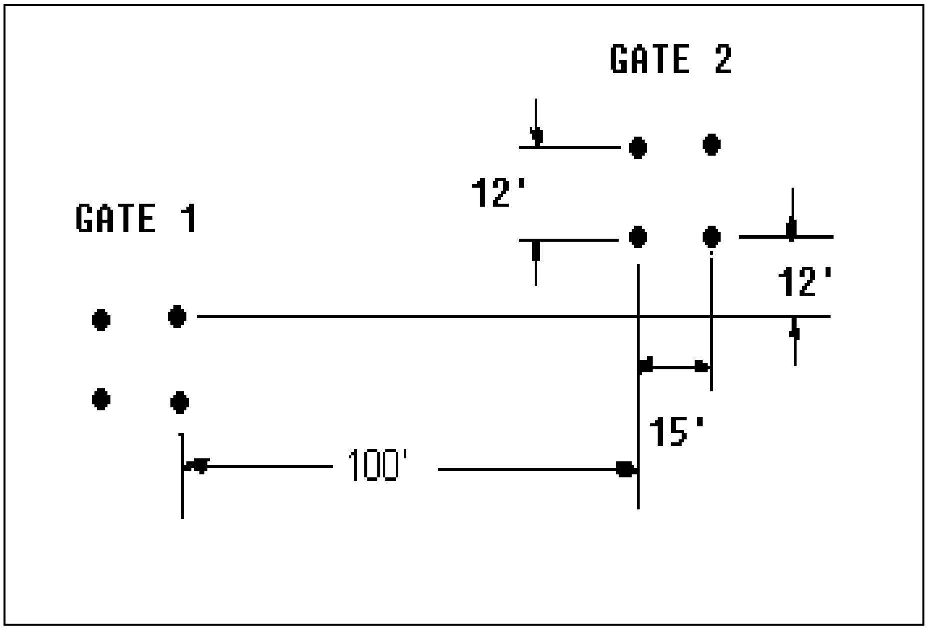

Figure 201. Diagram. Evasive Maneuver Pylon Layout.

Steady State Constant Radius Turn – 400 ft Radius

Background: This test is conducted to understand the steady-state behavior of the vehicle (understeer, yaw rate gain, lateral acceleration gain, etc.). This test will be conducted for slow speeds only. Multiple low speeds (5mph, 10mph, 15 mph, etc.) will be addressed (up to 35 mph) where the respective speed is held constant for 60 seconds at each speed. The cornering ability of a vehicle can be quantified by the level of lateral acceleration that is sustained in a stable condition.

Setup: This procedure utilizes a constant radius paved circle of 400 ft. The surface coefficient of friction shall be measured by TRC staff and documented.

Ballasting:	GVWR:	approximately (but not exceeding) 80,000 lbs
	UVW:	approximately 38,950 lbs

Vehicle Control: Human driver

Procedure: The human driver will attempt to maintain a constant circular path by adjusting the steering angle as necessary. Beginning from a static position, the speed of the vehicle will be increased to 5 mph. Once reaching this speed, it will be held for 60 seconds. The speed will then be increased in increments of 5 mph and held for 60 seconds until reaching 35 mph. The test will be repeated, but in the opposite direction. Both of these tests are performed with the ESC “OFF.”

NOTE: Previous attempts at achieving wheel lift-off during the constant radius maneuver were unsuccessful, and as a result, the ramp steer maneuver was added to the Test Plan to achieve wheel lift-off. If the vehicle is able to achieve wheel lift-off during the constant radius test with the single tires (step 1 of table 44), and with the new CG height(s), the ramp steer maneuvers in table 44 will not be performed. Instead, the additional constant radius turn maneuvers below will be conducted at speeds up to wheel-lift.

Additional CRTMs:

The test will involve the vehicle starting from a static position. While maintaining a constant turn radius, the vehicle's speed is increased by 10 mph per minute. The human driver will continue to increase the vehicle's speed by 10 mph per min until wheel lift-off occurs. This test will be repeated five additional times. This test is performed with the ESC "OFF."

The driver will then turn the ESC "ON" and travel in the opposite direction as the previous portion of the test. The driver will accelerate to a speed which is near the speed at which wheel lift-off occurred, and will increase his speed by 10 mph per minute until ESC actuation is experienced. This test will be repeated one additional time at this speed with the ESC "ON."

It shall be verified that sufficient data has been obtained at the end of this set of test maneuvers.

<u>Ramp Steer Maneuver</u>

Background: This test is conducted to determine the maximum lateral acceleration that can be achieved by a vehicle under "near-steady-state" cornering conditions. This test will be performed in the event that wheel lift-off cannot be achieved during the constant radius turn maneuver. If wheel lift-off is achieved in the constant radius maneuver, this test will not be performed. The cornering ability of a vehicle can be quantified by the level of lateral acceleration that can be sustained during this maneuver.

Setup: The TRC vehicle dynamics area will be utilized for this test. The surface coefficient of friction shall be measured by TRC staff and documented.

Ballasting:	GVWR:	approximately (but not exceeding) 80,000 lbs
	UVW:	approximately 38,950 lbs

Vehicle Control: Human driver

Procedure: This is a straight-line test with a gradual turn to a specified steering wheel angle. A sufficient test area is required to operate the test vehicle at a constant speed of up to 35 mph (the achievable speed will be determined at TRC) for approximately 45 seconds, with room to attain the test speed and to decelerate at the end of the test. The test will begin from a static position and the vehicle's speed will be increased to 20 mph (target speed). Once the target speed is reached, the human driver will attempt to maintain a constant speed and gradually turn the steering wheel at a constant rate until the target steering wheel angle is reached. Once successfully accomplished, the run will be repeated at the same speed.

If wheel lift-off is not achieved, the human driver will increase the vehicle's speed by 1-to-2 mph every other pass until wheel lift-off is achieved or until an unsafe speed (as determined by TRC or the research team) is reached; whichever comes first. When a speed is reached that produces wheel lift-off, the maneuver will be repeated five additional times at the same speed. This test is performed with the ESC "OFF."

Following the testing with the ESC "OFF," the test will be repeated at the wheel lift-off speed with the ESC "ON" until a consistent ESC activation pattern is observed (up to a maximum of six times).

Two additional runs with the ESC "ON" will be made at a speed above the wheel lift-off speed (TBD, mostly wheel lift-off speed plus two mph).

Due to the test-track layout and the allowable direction of travel around the test loops (see figure 198), this test maneuver will be conducted only in one direction.

It shall be verified that sufficient data has been obtained at the end of this set of test maneuvers.

TEST MANEUVER AND TIRE CONFIGURATION ORDER

The four maneuvers described above will be performed with both tire configurations, two different load CGs heights, and two ESC states (OFF and ON). The NGSWBT tire configuration will be tested only with the lower load CG height.

Table 44 lists the order of test maneuver accomplishment, and indicates where, in the entire rollover testing regimen, that data checks will be conducted.

Table 44. Test Maneuver and Tire Configuration Schedule.

Step No	Tire Configuration	Test Maneuver	Repetitions[1]	Direction	ESC	Load CG Height[3]
1	Singles	Const. Radius, to maximum speed limit (open differential) or wheel lift-off, for vehicle handling observation	1	Clockwise	OFF	9 ft
2	Singles	Tire pressure check				
3		Data Check, after first run				
4	Singles	Ramp Steer[4], speed and angle rate TBD	1	Clockwise	OFF	9 ft
5		Data Check				
6	Singles	Ramp Steer[4], speed and angle rate TBD	5	Clockwise	OFF	9 ft
7	Singles	Ramp Steer[4], speed and angle rate TBD	2[2]	Clockwise	ON	9 ft
8	Singles	Ramp Steer[4], wheel lift-off speed plus[5], TBD	2	Clockwise	ON	9 ft
9		Data Check				
10	Singles	Cont. Radius, 10 to 35 mph using 5 mph/min ramp	1	Clockwise	OFF	9 ft
11	Singles	Cont. Radius, 10 to 35 mph using 5 mph/min ramp	1	Counter-Clockwise	OFF	9 ft
12		Data Check				
13	Singles	Step Steer @ 30 mph, 180°	1	Right	OFF	9 ft
14		Data Check, after first run				
15	Singles	Step Steer @ 30 mph, 180°	5	Right	OFF	9 ft
16	Singles	Lane Change @ 25 mph	6	Left	OFF	9 ft
17		Data Check				
18	Singles	Step Steer, bracket speed at 180° for wheel lift-off	1	Right	OFF	9 ft
19		Data Check, after first run				
20	Singles	Step Steer 180° and speed for wheel lift-off	5	Right	OFF	9 ft
21	Singles	Step Steer 180° and speed for wheel lift-off	2[2]	Right	ON	9 ft
22	Singles	Step Steer 180° and wheel lift-off speed plus[5], TBD	2	Right	ON	9 ft
23		Data Check				
24	Singles	Lane Change until speed is bracketed	1	Left	OFF	9 ft
25	Singles	Data Check, after first run				
26	Singles	Lane change at wheel lift-off speed	5	Left	OFF	9 ft
27	Singles	Lane change at wheel lift-off speed	2[2]	Left	ON	9 ft
28	Singles	Lane change at wheel lift-off speed plus	2	Left	ON	9 ft
29		Data Check				

Step No	Tire Configuration	Test Maneuver	Repetitions[1]	Direction	ESC	Load CG Height[3]
30		**Tire Change, Duals**				
31		Tire warm-up/pressure check	-	30 min recommend		
32	Duals	Ramp Steer[4], speed and angle rate TBD	1	Clockwise	OFF	9 ft
33		Data Check				
34	Duals	Ramp Steer[4], speed and angle rate TBD	5	Clockwise	OFF	9 ft
35	Duals	Ramp Steer[4], speed and angle rate TBD	2[2]	Clockwise	ON	9 ft
36	Duals	Ramp Steer[4], wheel lift-off speed plus[5], TBD	2	Clockwise	ON	9 ft
37		Data Check				
38	Duals	Constant Radius, 10 to 35 mph using 5 mph/min ramp	1	Clockwise	OFF	9 ft
39	Duals	Constant Radius, 10 to 35 mph using 5 mph/min ramp	1	Counter-Clockwise	OFF	9 ft
40		Data Check				
41	Duals	Step Steer @ 30 mph, 180°	1	Right	OFF	9 ft
42		Data Check, after first run				
43	Duals	Step Steer @ 30 mph, 180°	5	Right	OFF	9 ft
44	Duals	Lane Change @ 25 mph	6	Left	OFF	9 ft
45		Data Check				
46	Duals	Step Steer, bracket speed at 180° for wheel lift-off	1	Right	OFF	9 ft
47		Data Check, after first run				
48	Duals	Step Steer 180° and speed for wheel lift-off	5	Right	OFF	9 ft
49	Duals	Step Steer 180° and speed for wheel lift-off	2[2]	Right	ON	9 ft
50	Duals	Step Steer 180° and wheel lift-off speed plus[5] TBD	2	Right	ON	9 ft
51		Data Check				
52	Duals	Lane Change until speed is bracketed	1	Left	OFF	9 ft
53		Data Check, after first run				
54	Duals	Lane Change at wheel lift-off speed	5	Left	OFF	9 ft
55	Duals	Lane Change at wheel lift-off speed	2[2]	Left	ON	9 ft
56	Duals	Lane Change at wheel lift-off speed plus[5] (TBD)	2	Left	ON	9 ft
57		Data Check				
58		**Change load CG height to 11 feet**				
59		Tire warm-up/pressure check	-	30 min recommend		11 ft

Step No	Tire Configuration	Test Maneuver	Repetitions[1]	Direction	ESC	Load CG Height[3]
60	Duals	Ramp Steer[4], speed and angle rate TBD	1	Clockwise	OFF	11 ft
61		Data Check				
62	Duals	Ramp Steer[4], speed and angle rate TBD	5	Clockwise	OFF	11 ft
63	Duals	Ramp Steer[4], speed and angle rate TBD	2[2]	Clockwise	ON	11 ft
64	Duals	Ramp Steer[4], wheel lift-off speed plus, TBD	2	Clockwise	ON	11 ft
65		Data Check				11 ft
66	Duals	Constant Radius, 10 to 35 mph using 5 mph/min ramp	1	Clockwise	OFF	
67	Duals	Constant Radius, 10 to 35 mph using 5 mph/min ramp	1	Counter-Clockwise	OFF	11 ft
68		Data Check				
69	Duals	Step Steer @ 30 mph, 180°	1	Right	OFF	11 ft
70		Data Check, after first run				11 ft
71	Duals	Step Steer @ 30 mph, 180°	5	Right	OFF	11 ft
72	Duals	Lane Change @ 25 mph	6	Left	OFF	
73		Data Check				11 ft
74	Duals	Step Steer, bracket speed at 180° for wheel lift-off	1	Right	OFF	
75		Data Check, after first run				11 ft
76	Duals	Step Steer 180° and speed for wheel lift-off	5	Right	OFF	11 ft
77	Duals	Step Steer 180° and speed for wheel lift-off	2[2]	Right	ON	11ft
78	Duals	Step Steer 180° and speed for wheel lift-off plus, TBD	2	Right	ON	11ft
79		Data Check				11 ft
80	Duals	Lane Change until speed is bracketed	1	Left	OFF	
81		Data Check, after first run				11 ft
82	Duals	Lane Change at wheel lift-off speed	5	Left	OFF	11 ft
83	Duals	Lane Change at wheel lift-off speed	2[2]	Left	ON	11ft
84	Duals	Lane Change at wheel lift-off speed plus[5] (TBD)	2	Left	ON	11ft
85		Data Check				

[1]Does not include passes made to bracket the wheel lift-off speed.
[2]The variability of the data at the point the ESC engaged will be checked to determine if two repetitions are sufficient. If not, more repetitions will be performed.
[3]The CG height is measured from the ground.
[4]The the Ramp Steer will only be performed if lift-off is not achieved with the CRTM for the Single tires.
[5]"Wheel lift-off speed plus" means that an addition increment of speed (TBD) will be added to the wheel lift-off speed as determined by the bracketing method.
Note: In addition to the tire warm up/pressure checks listed in Table 44, there will be additional tire warm up/pressures checks after every significant shutdown period.

TRACTOR CG MEASUREMENT

The research team has agreed that accurately knowing the CG height of the test tractor is very important for the project modeling efforts. TRC has indicated that this test was simple, could be performed quickly, and could be performed by them at the end of the test-track testing. The research team recommends that NTRCI fund the execution of this test by TRC.

CONTINGENCY TEST PLANS

The test outlined above can be modified to fit contingencies that might arise while at the test-track. That is, in the unlikely event that the fourth attempt at testing is not successful, the following contingency plans have been developed as a fall-back. It should be noted that there are very few major options remaining and that if the fourth attempt is unsuccessful, the best that can be achieved is to collect as much data from test that can be collected before the testing portion of this project comes to an end.

Contingency Plan A

If wheel lift-off cannot be achieved in a safe manner with the load CG height at 9 feet (the trigger event for Contingency Plan A), the load CG heights in the Test Plan will be changed from 9 and 11 feet, to 10 and 12 feet, respectively.

Contingency Plan B

If after engaging in Contingency Plan A, TRC determines that the vehicle reaction is too unstable with the tractor wheels lifting first (the trigger event for Contingency Plan B), the ballast above the tractor drive axles will be lowered to the deck of the trailer and the ballast above trailer axles will remain elevated. This configuration should cause the wheels on the trailer to lift before the drive axle wheels on the tractor.

Contingency Plan C

If after engaging Contingency Plan B, TRC determines that the lift reaction with the current tractor is still too unstable, then test-track testing to achieve wheel lift-off will not be possible. In the unlikely event that this occurs, all remaining slow-speed testing will be conducted, as well as all testing with the ESC "ON." Following the completion of this testing regimen, MARC has agreed to transport Michelin's Volvo tractor (a tractor with a shorter tractor frame, but without ESC) to TRC to conduct a quick study of the effects of changing the test tractor. The test instrumentation will not be hooked up, but a test maneuver (TBD) will be engaged in to determine if the response of the Michelin Volvo is dissimilar to that of the test tractor. This will allow a gross determination of whether the tractor design may have an influence on the sudden and dynamic wheel lift-off behavior. Michelin's Volvo tractor will be on standby as testing is resumed in late-April/early-May.

By performing the constant radius turn maneuver up to wheel lift-off speed and the ramp steer maneuver up to wheel lift-off speed first, the appropriate contingency plan (if needed) can be selected at the beginning of the test. This will minimize the risk of needing to switch to a different contingency plan (load configuration) during the middle test and having to repeat test maneuvers.

RESOLUTION OF CONCERNS

As a result of the three prior testing attempts, a number of issues have emerged that have been addressed by the team, and are elucidated here.

CRITICAL ELEMENTS IN MOVING FORWARD

The following items were identified as critical elements needing to be addressed before testing could proceed:

1. What are the options for achieving a successful test?
2. Are there options in addition to CG height modification and load distribution that should be considered?
3. Should the loading racks be constructed in such a way that would allow researchers to change the torsional stiffness of the trailer?
4. Should the torsional stiffness of the loaded flatbed be tested prior to the next testing attempt?

The research team feels that it has invested considerable time and effort within team discussions, discussions with external subject matter experts, analyses via TruckSim models (MARC's model and WMU's model), and discussions with experts at TRC to explore all of the contingencies that can be contemplated for the fourth attempt at testing. The current Test Plan reflects our best strategy for moving forward with rollover testing of the tractor-flatbed. The protocols have been reviewed and adjusted as necessary, and the list of data that will be collected has been reviewed by the data analysts and found to be sound.

What are the Options for Achieving a Successful Test?

For the initial testing attempt, the unexpected sudden and dynamic lift-off of the tractor drive axle and subsequent violent slamming-down of the axle on the test-track was an issue for concern. The outriggers had been mounted near the rear of the flatbed because the team expected, and the literature indicated that the rear trailer axle would be the axle which first experiences lift-off. The research team had discussed the sequence of events prior to testing to decide where the outriggers should be mounted. As a result they were mounted in such a way as to prevent a rollover stemming from a trailer axle-led event. When the tractor drive axle lifted first, the outriggers were ineffective in preventing the tractor from turning over.

The research team felt achieving a successful test was therefore related to the proper placement of the outriggers. After some discussion, the research team decided that it was likely that if the maneuver was repeated, that the tractor drive axle would also lead in the event. As a result, the team decided to move the outriggers to a more forward position on the flatbed. The second attempt was engaged in and occurred as expected, however with some additional concerns. Although the drive axle lifted first and the outriggers prevented the tractor from rolling over, a significant amount of "wind-up" was experienced in the trailer. Because the outriggers had been moved forward to protect the tractor, this movement left the flatbed unprotected. After some discussion, the team felt that no matter where the outriggers were placed, the tractor-flatbed could not be adequately protected by one set of outriggers. It was determined that a second set of outriggers had to be added; one to protect the tractor, the other to protect the flatbed. As a result, MARC shipped its outriggers to TRC, and a third set of experiments were planned.

It should be noted that up to this point that lift-off was being achieved, however it was more sudden and dramatic than what was expected, and the principal issue to overcome was the proper placement of the outriggers.

With the two outriggers placed on the trailer; one forward to protect the tractor and one rearward to protect the flatbed; and with the payload adjusted to compensate for the added weight of the second set of outriggers; the third testing event was engaged in. The third attempt, however, reached a speed of 55 mph in a transient maneuver, and no axle was lifting off. TRC halted the tests because they felt that if the speed was increased farther than 55 mph, a sudden and dramatic lift-off would occur, and that at this higher speed, that the outriggers would be ineffective in preventing a truck rollover. This was a totally new concern for the research team to consider.

After considerable discussion, the research team felt that although they had solved the outrigger placement problem that the trailer load needed to be configured differently in order to assure wheel lift-off at a lower speed. It was decided that the CG Height of the current ballast was too low. Raising the CG of the ballast would allow lift-off to occur at a slower speed, and subsequently would assure that the outriggers could do the job they were intended to do. A second but related concern was the distribution of the ballast. It was felt that the homogeneous loading along the trailer spine may be negatively impacting the torsional compliance of the trailer. As a result, it was determined by the research team that the load needed to be decoupled, and that half of the load should be placed over the drive axle while the other half would be placed over the rear trailer axle.

In order to have a decoupled load with a raised CG, it was determined that a set of loading racks needed to be constructed. Furthermore, these racks should be such that the ballast could be adjusted so that its CG could be changed.

There was some discussion about the possibility of saving project funding by proceeding with only one load rack instead of a pair of racks. There was agreement that it might be possible to achieve a successful test with only one loading rack, but the ability to use two loading racks would give the flexibility needed to result in a successful test, and had more face validity with regard to loads on public highways.

Lastly, the team discussed the adjustment options for the fourth testing attempt. With two sets of outriggers, the tractor and the trailer are protected from rollover. With two decoupled load racks, the trailer is de-sensitized to torsional compliance issues. With the ability to vary the CG height, a slower wheel-lift-off speed is expected. In discussions with TRC, it was determined that short of re-designing the tractor-flatbed, the adjustment options available for the fourth attempt are all of the achievable options that are available.

Recommendation One – Construct and Utilize a Set of Decoupled Load Racks

The research team of ORNL, MARC, Clemson, WMU and Battelle recommends that NTRCI invest project funding into the construction, mounting and initial loading of two load racks (per the specifications for the racks that NTRCI currently has). It will also be necessary for TRC to remove the existing load frames and ballast in order to mount the new load racks. At the conclusion of the testing, TRC will leave the new load racks on the flatbed but put the weights in their lowest CG configuration for transport back to MARC. The original Michelin-owned load racks, unused weights and outriggers will be shipped to MARC.

The research team feels that the adjustment options available via the new load racks will allow for the accomplishment of a successful fourth attempt at testing.

Are There Options Other Than CG Height Changes and Load Distribution?

This topic received considerable discussion amongst the research team. When the vehicle dynamics equations were examined, CG height, stiffness and velocity are the primary contributors to the rollover propensity. These parameters were being addressed through the use of the load racks. In addition, the tractor and trailer would be protected from rolling over because of the dual set of outriggers. In one discussion, it was suggested that one other option that existed, but was cost prohibitive, was the re-design of the tractor-flatbed system. This suggestion suggested that perhaps in the unlikely event that the fourth round of testing was not successful, that a different tractor, with a shorter frame be substituted as a last attempt effort (see contingency Plan D). Use of this different tractor would be a substitute for re-designing the tractor-flatbed configuration. The research team concluded that for the fourth testing attempt that there were no other adjustment options available, that the flexibility inherent in the adjustment options were sufficient to ensure a successful test, and that Contingency Plan D would be considered in the unlikely event that the fourth attempt at testing was unsuccessful.

Should the Loading Racks be Constructed to Allow Researchers to Change the Torsional Stiffness of the Trailer?

The research team discussed the option of constructing the load racks in such a way as to allow for the capability of adjusting the torsional stiffness. To pursue this option, the design of the load racks would be such that it would have a removable top section that would change the stiffness of the system. After considerable debate, it was concluded that this option, while interesting would necessitate the conduct of additional runs to be accomplished, and with parameters such as NGSWBTs/standard duals, ESC ON/OFF, and with two different CG heights, the resources were not available to take appropriate advantage of this experimental parameter option. Furthermore, the research team felt that if it were important to change the torsional stiffness of the flatbed, other, albeit less elegant means, are likely available

Recommendation Two – Do Not Construct the Load Racks to Facilitate a Change in the Torsional Stiffness of the Trailer

The research team of ORNL, MARC, Clemson, WMU and Battelle recommends that NTRCI not invest project funding into the construction of two-piece load racks.

Should the Torsional Stiffness of the Loaded Flatbed be Tested Prior to the Next Testing Attempt?

WMU indicated that they have some limited resources to conduct a torsional stiffness test of the flatbed trailer in a loaded condition. The research team felt that having torsional stiffness data of the trailer in the loaded condition was important in trying to understand the dynamics associated with the first three testing events. In order to accomplish this effort, the tractor had to be disconnected from the flatbed. Fortunately, the instrumentation cabling was sufficiently long to disconnect the fifth wheel without disconnecting the instrumentation. Initially it was feared that in order to do the torsional stiffness test that it would have to be done inside of a garage which would require the outriggers to be removed from the flatbed. If the torsional stiffness testing could be done just before the new load racks were to be mounted on the flatbed, the weather would likely not require the testing to be done within the garage. As a result, the outriggers would not have to be removed; saving about three days labor. Furthermore, it was suggested, and WMU agreed, that one set of torsional stiffness tests could be conducted with the currently distributed ballast (just prior to TRC mounting the new load racks), and a second set of torsional stiffness tests could be conducted after TRC had mounted and loaded the newly constructed load racks. The project funding saved by not having to dismount and remount to two sets of outriggers will more than compensate for the second set of torsional stiffness tests. Furthermore, conducting the torsional stiffness tests with the outriggers mounted will give more realistic results when the derived data is used within the TruckSim model. This information would be extremely valuable in understanding the dynamics associated with the first three testing attempts, and would also allow the modelers to assess what might be expected from the new load distributions.

Recommendation Three – WMU Should Conduct Torsional Stiffness Testing on the Loaded Flatbed for the Current Loading and New Loading Configurations

The research team of ORNL, MARC, Clemson, WMU and Battelle recommends that NTRCI invest project funding into WMU conducting two sets of torsional stiffness tests on the loaded flatbed.

MAXIMIZING ATTEMPT-4 SUCCESS

The research team has invested considerable time and effort into examining and discussing the fourth testing attempt. WMU and MARC have both engaged in running their TruckSim models and feel confident that with the adjustment options that are now available to the team that there is a high probability of success. Several conference calls have been engage in with TRC to identify the envelope of performance that would be acceptable to them. TRC provided the following stop-test criteria: if the experiment did not exceed 40 mph and a 20-degree articulation angle there was very little likelihood that the test would be halted. With these parameters, both MARC and WMU conducted analyses that indicated the CG height necessary to achieve wheel-lift-off at a speed less than 40 mph. This CG height was not very realistic, but obtainable, and formed the basis for recommending the general design criteria of the load racks to TRC. As a result, based on discussions with TRC and on the analyses conducted by MARC and WMU, there is a high probability that the fourth attempt at testing will be successful. It should be noted that the load racks will have several loading heights, and that the forward rack could be loaded at a different CG height than the rearward rack. With this flexibility, adjustments can be made that will contribute to achieving positive results in the fourth attempt at testing.

Known-Unknowns And Unknown-Unknowns

The research team has taken long, hard look at all controllable factors within the current experimental design. Despite the best made plans, no experimental research can guarantee success. It is the nature of research to involve certain levels of risk. Since the third testing attempt in late-January, 2008, the research team has engaged in numerous conference calls, exchanged countless e-mails, has run two versions of the TruckSim model, conferred with subject matter experts outside of the project, and discussed the testing with the experts at TRC.

With this background, expertise, and analyses, the research team feels that nothing more can be done to minimize the risk associated with the fourth attempt at testing. As such, the research team feels confident that the known-unknowns have been addressed to the extent possible given the project resources. Regarding the unknown-unknowns; if the research team has addressed the known-unknowns to the greatest extent possible and reasonable, then the opportunities for the occurrence of unknown-unknowns is minimized. Beyond this minimization, there is very little else that can be done.

Increased Staffing Emphasis For Fourth Attempt Testing

The research team feels that one of the best ways to address any uncertainties that might arise in the testing will be to have a number of the research team members present as the fourth attempt at testing gets underway. It is suggested that at a minimum, the following people be present for the testing, the stated purposes:

Michael Arant (MARC): Test development, modeling correlation, data analysis.
Nathan Wood (ORNL): Test development, data acquisition, data verification.
Gary Capps (ORNL): Test development, especially in correlating this testing to previous phases, in particular, the lane change testing.
Oscar Franzese (ORNL): Data acquisition.

This team can also assist in bracketing the adjustment options at the beginning of the test, and will be able to quickly respond in the unlikely event that any unknown-unknowns occur. In addition, the data analysts will be on-site to assure that valid data is taken during testing.

APPENDIX

Definitions

Gross Axle Weight Rating (GAWR): Value specified by the vehicle manufacturer as the load-carrying capacity of a single axle system, as measured at the tire-ground interfaces.

Gross Vehicle Weight Rating (GVWR): Value specified by the manufacturer as the loaded weight of a single vehicle.

Unloaded Vehicle Weight (UVW): Weight of a vehicle with maximum capacity of all fluids necessary for operation of the vehicle, but without cargo or occupants, or accessories that are ordinarily removed from the vehicle when it is not in use.

Lift-Off**:** For this test, lift-off will be defined as the first point during a maneuver when the load on one side of any axle goes to zero. Lift-off may or may not be visible and may occur on the tractor or trailer.

Data Check**:** Data reviewed to ensure all sensors are working properly and the signal levels are within the "normal range." For example, the data from a speed sensor would be checked to ensure that the value recorded is a positive number and that it is physically possible (e.g., below 120 mph). When necessary, the data will also be reviewed to ensure that the maneuvers preformed are producing the desired events (e.g., lift-off) and the data needed for analyses.

TRC will be responsible for providing the data at the requested times for Data Checks. The Rollover research team will be responsible for performing the actual data checks**.**