

Low-Speed Maneuverability, High-Speed Roll-Stability, and Brake Type Performance of Heavy Truck 33-ft Double Trailers

Campbell Neighborgall

Dissertation submitted to the faculty of the Virginia Polytechnic Institute and State University in partial fulfillment of the requirements for the degree of

Doctor of Philosophy
In
Mechanical Engineering

Mehdi Ahmadian, Chair
Kaveh A. Hamed
Steve C. Southward
Daniel J. Stilwell
Saied Taheri

06/29/2022
Blacksburg, Virginia

Keywords: Vehicle dynamics, long combination vehicles, off-tracking, disc brakes, drum brakes, roll-stability control, electronic stability control

Copyright 2022, Campbell Neighborgall

Low-Speed Maneuverability, High-Speed Roll-Stability, and Brake Type Performance of Heavy Truck 33-ft Double Trailers

Campbell Neighborgall

ABSTRACT

This dissertation details the methods and analysis of extensive physical tests and simulation conducted by the Center for Vehicle Systems and Safety (CVeSS) at Virginia Tech on the maneuverability, roll-stability, and brake type performance of 33-ft double trailers. Little literature exists for 33-ft doubles because they are uncommon on the U.S. roads due to current federal restrictions limiting long-combination vehicles to 28-ft doubles. With the continual rise in e-commerce, however, there is a push by package carriers on legislation to permit carriers to introduce 33-ft doubles into their fleets. Three separate studies detailed herein highlight 33-ft double trailers' off-tracking, roll-stability with stability control systems, and brake type influence on braking performance.

The first study compares low-speed off-tracking of a 33-ft double to 28-ft double and 53-ft single configurations via simulation and full-scale tests. Novel numerical tractrix models are introduced and compared to existing models commonly used to evaluate low-speed off-tracking of long combination vehicles (LCVs). Unlike pre-existing models, accuracy of one of the proposed models is largely unaffected by input path resolution and regularity—a significant benefit for reducing computational cost and easing implementation for many applications. Full-scale tests are conducted at Virginia Tech and an extensive uncertainty analysis is detailed for the test procedure and measurements. Field tests compare favorably with simulations for all tested maneuvers and trailer configurations and clearly demonstrate the order from least to most off-tracking as 28-ft double, 33-ft double, and 53-ft single. The 33-ft doubles have slightly larger off-tracking than 28-ft doubles, whereas 53-ft singles have substantially larger off-tracking than 28-ft and 33-ft doubles.

The second study evaluates 33-ft double straight-rail trailers rollover propensity with different stability control system implementations: stock (none), tractor electronic stability control (ESC), trailer roll-stability control (RSC), and RSC+ESC. Extensive test vehicle instrumentation and structural reinforcement are detailed for the test preparations. Tests are conducted on a test track with either driver or robot steering. On their own, both ESC and RSC clearly reduce the rollover propensity of the trailers for all maneuvers, and

the trailers exhibit the highest roll-stability when both RSC and ESC are active. The tested ESC and RSC modules are off-the-shelf products from industry suppliers chosen by the program sponsor.

The third study compares trailer drum and disc brake performance in three conditions: straight-line braking distance, brake type influence on RSC performance, and roll dynamics in a combined braking and turning maneuver. A braking robot is designed, fabricated, and implemented to provide precise and repeatable brake pedal application. Test results suggest that disc brakes tend to provide reduced braking distance and are less susceptible to brake fade than drum brakes. Anti-lock braking system (ABS) and suspension dynamics react differently to the two brake types. Small, noticeable differences in RSC performance are evident between the two brake types. Within the test limitations, rollover dynamics were not clearly different between the two brake types for braking-in-turn maneuvers, performed for a large range of entry speeds and brake activation delay relative to the start of steering.

Low-Speed Maneuverability, High-Speed Roll-Stability, and Brake Type Performance of Heavy Truck 33-ft Double Trailers

Campbell Neighborgall

GENERAL AUDIENCE ABSTRACT

Due to their large size, mass, and high center-of-gravity, heavy vehicles, especially long combination vehicles (LCVs) require a substantial amount of space to negotiate turns, long distances to brake from highway speeds to a stop and are susceptible to rollover. Combination vehicles on the U.S. roads are commonly in 53-ft single trailer or 28-ft double trailer configurations. With the continual rise of e-commerce, package carriers are pursuing 33-ft double trailers to increase each vehicle's cargo volume. Before introducing these trailers into a fleet, there is a need to understand (1) if 33-ft doubles can negotiate existing routes traveled by 28-ft double and 53-ft single configurations, (2) if 33-ft doubles can benefit from existing stability control systems, and (3) how trailer brake types perform on 33-ft doubles. Three separate studies are conducted to address these topics. The first study compares off-tracking for the three mentioned trailer configurations through low-speed, real-world maneuvers via physical full-scale tests and simulation. Off-tracking is a metric illustrative of maneuverability and is defined as the relative distance in paths of the rearmost axle to the lead steer axle. New mathematical models are introduced and used to simulate vehicle motion through low-speed maneuvers. The simulation and field tests determine that, for all tested maneuvers, the order from smallest to largest off-tracking is 28-ft double, 33-ft double, and 53-ft single configurations, with the 33-ft doubles having slightly larger off-tracking than 28-ft doubles. This suggests that 33-ft doubles can travel through routes typically traveled by a 53-ft single but need slightly more space on the road than a 28-ft double. The second study tests 33-ft double trailers with and without stability control systems. Tests, conducted at a test track, are designed to replicate real-world maneuvers that induce trailer rollover. It is found that the 33-ft double trailers are clearly less likely to rollover with the tested stability enhancement systems than without. The tests also illustrate that the different tested control systems' effectiveness in reducing rollover propensity is maneuver dependent. The third study tests the braking distance, brake influence on the stability control systems, and rollover dynamics while braking-in-turn for two different types of brakes, drum brakes and disc brakes. Small but evident differences in the performance of the two brake types suggest disc brakes could provide shorter stopping distance and time at highway speeds, compared with drum brakes. The studies detailed in this dissertation provide valuable information on 33-ft doubles dynamics and provide guidance for their safe introduction on the U.S. roadways.

Acknowledgements

I would like to first thank my family, especially my parents, for the endless support and encouragement over the years. From supporting me in Lego® robotics as a little kid to driving me across another state to visit the Corvette assembly plant for my 12th birthday, they have consistently gone out of their way to ensure I had opportunities to pursue my interests in engineering.

Secondly, I would like to thank my advisor, Dr. Mehdi Ahmadian, for giving me a chance at CVeSS many years ago as an undergraduate student. The culture of care, respect, and curiosity that he cultivates at CVeSS was a large factor in why I decided to stay at Virginia Tech and CVeSS for graduate school. I have learned countless things from him that apply to all aspects of life in and outside the lab. He is the hardest worker I know and is always happy to lend a helping hand to his students.

I would also like to thank my committee members, Dr. Steve Southward, Dr. Saied Taheri, Dr. Kaveh Akbari Hamed, and Dr. Dan Stilwell for their review and feedback on my research. They are exemplary examples of the many great professors at Virginia Tech who greatly care for their students and actively shape the future of engineering research.

There have been so many great students and researchers at CVeSS who I am fortunate to call life-long friends. I want to first thank my team, Dr. Yang Chen, Xiaohan Zheng, Zichen Zhang, and Zachary Van Kat. We had little to no experience going to the testing program and due to our diligent, persistent efforts we successfully completed the program while the world was enduring a global pandemic. Secondly, I would like to thank the support from Sara Vallejo and Crystal Funk for enabling us to obtain and maintain the necessary tools and equipment throughout all lab activities over the years (not to mention the office candy to keep spirits high). Additionally, I want to thank Carvel Holton for his mentorship and conversations on a variety of projects related and unrelated to this dissertation. I am also grateful for the many great friendships formed with other CVeSS members including Mike Molzon, Tim Mast, Dr. Yu Pan, Dr. Ahmad Radmehr, Aashish Shaju, Ian Larson, Mohammad Hosseini, Dr. Arash Ahangarnejad, Dwight Jones, Ali Sadeghi, and so many others.

Lastly, I would like to thank all my friends outside the lab who have supported me over the years at Virginia Tech and from afar, including my brother Quentin. A PhD requires an immense amount of sacrifice in time and energy, but the occasional breaks to play disc golf, take a long hike, mountain bike, workout, play board games, and hangout kept me grounded.

Table of Contents

Chapter 1 Introduction	1
1.1 Motivation	1
1.2 Objectives	3
1.3 Approach	4
1.4 Contributions	6
1.5 Outline	7
Chapter 2 Background and Literature Review	8
2.1 Terminology	8
2.2 Types of LCVs	12
2.3 Low-Speed Maneuverability	17
2.4 Overview of Uncertainty Analysis for Field Test Data	19
2.5 Roll- and Directional-Stability of LCVs	20
2.6 Stability Control Systems	24
2.6.1 Active and Passive Suspension to Improve Trailer Roll-Stability	25
2.6.2 Electronic Stability Control (ESC) and Roll-Stability Control (RSC)	28
2.7 Heavy Duty Vehicle Braking Systems	30
2.7.1 Trailer Drum Brakes	31
2.7.2 Trailer Disc Brakes	32
Chapter 3 Low-speed Off-tracking of LCVs – Comparison of Numerical Tractrix Models	33
3.1 Kinematic models	34
3.1.1 Bicycle Model and Vehicle Trajectory	34
3.1.2 Steady-State, Maximum Off-Tracking Model	35
3.1.3 Tractrix Model of Articulated Vehicle Motion	37
3.1.4 Simple Geometric Models of Long Combination Vehicle Motion	38
3.1.5 Model Sensitivity to Input Path Resolution	44
3.1.6 Trailer Configurations with Equivalent Steady-State Off-Tracking	53
3.2 Kinematic Models Compared to TruckSim® High-Order Dynamic Model	59
3.2.1 Maneuvers	59
3.2.2 Peak Off-Tracking Comparison	61
3.3 Off-Tracking Comparison of Common Trailer Configurations	62
Chapter 4 Low-speed Off-tracking Field Test and Instrumentation Calibration	67
4.1 Peak Off-Tracking Sensitivity Analysis	70
4.1.1 Peak Off-Tracking Sensitivity to Spatial Errors	71
4.1.2 Peak Off-Tracking Sensitivity to Temporal Errors	76
4.1.3 Sensitivity Analysis Summary	80

4.2	Test Setup and Procedure	81
4.3	Non-Contact Method to Measure Articulation Between Vehicle Units	87
4.3.1	Converting LiDAR Data to Articulation Angle	87
4.3.2	Improving LiDAR Performance by Increasing Trailer Surface Diffuse Reflectivity	90
4.3.3	Calibrating LiDAR Sensors for Articulation Angle Measurements	91
4.3.4	Determining LiDAR Location for Trailer-Trailer Articulation Angle Measurement	100
4.3.5	Accounting for Sensor Mounting Alignment on Vehicle	102
4.3.6	Time Synchronization of LiDAR Data	104
4.4	Estimating Tractor Steer Axle Path	104
4.5	Test Results and Uncertainty Propagation	111
Chapter 5	High-speed Vehicle Dynamics Test Vehicle and Equipment	118
5.1	Test Vehicle Parameters	118
5.2	Instrumentation and Data Acquisition	120
5.2.1	Sensor Selection and Calibration	121
5.2.2	Post-processing Digital Filtering	131
5.2.3	Junction Boxes	133
5.2.4	Data Acquisition and Synchronization	138
5.2.5	Cumulative Measurement Uncertainty	142
5.2.6	Preliminary System-wide Shakedown Testing	145
5.3	Steering Robot	146
5.3.1	Overview	146
5.3.2	Upfitting the CVeSS Steering Robot	147
5.3.3	Steering Robot Performance	150
5.4	Braking Robot	151
5.4.1	Brake Pedal Extension	152
5.4.2	Braking Robot Pneumatics	153
5.4.3	Partial Braking	155
5.5	Trailer Mechanical Safety Equipment	157
5.5.1	Outriggers	157
5.5.2	Anti-Jackknifing Mechanisms	158
5.5.3	Structural Reinforcements	159
5.5.4	Load Frames	161
Chapter 6	Stability Control Systems' Effect on Trailer Rollover Propensity	163
6.1	Test Maneuvers	163
6.1.1	Driver J-turn	163
6.1.2	Robot J-turn	164

6.1.3	Double Lane Change (DLC)	165
6.1.4	Sine-with-Dwell (SWD)	166
6.2	Determining Rollover Threshold from Test Data	167
6.3	Test Environment & Test Sessions	170
6.4	Tractor ESC and Trailer RSC Operation	172
6.5	Results & Analysis	175
6.5.1	Driver J-turn	175
6.5.2	Robot J-turn	179
6.5.3	Double-lane Change	184
6.5.4	Sine-with-Dwell	191
6.6	Conclusions	195
Chapter 7	Trailer Brake Type Performance Evaluation	201
7.1	Straight-line Braking Tests	202
7.1.1	Test Setup and Procedure	202
7.1.2	Braking Distance, Time, and Deceleration	204
7.1.3	Brake Temperature ABS Intervention	211
7.1.4	ABS Intervention	214
7.1.5	Braking-Induced Suspension Jounce	215
7.2	Braking-in-turn Maneuver Testing	221
7.2.1	Results Analysis of Full-Braking J-turn	222
7.2.2	Result Analysis of Partial-Braking J-turn	224
7.2.3	Conclusions from Braking-in-turn Maneuvers	228
7.3	Trailer Brake Type Effect on RSC Performance	228
7.3.1	Robot J-turn	228
7.3.2	Driver J-turn	229
7.3.3	Double Lane Change (DLC)	230
7.3.4	Sine with Dwell (SWD)	231
7.3.5	Comparing Brake Type Effect on Rollover Speed & Deceleration with RSC On & ESC Off	232
7.3.6	Comparing Brake Type Effect on Rollover Speed & Deceleration with Both RSC & ESC On	234
7.4	Conclusions on Trailer Brake Type Performance Comparison	235
Chapter 8	Conclusions and Recommendations	236
8.1	Summaries and Conclusions	236
8.1.1	Low-Speed Off-tracking & Maneuverability	236
8.1.2	Rollover Propensity and Stability Control Systems	237
8.1.3	Trailer Brake Type Performance Evaluation	237

8.2	Recommended Future Work	239
8.2.1	Tire Slip Effect on Off-tracking	239
8.2.2	Stability Control Tuning and Driver Behavior	239
8.2.3	Further Analysis on Brake Type Effect on Suspension Dynamics, ABS, and RSC Tuning	240
	References	241
	Appendix A MATLAB Script for Sayers' Numerical Tractrix Algorithm	248
	Appendix B MATLAB Script for SGM-A	250
	Appendix C MATLAB Script for SGM-B	252
	Appendix D Analog Sensor Calibration Curves	253
	Appendix E Pin Assignments for DB-37 Cables and NI cRIO	256

List of Figures

Figure 2-1. Body coordinate system [29].	8
Figure 2-2. Tire coordinate system [29].	9
Figure 2-3. Lateral force at tire-ground interface generated by tire sideslip angle [29].	10
Figure 2-4. Friction force vs. longitudinal slip ratio [30].	11
Figure 2-5. Friction ellipse for maximum combined longitudinal and lateral forces generated by a tire [29].	12
Figure 2-6. Common vehicle units and hitches in combination vehicles [32].	14
Figure 2-7. Common types of double-trailer configurations [37].	15
Figure 2-8. Tested trailer configurations.	17
Figure 2-9. Illustration of off-tracking and swept width for an A-train double-trailer configuration.	18
Figure 2-10. Free-body diagram for roll equation of LCV trailer [40].	22
Figure 2-11. Potential jackknifing events: (a) tractor, (b) trailer, and (c) dolly.	23
Figure 2-12. Jackknifed A-train double (Massachusetts State Police, 2019).	24
Figure 2-13. Picture of load-leveling valve for trailer air-spring suspension.	27
Figure 2-14. Plumbing circuit for (a) standard (single load-leveling valve) and (b) balanced (two load-leveling valves) pneumatic suspensions.	28
Figure 2-15. Complete air brake circuit of a tractor-semitrailer [65].	31
Figure 2-16. (a) Illustration of pneumatic drum brake mechanism [66] and (b) internals of drum brake assembly.	32
Figure 2-17. Trailer air disc brake.	32
Figure 3-1. Tractrix integrator [9].	33
Figure 3-2. Bicycle model of 6x4 tractor.	35
Figure 3-3. Bicycle model of tractor-semitrailer in steady-state turn.	36
Figure 3-4. Sayers' numerical tractrix algorithm flowchart for LCVs. Adapted from [12].	38
Figure 3-5. Illustration of SGM-A algorithm for a multi-unit vehicle.	40
Figure 3-6. SMG-A algorithm flowchart.	41
Figure 3-7. Illustration of SGM-B algorithm for a multi-unit vehicle.	43
Figure 3-8. SGM-B algorithm flowchart.	44
Figure 3-9. Simulated steer axle paths for tractrix model comparisons.	45
Figure 3-10. Percent error of each kinematic model's peak off-tracking estimate from the continuous tractrix curves for each of the simulated maneuvers with the 33-ft double trailer configuration.	48
Figure 3-11. Kinematic model comparison for A-double 33-ft trailers axle paths through 50-ft radius J-turn with (a) 4 in (0.1 m) and (b) 3.3 ft (1.0 m) steer axle path resolution.	49
Figure 3-12. Kinematic model comparison for A-double 33-ft trailers off-tracking through 50-ft radius J-turn with (a) 4in (0.1m) and (b) 3.3ft (1.0m) steer axle path step size.	50

Figure 3-13. Kinematic model comparison for A-double 33-ft trailers axle paths through compound 50-ft radius J-turns with (a) 4 in (0.1 m) and (b) 3.3 ft (1.0 m) steer axle path resolution.	51
Figure 3-14. Kinematic model comparison for A-double 33-ft trailers off-tracking through compound 50-ft radius J-turns with (a) 4 in (0.1 m) and (b) 3.3 ft (1.0 m) steer axle path step size.	52
Figure 3-15. Equivalent steady-state model peak off-tracking difference from A-double.	55
Figure 3-16. Axle paths of steady-state off-tracking equivalent configurations through the 50-ft J-turn maneuver.	56
Figure 3-17. Off-tracking of steady-state off-tracking equivalent configurations through the 50-ft J-turn maneuver.	57
Figure 3-18. Axle paths of steady-state off-tracking equivalent configurations through the 50-ft Compound J-turn maneuver.	58
Figure 3-19. Off-tracking of steady-state off-tracking equivalent configurations through the 50-ft Compound J-turn maneuver. <i>Needs to be positive.</i>	58
Figure 3-20. Steer axle path for the 50-ft and 75-ft J-turn maneuvers.	60
Figure 3-21. Steer axle path implemented in TruckSim simulations and field testing.	61
Figure 3-22. Peak off-tracking difference from numerical tractrix models to TruckSim® dynamic model.	62
Figure 3-23. Percent of peak off-tracking difference from numerical tractrix models to TruckSim® dynamic model.	62
Figure 3-24. Illustration of three tested trailer configurations. The trailer lengths are scaled relative to each other to provide a visual representation of their relative lengths.	63
Figure 3-25. Peak off-tracking determined by the numerical tractrix model for each trailer configuration through the four test maneuvers.	64
Figure 3-26. Steer axle and rearmost axle paths of compared trailer configurations through a (a) 50-ft J-turn and (b) 75-ft J-turn.	65
Figure 3-27. Steer axle and rearmost axle paths of compared trailer configurations through a (a) roundabout and (b) right-turn at an intersection.	65
Figure 3-28. Rearmost axle off-tracking through the (a) 50-ft J-turn and (b) 75-ft J-turn.	66
Figure 3-29. Rearmost axle off-tracking through the (a) roundabout and (b) intersection maneuvers.	66
Figure 4-1. Low-speed off-tracking field test procedure. All key steps are shown from test setup to test procedure and data analysis.	69
Figure 4-2. Block diagram illustrating process flow for simulating measurement errors.	70
Figure 4-3. Error in peak off-tracking with respect to tractor-trailer articulation angle measurement error through the 50-ft J-turn.	72
Figure 4-4. Peak off-tracking sensitivity to tractor orientation, tractor-trailer articulation, and trailer-trailer articulation measurements.	73
Figure 4-5. Percent peak off-tracking sensitivity to tractor orientation, tractor-trailer articulation, and trailer-trailer articulation measurements.	74
Figure 4-6. Peak off-tracking sensitivity to steer axle path lateral position error.	75
Figure 4-7. Peak off-tracking sensitivity to steer axle path lateral position error.	75

Figure 4-8. Error in peak off-tracking with respect to tractor-trailer articulation angle temporal error through the 75-ft J-turn.....	77
Figure 4-9. Peak off-tracking sensitivity to tractor-trailer articulation and trailer-trailer articulation temporal errors.	78
Figure 4-10. Percent peak off-tracking sensitivity to tractor-trailer articulation and trailer-trailer articulation temporal errors.	78
Figure 4-11. Peak off-tracking sensitivity to steer axle path temporal error.	79
Figure 4-12. Percent peak off-tracking sensitivity to steer axle path temporal error.....	79
Figure 4-13. (a) GoPro installed on the front bumper as live-feed camera. (b) dashboard monitor for live-feed camera.....	81
Figure 4-14. Illustration of the difference in the laid camera path and the target steer axle path through a turn.	82
Figure 4-15. Schematic of steer axle position error due to offset of the camera path from the camera's FOV centerline. Tractor is represented as single-track bicycle model and dimensions are exaggerated for clarity.	83
Figure 4-16. Schematic of tractor orientation error due to offset of the camera path from the camera's FOV centerline. Tractor is represented as single-track bicycle model and dimensions are exaggerated for clarity.	84
Figure 4-17. (a) Bosch GLM400C laser distance meter mounted to the rotary table used for positioning maneuver waypoints. (b) Schematic of locating maneuver waypoints.....	86
Figure 4-18. Illustration of articulation angles measured by multi-segment LiDAR.	87
Figure 4-19. LeddarTech™ IS16 segment field-of-views.	87
Figure 4-20. Front trailer wall with black cloth used to increase surface diffuse reflectivity to improve articulation angle measurement accuracy.....	90
Figure 4-21. Illustration of different surface reflectivity patterns [19].	91
Figure 4-22. LiDAR sensor segment-specific calibration setup.....	92
Figure 4-23. Segment-specific distance measurement error.....	93
Figure 4-24. Angle measurement error comparison of raw and calibrated LiDAR data. Segments 1 & 16 demonstrated poor performance and were not used directly for articulation angle measurements.	94
Figure 4-25. (a) Illustration of top-down view for jig geometry. (b) Jig staged at trailer for step 2 of LiDAR calibration process.....	95
Figure 4-26. Articulation angle LiDAR choosing appropriate segments for the surface walls for determining the articulation angle. (a) All segments observe the front wall. (b) Some segments observe the trailer corner and are omitted. (c) Both front wall and side wall are observed.	96
Figure 4-27. Jig misalignment of (a) 0° reference and (b) LiDAR sensor orientation relative to swingarm (AC).	97
Figure 4-28. Comparison between corrected (adjusted) and unadjusted measurements for 2 nd calibration step.	98
Figure 4-29. Curve-fits for converting LiDAR articulation angle measurement to actual articulation angle.	99

Figure 4-30. Geometry of trailer-trailer articulation used to determine trailer corners with respect to LiDAR frames.....	101
Figure 4-31. Graphs show the LiDAR segment edges and the observed trailer's driver (left) and passenger (right) side corners as the vehicle travels through the tight-turn maneuver.	101
Figure 4-32. LiDAR mount on the back of a front trailer.	102
Figure 4-33. Articulation angle LiDAR mounting offset as determined by the estimated zero articulation angle measured near the beginning of each maneuver. X marks the mean.	103
Figure 4-34. Steer axle speed as measured by GPS receiver. Raw data is shown as scatter of grey dots, filtered data with moving average is shown as smooth curves.	107
Figure 4-35. GPS position data cloud and target steer axle path for each of the four tested maneuvers. .	109
Figure 4-36. GPS position error from target path.....	110
Figure 4-37. Steer axle path and measured rearmost axle paths for the (a) 50-ft J-turn and (b) 75-ft J-turn. Locations of peak off-tracking are marked by solid lines of appropriate color.....	113
Figure 4-38. Steer axle path and measured rearmost axle paths for the (a) roundabout and (b) intersection maneuvers. Locations of peak off-tracking are marked by solid lines of appropriate color.	113
Figure 4-39. Peak off-tracking for each trailer configuration through the (a) 50-ft J-turn, (b) 75-ft J-turn, (c) Roundabout, and (d) Intersection maneuvers.....	117
Figure 4-40. Difference in peak off-tracking from field test measurements to tractrix model.	117
Figure 5-1. Test vehicle.	119
Figure 5-2. Illustration of sensor type, count, and locations on the double-trailer truck. The DAQ is not shown in this illustration, but it was placed in the back of the tractor cab.	121
Figure 5-3. Illustration of the tractor with photoelectric sensor and reflective target to flag the beginning of a maneuver for data processing and synchronization.....	122
Figure 5-4. Pictures of installed string potentiometers: (a) straight-rail suspension, (b) steering wheel angle, (c) suspension string-potentiometer calibration jig.	124
Figure 5-5. (a) Straight-rail triaxial and single-axis accelerometers on extruded aluminum structure mount, (b) drop-frame triaxial and single-axis accelerometers, (c) tractor triaxial accelerometer mounted inside waterproof case on aluminum extension from tractor frame.	126
Figure 5-6. Schematic for calibrating the pressure transducers.....	128
Figure 5-7. Example of pressure transducer calibration.....	128
Figure 5-8. Test setup for determining the force ratio from the outrigger wheel to the outrigger load cell.	129
Figure 5-9. Example plots of the measured force at the outrigger wheel versus the force measured by the load-cell inside the outrigger for a straight-rail trailer outrigger arm.	130
Figure 5-10. Schematic representation of the power provided by the batteries and distributed by the junction boxes.	134
Figure 5-11. Junction box external connections. (a) Tractor sensor connectors and GPS power. (b) Tractor DB-37 output and DB-37 input from front trailer, LiDAR power port and connectors for flag LEDs on the tractor. (c) Trailer analog sensor connectors and power ports. (d) Trailer flag LED connectors and cutout for DB-37.....	135

Figure 5-12. Tractor junction box internals.	136
Figure 5-13. Trailer junction box internals.	137
Figure 5-14. Sensor connector wired signals and supply voltage on the trailer JB as viewed from the outside. The code names for each sensor/relay are listed next to the respective connections (i.e.: LC-D = driver side load cell).....	138
Figure 5-15. National Instruments CompactRIO system used for data acquisition.	139
Figure 5-16. LabVIEW interface. The window on the left is the front panel of the block diagram code on the right. Only a very small portion of these programs is displayed within this screenshot.	140
Figure 5-17. Screenshot of Leddar Configurator™ software for real-time display of sensor data.....	141
Figure 5-18. GoPro mounting locations.	142
Figure 5-19. Example of signal noise from front trailer triaxial accelerometer signal for lateral acceleration.	144
Figure 5-20. GPS paths of preliminary testing maneuvers (Commonwealth of Virginia, Maxar Technologies, 2022).....	146
Figure 5-21. Steering robot servomotor mounted on the steering column within the engine bay.	147
Figure 5-22. Operation flow for robot-steered maneuver.	148
Figure 5-23. Steering and braking robot control box. The E-stop, deadman switch, and photoelectric sensor are also shown.	150
Figure 5-24. Steering wheel angle comparison of driver-steered (top) and robot-steered (bottom) J-turn maneuver during roll-stability testing at TRC.	151
Figure 5-25. Braking robot pneumatic actuator installed in tractor cab.....	152
Figure 5-26. Close-up view of the installed brake pedal extension.	153
Figure 5-27. Pneumatic supply to the braking robot.	154
Figure 5-28. Pneumatic plumbing arrangement of braking robot system.....	155
Figure 5-29. Braking robot pneumatic time delay.....	155
Figure 5-30. Tractor brake pressure as measured from different brake pedal actuator stroke reducers....	156
Figure 5-31. Tractor drive axle brake pressure on right side for full and partial braking J-turn maneuver.	157
Figure 5-32. CVeSS outrigger assembly.	158
Figure 5-33. Anti-jackknifing system configuration.	159
Figure 5-34. Yaw reinforcement structure: (a) CAD model, and (b) installed in trailer.	160
Figure 5-35. Exploded view of the pintle hook assembly installed on trailer front wall and yaw reinforcements.	160
Figure 5-36. 33-ft straight-rail roll reinforcement structure: (a) CAD model of “H” shaped reinforcement structure, (b) CAD model of the top mounting plates, (c) “H” shaped reinforcement underneath the trailer, (d) top mounting plates on the trailer floor to hold the reinforcement structure.	161
Figure 5-37. Installed load frame on 33-ft straight-rail trailer. (a) Back of load frame, (b) sandbags, (c) truss.	162

Figure 6-1. J-turn maneuver path layout.	164
Figure 6-2. Robot J-turn steering profile.	165
Figure 6-3. DLC path layout.	165
Figure 6-4. Steering wheel angle changes with speed for DLC maneuver. ESC and RSC off.	166
Figure 6-5. 0.25-Hz SWD maneuver steering wheel angle profile.	167
Figure 6-6. 0.25-Hz SWD maneuver path in stock condition (RSC and ESC off).	167
Figure 6-7. Aerial view of TRC Vehicle Dynamics Area (Maxar Technologies, State of Ohio, 2022). ..	171
Figure 6-8. Example tractor ESC intervention and resultant brake pressures during a DLC maneuver. ..	173
Figure 6-9. Example front and rear trailer RSC interventions and resultant brake pressures during a DLC maneuver.....	174
Figure 6-10. Front trailer right side outrigger wheel contact force for the driver J-turn maneuver.....	176
Figure 6-11. Front trailer lateral acceleration during driver J-turn maneuver in stock condition.	176
Figure 6-12. Speed reduction during driver J-turn maneuver for each stability control system implementation.....	177
Figure 6-13. Peak articulation angles during driver J-turn maneuver.	178
Figure 6-14. Example comparison of articulation angle during driver J-turn maneuver for different stability control system implementations at same entry speed (34 mph).	179
Figure 6-15. Front trailer right side outrigger wheel contact force for the robot J-turn maneuver.....	180
Figure 6-16. Speed reduction during robot J-turn maneuver for each stability control system implementation.....	181
Figure 6-17. ESC intervention duration during the robot J-turn maneuver for ESC only and RSC+ESC configurations.....	181
Figure 6-18 . Peak articulation angles during robot J-turn maneuver.	182
Figure 6-19. Example comparison of articulation angle during robot J-turn maneuver for different stability control system implementations at similar speeds (32 – 33 mph).....	183
Figure 6-20. Tractor steer axle path through the robot J-turn maneuver with RSC+ESC.....	184
Figure 6-21. Steering wheel angle examples in DLC maneuver at 40 mph.	186
Figure 6-22. Peak steering wheel angles in turn 1, 2, and 3 of the DLC maneuver.	187
Figure 6-23. Speed drop through DLC maneuver.	188
Figure 6-24. Peak absolute lateral accelerations in turn 2 and turn 3 of the DLC maneuver.	189
Figure 6-25. Peak articulation angles through the DLC maneuver.	190
Figure 6-26. ESC intervention duration in SWD maneuver.	193
Figure 6-27. Speed drop through the SWD maneuver.	194
Figure 6-28. Example of tractor speed through the SWD maneuver at 60% steering.	194
Figure 6-29. Peak articulation angles in SWD maneuver.	195
Figure 6-30. Rollover speed and steering percentage for all tested maneuvers and stability control configurations.....	197

Figure 6-31. Roll-stability improvement trends by maneuver type for each stability control system.....	198
Figure 6-32. Overall roll-stability improvement for each stability control configuration.	198
Figure 6-33. Average deceleration while stability control systems are intervening.	199
Figure 7-1. Brake type at each axle for testing trailer brake type performance.	202
Figure 7-2. LiDAR was used to measure the stopped distance from the reference end target. The reference end target was positioned at a known distance from the starting gate's spatial trigger.	203
Figure 7-3. (a) Satellite image of vehicle dynamics area (VDA) and north loop (NL) used for testing. (Maxar Technologies, State of Ohio, 2022). (b) Test scheme of alternating speeds, measuring brake temperatures, and cool down laps between trials and test speed sets.	203
Figure 7-4. Comparison of measured braking distance between disc and drum brakes at different speeds.	205
Figure 7-5. Expected braking distance reduction in terms of average passenger car length by switching trailer brakes from drum to disc according to 2nd-order polynomial curve-fits from test data.	206
Figure 7-6. Average deceleration for entire braking maneuver for each tested entry speed.	207
Figure 7-7. Pneumatic transport delay as percentage of total brake time.	208
Figure 7-8. Pneumatic transport delay as percentage of total braking distance.	208
Figure 7-9. Average deceleration at full braking for straight-line testing.....	209
Figure 7-10. Range of braking distance of among the three trials for each test speed.	210
Figure 7-11. Comparison of braking distance at the 65-mph test speed set for disc and drum brakes.....	211
Figure 7-12. Brake temperatures measured after each test speed test.	212
Figure 7-13. Total average axle brake temperatures for all speeds excluding 20-mph.	212
Figure 7-14. Comparison of front and rear trailer brake pressures during a 65-mph straight-line braking test.	214
Figure 7-15. Average ABS activation duration of the three trials for each test speed set during straight-line braking.	215
Figure 7-16. Dolly and trailers free-body diagrams during steady braking and constant deceleration.	216
Figure 7-17. Free-body diagrams of trailer suspension trailing arm for an axle with (a) drum brakes and (b) disc brakes.	218
Figure 7-18. Front and rear trailer air spring pressures for drum and disc brakes during 65-mph straight-line braking.	221
Figure 7-19. Tested brake delay vs entry speed for the braking-in-turn maneuver with full braking (the bubble areas are proportional to the peak outrigger contact forces).	222
Figure 7-20. Summary of outrigger contacts for full-braking J-turn maneuver.	223
Figure 7-21. ABS activation time delay from brake pedal application during full-braking J-turn tests. ..	224
Figure 7-22. Tested brake delay vs entry speed for the braking-in-turn maneuver with partial braking (the bubble areas are proportional to the outrigger contact forces).	225
Figure 7-23. Summary of outrigger contacts for partial-braking J-turn maneuver.	226

Figure 7-24. Lateral acceleration comparison of drum and disc trailer brakes for partial-braking J-turn maneuver with 1.25 sec brake delay and 35 mph entry speed.....	227
Figure 7-25. ABS activation time delay from brake pedal application during partial-braking J-turn tests.	228
Figure 7-26. Peak outrigger wheel contact force for Robot J-turn with RSC on for both tested trailer brake types	229
Figure 7-27. Peak outrigger wheel contact force for Driver J-turn with RSC on for both tested trailer brake types.	230
Figure 7-28. Box and whisker plot of average deceleration while RSC is active on either or both trailers for all maneuvers.....	233
Figure 7-29. Lateral acceleration comparison of drum and disc trailer brakes for 0.25 Hz SWD maneuver for 45 mph entry speed and 60% steering.	234

List of Tables

Table 2-1. Severity of large truck rollovers in 2018 [1].....	20
Table 3-1. Wheelbase and hitch length of simulated 33-ft A-double with sleeper cab tractor.	45
Table 3-2. Equivalent tractor-trailer dimensions for steady-state off-tracking that matches 33-ft A-double with sleeper-cab tractor.....	54
Table 3-3. Test tractor, trailers, and dolly wheelbases and hitch lengths.	63
Table 4-1. List of peak off-tracking sensitivity indices for each parameter of interest for the test vehicle configurations and maneuvers.....	80
Table 4-2. LeddarTech IS16 segment centerline and outer edge angles from the sensor's field of view center.	88
Table 4-3. Calibrated LiDAR distance offset for each segment.	93
Table 4-4. Articulation angle offset correction added to the articulation angle measurements to remove error from LiDAR mounting orientation.	104
Table 4-5. Maximum position uncertainty along the target steer axle path via integrating filtered GPS speed.	107
Table 4-6. Estimated uncertainty for each parameter.....	114
Table 4-7. Peak off-tracking uncertainty due to uncertainty in each parameter and peak off-tracking's sensitivity.....	115
Table 4-8. Average sensitivity index, parameter uncertainty, and peak off-tracking uncertainty with respect to each parameter for the tested maneuvers and trailer configurations. Color coding is specific to each row.	115
Table 5-1. Test vehicle specifications.	120
Table 5-2. A comprehensive list of installed sensors for the roll-stability and brake-type studies.	122
Table 5-3. Steer angle string potentiometer calibration test results. A negative steering wheel angle turns the vehicle right and a positive steering wheel angle turns the vehicle left.....	125
Table 5-4. Lateral acceleration just before rollover as determined by triaxial and single-axis accelerometers for a range of low-pass filter cutoff frequencies.....	132
Table 5-5. Low-pass filter cutoff frequency by measurement.....	132
Table 5-6. Junction box wire color coding.....	135
Table 5-7. Measurement sampling rates.....	138
Table 5-8. Measurement uncertainty (95%) in engineering units caused by electrical noise for each sensor type.....	143
Table 5-9. Total measurement uncertainty by sensor type in engineering units.	145
Table 5-10. Old and new battery pack specifications for the steering robot.	149
Table 5-11. PID gains for the steering robot servomotor for each maneuver.	151
Table 6-1. Decision matrix for determining rollover threshold.....	169
Table 6-2. Summary of test sessions for evaluating stability control systems.	172

Table 6-3. ESC activation analysis for RSC+ESC configuration during robot J-turn at 35 mph entry speed.	180
Table 6-4. Peak outrigger contact forces for DLC maneuver.	185
Table 6-5. Outrigger contact forces in SWD maneuver.	192
Table 6-6. Roll-stability test maneuver categorization.....	196
Table 6-7. Stability control system effect on articulation angles.	200
Table 7-1. Brake chamber pressure time delay from brake pedal initiation to start of brake chamber pressure rise.	207
Table 7-2. Peak outrigger wheel contact forces for DLC with RSC on for both tested trailer brake types.	231
Table 7-3. Peak outrigger wheel contact forces for SWD with RSC on for both tested trailer brake types.	232
Table 7-4. Rollover speed and steering percentage comparison between drum and disc brake systems for the four tested maneuvers with RSC on & ESC off.	232
Table 7-5. Rollover speed and steering percentage comparison between drum and disc brake systems for the four tested maneuvers with RSC on & ESC on.	235
Table D-1. Calibration curve-fits for suspension travel string potentiometers.	253
Table D-2. Accelerometer calibration data provided by the manufacturer, PCB Piezotronics®.	253
Table D-3. Curve-fits for all pressure transducers, operational and spare.	254
Table D-4. Sensitivity analysis of the load cells that are used in the outrigger to detect the outrigger contact force.	255

Chapter 1

Introduction

This chapter outlines the motivation, objectives, approach, and contributions of the research described in this dissertation. The chapter concludes with a brief outline describing the topics covered in each chapter.

1.1 Motivation

Large trucks (gross weight rating $\geq 10,000$ lb.) are essential to distribute goods regionally and across nations. In 2018, there were 10.3 million and 2.9 million registered single-unit trucks and tractor-trailers, respectively, in the United States. Combined, these make up 4.83% of the 273.6 million registered vehicles [1]. Although large trucks constitute a small number of registered vehicles on U.S. roads, large trucks tend to travel more miles than passenger vehicles. Large trucks traveled 9.4% of all vehicle miles traveled in 2018 [1]. Compared to passenger vehicles, large trucks are significantly longer, heavier, and have higher centers-of-gravity. Their dimensions impede maneuverability within spatially constrained environments common to urban roads and around loading docks to transfer goods. Large truck drivers, fleet operators, and road design engineers must consider the physical limitations to maneuver these vehicles and provide sufficient space within their routes. One metric commonly used to assess the space needed for maneuvering a long vehicle is off-tracking. Off-tracking is the distance between the path of the rearmost axle relative to the lead vehicle axle. Several studies have been conducted for experimentally measuring off-tracking in long vehicles, both on- and off-highway. These studies, however, typically only evaluated a long straight truck or tractor semitrailer. Some of the methods did not provide precise and accurate measurements [2], were not transparent about the uncertainty in their measurements [2,4], or only used scale models [8]. The studies presented herein were designed to bridge the gap in literature of evaluating off-tracking of long combination vehicles (LCVs) – heavy vehicles with towing multiple trailers – in both simulation and field testing while clearly detailing uncertainty in test procedure and measurements.

Advances in vehicle dynamics analysis from the 1970s to date have generated many different methods for modeling vehicle motion. At low-speed, the vehicle's inertia, compliance, and tire forces are insignificant compared to the vehicle's geometry when estimating motion of trailing axles with respect to the steer axle. For low-speed maneuvers, tractrix curves are an elegant and widely accepted method for estimating trailing axle paths for a given steer axle path and vehicle geometry [2,3,4,5,6,9,11,12,23]. For LCVs, however, calculating tractrix curves is non-trivial. A fast numerical model representative of tractrix curves can be beneficial for many applications including road design and path planning for autonomous vehicles. In the 1980s, a numerical tractrix algorithm was suggested for LCVs with any number of vehicle units and

wheelbase lengths [12]. However, its accuracy is largely impacted by the size and regularity of distance between successive data points along the input steer axle path. Furthermore, the input path is required to be in the form of arc lengths and curvatures. In a physical study, the input path may be measured in cartesian coordinates – such as GPS coordinates – and if the data is noisy, numerical challenges can arise when converting the data to arc length and curvature. Smoothing techniques can reduce some numerical limitations but increase computational cost and create an additional layer of uncertainty. In response to these challenges, a simple numerical model is needed to accurately estimate LCV axle paths for a given input steer axle path provided as a series cartesian coordinates that may be irregular and have large and/or varying distances between successive path points.

Unlike low-speed maneuvers, the vehicle's inertia, compliance, and tire forces significantly influence LCVs motion during moderate- to high-speed maneuvers. Compared with passenger vehicles, large trucks have significantly more mass and higher center-of-gravity that often lead to undesirable dynamics. The articulation between vehicle units of an LCV can further complicate the vehicle's stability. These attributes impact the driver's ability to control the vehicle and can result in an accident. Crashes involving large trucks often pose serious threat to life and property. In 2018, 4,862 large trucks were involved in fatal crashes, 59,933 in injury crashes, and 111,291 in towaway crashes [1]. 23% of those fatal crashes are estimated to be caused by critical pre-crash events of the large trucks' own movement or loss of control [1]. Since the 1990s and 2000s, electro-mechanical systems have been used to reduce LCVs' inherent dynamic instabilities. These systems include Electronic-Stability Control (ESC) on tractors and Roll-Stability Control (RSC) on trailers. Both systems typically apply differential braking to counter yaw and roll dynamics and are well documented at reducing or mitigating undesirable dynamics for the common tractor-semitrailer. However, there is a gap in public data of ESC and RSC performance on 33-ft double trailers. Current federal regulation limits double trailers to 28-ft in length, but many states and turnpikes permit longer vehicle combinations. Cargo transported by package carriers is often limited by volume rather than by gross weight rating of the vehicle. Additionally, straight-rail trailers are easier to load and unload, and a 33-ft straight-rail trailer has about the same cargo-carrying volume as the common 28-ft drop-frame trailer. Thus, some package carriers are pushing federal legislation to accept longer vehicle configurations, and an understanding of RSC and ESC performance on 33-ft doubles will be required. A tractor pulling 33-ft A-double trailers is tested through multiple maneuvers designed to introduce and evaluate rollover propensity. Different combinations of stability control systems are evaluated by their performance at reducing rollover propensity.

Trailer brake type can also impact the overall dynamic performance of LCVs. There are two types of friction brakes: drum and disc. Drum brakes are more commonly found on LCVs than disc brakes in the United States. Disc brakes have been shown throughout literature to outperform drum brakes with respect to stopping distance and resistance to brake fade [70,71,72,73,74,75,76]; however, tests specific to 33-ft double trailers have not been conducted. The trailer brakes on their own are of interest – rather than including the tractor – because some package carriers own their trailers and higher contract tractors and drivers. Thus, the trailer fleet operators do not have direct control of the tractor’s equipment. Additionally, there is no public record of how different types of trailer brakes impact differential-braking trailer RSC performance. To address these topics, physical tests were conducted with the 33-ft double trailer configuration for straight-line braking performance and RSC performance at reducing rollover propensity for the two brake types.

1.2 Objectives

The primary objectives for the three studies are as follows:

Low-Speed Off-tracking of Common U.S. On-highway LCVs

1. To compare 33-ft double off-tracking to 28-ft double and 53-ft single via simulation & full-scale tests of real-world maneuvers;
2. To design test procedure that uses existing CVeSS equipment;
3. To provide transparency in test procedure and measurement uncertainty;
4. To create numerical model of low-speed tractor-trailer motion that may be used as part of field test data analysis and simulation study; and

33-ft A-Double Rollover Mitigation by Differential Braking Stability Control Systems

5. To conduct physical tests to evaluate rollover propensity of fully-loaded 33-ft A-double straight-rail trailers with the following stability control implementations: stock (none), tractor ESC, trailer RSC, and RSC+ESC;
6. To highlight differences between stability systems at reducing trailer rollover propensity; and

Trailer Brake Type Performance Evaluation on 33-ft A-Double Trailers

7. To test and compare trailer disc and drum brakes with respect to straight-line braking distance, influence on trailer RSC performance at rollover mitigation, and rollover propensity during combined braking and turning;
8. To investigate trailer brake type effect on ABS intervention and suspension dynamics.

1.3 Approach

All three studies – low-speed off-tracking, rollover propensity with stability control systems, and trailer brake type performance – are conducted by physical testing of class 8 tractor-trailer vehicles. The key steps for each study are detailed below.

Low-Speed Off-Tracking of Common U.S. On-highway LCVs

- A thorough literature review is conducted on physical tests, mathematical models, and graphical methods designed to identify LCV axle paths and off-tracking through a maneuver.
- Two new numerical algorithms that model LCV axle trajectories through a given maneuver and initial conditions are developed and compared to an existing tractrix curve algorithm. The new algorithms' primary benefits include reduced computational expense and easier implementation as part of processing field test data.
- The two new algorithms are combined as a single model that is inherently less sensitive to the input path resolution compared to other numerical tractrix models. This model is then used to evaluate off-tracking of different trailer configurations – both real and hypothetical – through a variety of maneuvers.
- Field tests are conducted to physically compare off-tracking of common LCV configurations in real-world maneuvers, detail uncertainty propagation of measurements, and provide a reference for comparison to model simulation.

33-ft A-Double Rollover Mitigation by Differential Braking Stability Control Systems

- Physical tests are conducted on a test track with a professional driver. Maneuvers were chosen to simulate real-world driving behavior on highways and excite key vehicle dynamics of interest.
- The 33-ft A-double straight-rail test trailers and sleeper-cab test tractor are modified and outfitted with extensive mechanical and electrical equipment including:
 - Load frames to control the center-of-gravity (CG) height and total payload;
 - Outriggers to measure impact forces while preventing a complete trailer rollover;
 - Anti-jackknifing system to prevent jackknifing between any two vehicle units without hindering vehicle unit yaw motion within range of interest during test maneuvers;
 - An extensive suite of analog sensors to measure key dynamics such as body acceleration, brake chamber and suspension airbag pressures, suspension jounce, stability control intervention timing, vehicle speed, tractor steering, etc.;
 - Multi-segment LiDAR between vehicle units to measure articulation angles;

- Several video cameras to validate analog sensor data such as outrigger contacts and to identify dynamics not captured by sensors such as wheel lift; and
 - Designed and built-in-house electrical junction boxes to power sensors and route their signals to the data acquisition unit.
- Analog sensor and GPS data are collected simultaneously by the data acquisition unit. LiDAR data was recorded by separate software applications on the data acquisition laptop. LiDAR data is synchronized with the analog sensor and GPS data in post-processing.
- Video data is synchronized with sensor data by way of illuminating an LED within each camera's field-of-view during each test trial. Once the first trial of a dataset is identified with the video and sensor data, later trials are synchronized by the time difference from the first trial.
- A custom CVeSS designed and built steering robot from past studies was upfitted and retuned to perform test maneuvers that required repeatable and accurate steering profiles unattainable by a professional driver.
- Tested maneuvers are selected from standard test maneuvers by the National Highway Traffic Safety Administration (NHTSA) or some variation determined by past CVeSS studies most adept to capture dynamics of interest for the test objectives.
- Tires are regularly replaced to ensure fair comparison among different test days.

Trailer Brake Type Performance Evaluation on 33-ft A-Double Trailers

- The same 33-ft A-double straight-rail trailers and test equipment from the roll-stability study are used for the brake type performance tests.
- A braking robot, controlled by the steering robot controller, is designed and fabricated to press and hold the brake pedal. Spatial triggering is attained by a photoelectric sensor onboard the tractor and a reflective target staged at the maneuver starting gate. The robot is designed to provide either full braking or partial braking. Full-braking uses the entire brake pedal stroke whereas partial braking limits the actuator (and thus brake pedal) stroke.
- Straight-line braking distance and rollover in braking-in-turn tests are conducted with the tractor and trailers' original drum brakes after the roll stability tests. The brakes are thoroughly inspected and serviced by certified a technician before the tests.
- Once drum brake testing is completed, the trailer axles are swapped with the ones with new disc brakes. The original drum brakes on the tractor are kept unchanged.
- Straight-line braking, rollover in braking-in-turn, and roll-stability tests with RSC and RSC+ESC are repeated with the trailers equipped with disc brakes.
- Brake temperatures are regularly measured using an infrared thermometer throughout all testing.

1.4 Contributions

Low-Speed Off-Tracking of Common U.S. On-highway LCVs

- Creation of a new numerical tractrix model that is computationally inexpensive and is largely unaffected by the input path resolution and regularity between successive points. The new numerical model also accepts a path input in cartesian coordinates, a unique feature compared to other similar models.
- The new model is used to compare common LCV trailer configurations in a variety of maneuvers.
- The new model is used to briefly investigate if an LCV can be modeled as a single vehicle unit to result the same off-tracking within a maneuver.
- Physical tests to measure off-tracking of common LCVs through real-world maneuvers are conducted. Uncertainty within the test procedure and measurements is highly detailed to provide transparency unseen in literature.
- Sensitivity analysis is conducted to define how each procedure step's and measurement's uncertainties directly influenced peak off-tracking uncertainty for each tested trailer configuration and maneuver.

33-ft A-Double Rollover Mitigation by Differential Braking Stability Control Systems

- Trailer rollover propensity of heavily-loaded 33-ft straight-rail double trailers is determined for the following stability control system implementations through multiple driver- and robot-steered maneuvers: stock (none), tractor electronic stability control (ESC), trailer roll-stability control (RSC), and RSC+ESC.
- The tested stability control systems are ranked in the order of trailer roll-stability improvement for the tested trailer configuration.
- Stability control influence on vehicle directional dynamics is also investigated.

Trailer Brake Type Performance Evaluation on 33-ft A-Double Trailers

- Trailer brake types are accurately compared on 33-ft A-doubles in straight-line braking tests. A spatial trigger with robotic braking is integrated into the test to accurately measure braking distance with the available on-board sensors.
- The effect of brake type (drum vs. disc) on ABS intervention and trailer suspension jounce is investigated.

- The effect of brake type on the trailer RSC performance is evaluated experimentally for various types of maneuvers experienced during highway driving and emergency steering maneuvers.

1.5 Outline

- Chapter 1 provides the motivation for the studies and lists the objectives, approach, and contributions of the research.
- Chapter 2 provides the background and technical review necessary to facilitate the studies.
- Chapter 3 provides a brief literature review on models for low-speed motion of LCVs, introduces new numerical tractrix algorithms, and implements the models to compare different trailer configurations through a variety of maneuvers.
- Chapter 4 details the test procedure, instrumentation, and data analysis for physical tests on common LCVs' off-tracking through real-world maneuvers. Propagation of measurement and procedural uncertainty into the results is highlighted.
- Chapter 5 details the test equipment for high-speed dynamics tests conducted to evaluate stability control systems and brake type performance on 33-ft double trailers. Designed-in-house instrumentation and steering & braking robotics for the tested equipment are highlighted. Measurement uncertainties are discussed.
- Chapter 6 details the test procedure, results, and analysis of rollover propensity of the 33-ft double trailers tested with different stability control system configurations.
- Chapter 7 details the test procedure, results, and analysis of trailer brake type performance testing on the 33-ft double trailers.
- Chapter 8 summarizes conclusions on the three studies and provides recommendations of future work.

Chapter 2

Background and Literature Review

This chapter details class-8 heavy truck equipment as well as stability concerns of long combination vehicles (LCV). First, key vehicle dynamics terminology is defined and a detailed background on common types of on-highway LCVs in the U.S. is provided. Second, off-tracking and uncertainty analysis are defined. Then, roll- and yaw-stability concerns of LCVs in dynamically aggressive maneuvers are detailed as well as common electro-mechanical systems designed to counter undesirable dynamics. Lastly, LCV braking systems are highlighted, and mechanical operations of drum and disc brakes are described.

2.1 Terminology

Throughout this document, many terms will be used frequently to specify certain LCV dynamics. This briefing is intended to provide a quick reference for those not regularly participating within the vehicle dynamics community. First, the coordinate system for a vehicle unit is defined by the ISO 8855 standard illustrated in [29,33]. Although this is shown for a passenger vehicle, the same body-specific coordinate system would be attached to each vehicle unit's (tractor, trailer, etc.) center-of-gravity (CG). Vehicle dynamics described throughout this document are assumed to follow this coordinate system unless stated otherwise for specific cases such as comparing sprung (trailer body and cargo) and unsprung (suspension and tires) masses. As shown in the figure, longitudinal dynamics occur along the x-axis of a vehicle unit, lateral dynamics occur along the y-axis, and vertical dynamics occur along the z-axis. Rotation about each of the x-, y-, and z-axis is called roll (ϕ), pitch (θ), and yaw (ψ), respectively. Positive roll indicates leaning to the right, positive pitch indicates tilting forward, and positive yaw indicates rotating to the left.

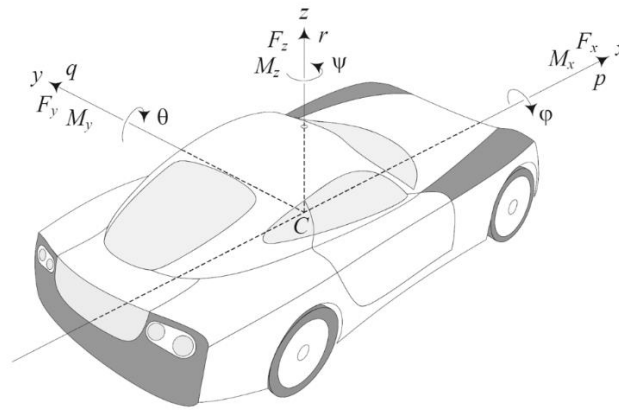


Figure 2-1. Body coordinate system [29].

A critical component to vehicle dynamic performance is the tire-road interface. This interface is called the contact patch, and before determining forces at the contact patch, a coordinate system is first declared for the tire as defined by Figure 2-2 [29]. Notice that the origin lies on the ground xy-plane at the center of tire-ground interface. The x-axis direction is along the tire and wheel's headed direction, but the actual direction of travel may be along the velocity vector v that is oriented at the sideslip angle α from the x-axis. Similarly, the tire may be tilted. Because the coordinate system xy-plane is the ground plane, the z-axis is perpendicular to the ground. The amount that the tire and wheel are tilted with respect to the z-axis is the camber angle γ .

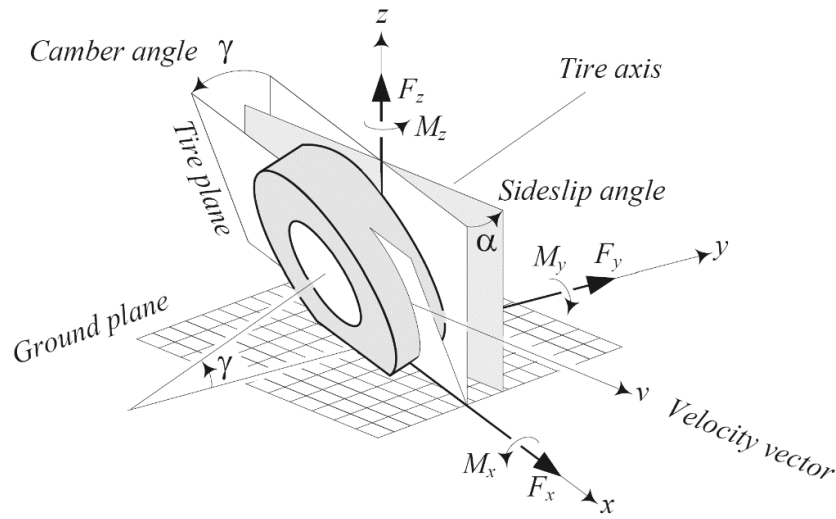


Figure 2-2. Tire coordinate system [29].

During a turn, the sideslip (lateral slip) angle (α) generates a lateral force at the contact patch. The ratio of the lateral force with respect to the sideslip angle is the cornering stiffness of the tire. Figure 2-3 plots the lateral force with respect to tire sideslip angle for a variety of normal loads on the example tire. Notice there is an approximately linear relationship between lateral force and sideslip angle for angles less than $\sim 5^\circ$. For this region and under constant vertical load, the cornering stiffness of the tire is approximately constant, and this value is typically used for first-order modeling and simulation of vehicle dynamics. In general, at very low speeds a tire's heading is along its velocity vector that results zero or negligible sideslip ($\alpha \approx 0$). In this case, the vehicle's motion may be estimated by the tires' headings as defined geometrically by a kinematic model rather than by a summation of forces like that in a dynamic model [6,29,31]. This is of particular interest in the low-speed maneuverability study and will be discussed further in Chapter 3.

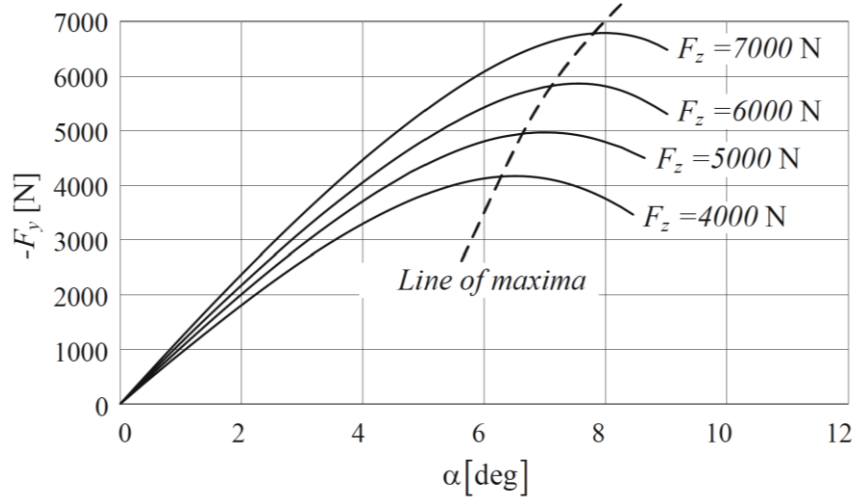


Figure 2-3. Lateral force at tire-ground interface generated by tire sideslip angle [29].

While accelerating or braking, shear forces develop longitudinally along the contact patch. Due to tire deformation under its supported vertical load and compliance under tractive or braking torque, the shear forces are not constant throughout the entire contact patch. These forces develop nonlinearly and depend on a multitude of factors. Under tractive torque (accelerating), the tire tread ahead of the contact patch is compressed and then, once rotated to behind the contact patch, expands. While braking, the tire tread experiences the opposite trend such that it is first stretched in front of the contact patch then compressed afterward. These transitions between compression and expansion along the tread surface are an influential factor that cause the nonlinear behavior of tire-ground forces. These deformations along with tire compression from the supported vertical load create a difference in the tire's tangential speed and the axle's speed with respect to the road. The ratio of these speeds is defined as the slip ratio or longitudinal slip. The slip ratio λ increases as the absolute difference of speeds increase as illustrated by Equation (2-1) where v is the axle (vehicle) forward velocity, Ω is the tire's angular velocity, and r_e is the tire's effective (loaded) radius. The traction coefficient – the ratio of longitudinal (forward) force with respect to the supported load (vertical force) – is a nonlinear function of the slip ratio. Presented in Figure 2-4, the friction force first increases with slip ratio approximately linearly, then the rate slows down until it reaches a peak, and then the traction coefficient reduces until the tire locks-up and slides across the road surface (100% longitudinal slip).

$$\lambda = \begin{cases} 1 - \frac{\Omega r_e}{v} & : v \geq \Omega r_e \text{ (Braking)} \\ 1 - \frac{v}{\Omega r_e} & : v < \Omega r_e \text{ (Accelerating)} \end{cases} \quad (2-1)$$

For wet or snowy roads that are more slick than dry asphalt, the traction coefficient is reduced, but the general shape of the traction coefficient as a function of longitudinal slip remains similar. To maximize the amount of tractive or braking force, some longitudinal slip must exist, but not too much. Anti-lock brake systems (ABS) modulate brake pressure to control tire slip in attempt to maximize the braking force. This is important and further discussed in the brake type performance studies in Chapter 7.

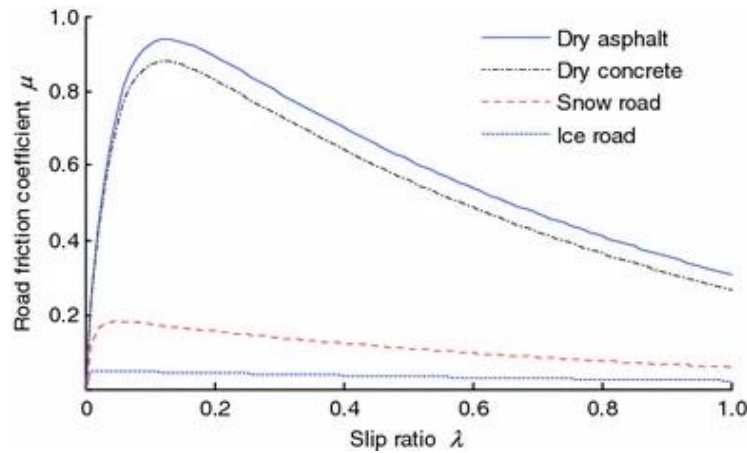


Figure 2-4. Friction force vs. longitudinal slip ratio [30]

As one may infer from Figure 2-3 and Figure 2-4, a tire can only generate a limited amount of combined longitudinal and lateral forces. Although this limitation is dependent on several factors such as road conditions, dynamic load transfer onto the tire, tire and road properties, slip angle, and so forth, there is always a tradeoff in maximum longitudinal forces (braking or accelerating) and lateral forces (steering) [29,31]. The friction ellipse described this tradeoff as shown by Figure 2-5. The friction ellipse is a first-order model of the maximum combined longitudinal (F_x) and lateral (F_y) forces that a tire can generate for a given vertical load (F_z) [36]. These limitations affect vehicle stability and braking performance and is discussed in later chapters.

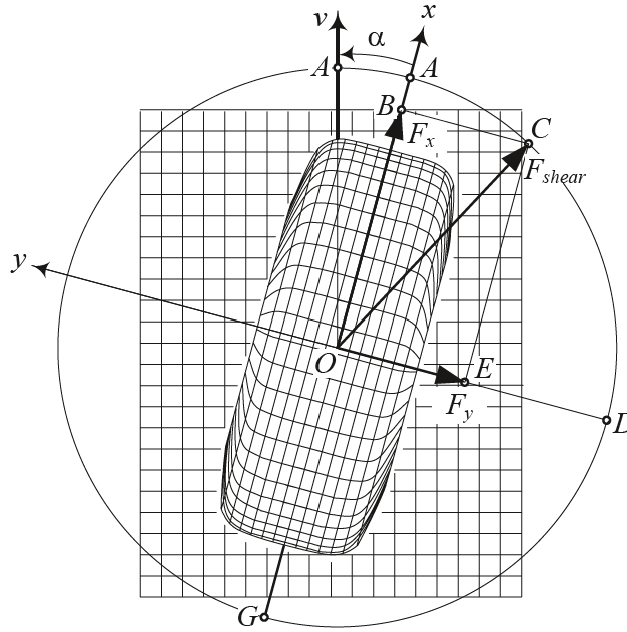


Figure 2-5. Friction ellipse for maximum combined longitudinal and lateral forces generated by a tire [29].

Just as a tire's velocity may be in a different direction than it's heading, so can a vehicle. This is also called the sideslip angle but is specific to the entire vehicle unit rather than a single tire. Sideslip of the vehicle is also related to another important vehicle dynamics concept: oversteer and understeer. Oversteer is when a vehicle rotates more than the intended path and understeer is when the vehicle does not turn as quickly as intended. In terms of tire sideslip of a 2-axle passenger vehicle, in oversteer conditions the rear axle has more sideslip than the front axle. Conversely, in understeer conditions the rear axle has less sideslip than the front axle. Neutral steer is when the vehicle follows the intended path, and the sideslip of the front and rear axles are nearly equal.

2.2 Types of LCVs

Articulated heavy vehicles (AHVs) are vehicles with a gross weight greater than 10,000 lb. and consist of at least two rigid vehicle units connected through a hitch. The front unit includes at least one steered axle and typically provides power to move the entire vehicle [32]. If the front unit carries cargo, then it can be referred to as a truck, but if it is not designed to carry cargo and instead only provides the power to pull trailing units, it is referred to as a tractor. Long combination vehicles (LCVs) are a subset of AHVs and consist of one tractor towing multiple trailers. For the LCVs discussed throughout this document, the tractor's front axle is the only steered axle in the entire combined vehicle.

Figure 2-6 presents common vehicle units and hitches used in combination vehicles. There are two common types of trailering units: semitrailers and full trailers. A full trailer is supported at the front and rear by separate axles that are part of the full trailer. The front axle is typically steerable whereas the rear is typically fixed. A semitrailer does not have an axle on the front and is instead typically supported vertically by a leading vehicle unit. A semitrailer can be converted to a full trailer by a converter dolly. The converter dolly includes an axle and fifth-wheel hitch to provide vertical support for the front of a semitrailer. A drawbar connects the dolly to the towing vehicle unit. A tractor uses a fifth-wheel hitch to support the front of a semitrailer. A truck that carries its own cargo will often tow a full trailer using a pintle hitch on the truck and drawbar on the full trailer. A less common type of trailer is the center-axle trailer that is supported vertically almost entirely by axles placed near its CG longitudinal position. The center-axle trailer is towed by a pintle-hitch and drawbar rather than a fifth-wheel coupling because, unlike a semitrailer, very little vertical support from the leading unit is needed. Both the fifth-wheel and pintle hitches provide translational constraints in all three directions. Within their normal operating range, a pintle hitch allows rotation in three directions (yaw, roll, and pitch) like a spherical joint, whereas the fifth-wheel hitch permits yaw rotation, some pitch rotation, and very limited roll compliance [33].

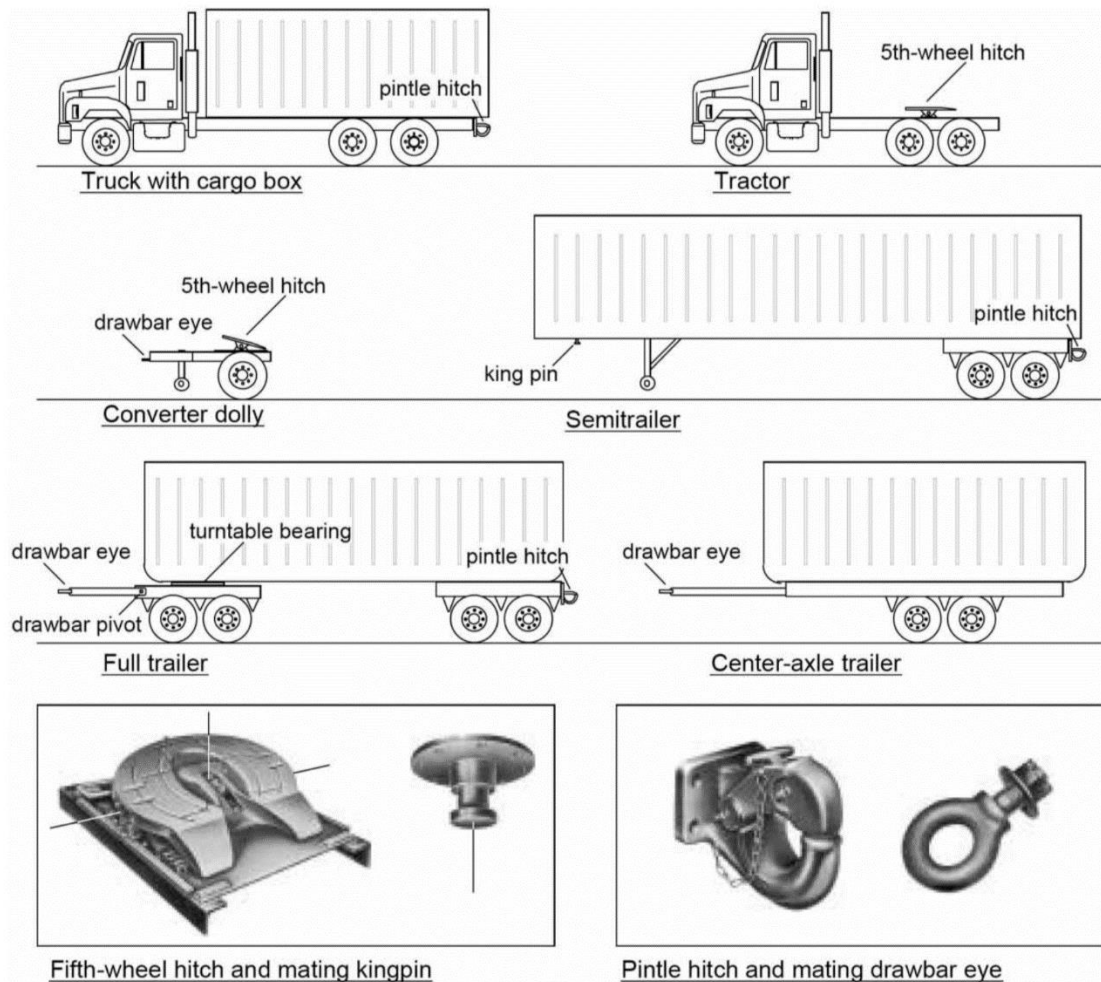


Figure 2-6. Common vehicle units and hitches in combination vehicles [32].

There are many possible combinations of truck/tractor, full trailers, semitrailers, and converter dollies. In the United States, the most common combination is the tractor-semitrailer that is often referred to as “semi-truck”. However, many package carriers utilize double-trailer configurations that include a tractor, semitrailer, and full trailer. When there are two or three trailers in combination, these are referred to as doubles and triples, respectively. These AHVs are often called long-combination vehicles (LCVs), road trains, or vehicle trains [25,34]. There are three distinct types of road trains with labels indicating how multiple trailing units are combined: A-, B-, and C-trains. An A-train double uses a converter dolly on the rear semitrailer to connect to the front semitrailer while providing articulation at the pintle hitch of the front trailer and fifth-wheel hitch of the converter dolly. On a B-train double, the front semitrailer incorporates additional axles rearward of the cargo area and a fifth-wheel coupling. The rear semitrailer connects to the front semitrailer at this point. Between the two trailers of a B-train there is only one point

of articulation rather than two. Lastly, the C-train employs a converter dolly to link the front and rear semitrailers. The C-converter dolly uses two drawbars that connect to two pintle hitches on the front trailer. This effectively results in only one yaw articulation point between the two trailers much like the B-train double, while allowing the front and rear semitrailer to be identical like that in the A-train double. In the United States, A-trains are the most common on-highway LCVs [25] due to their logistical and cost benefits for package carriers. Figure 2-7 illustrates the three types of doubles and their converter dollies.

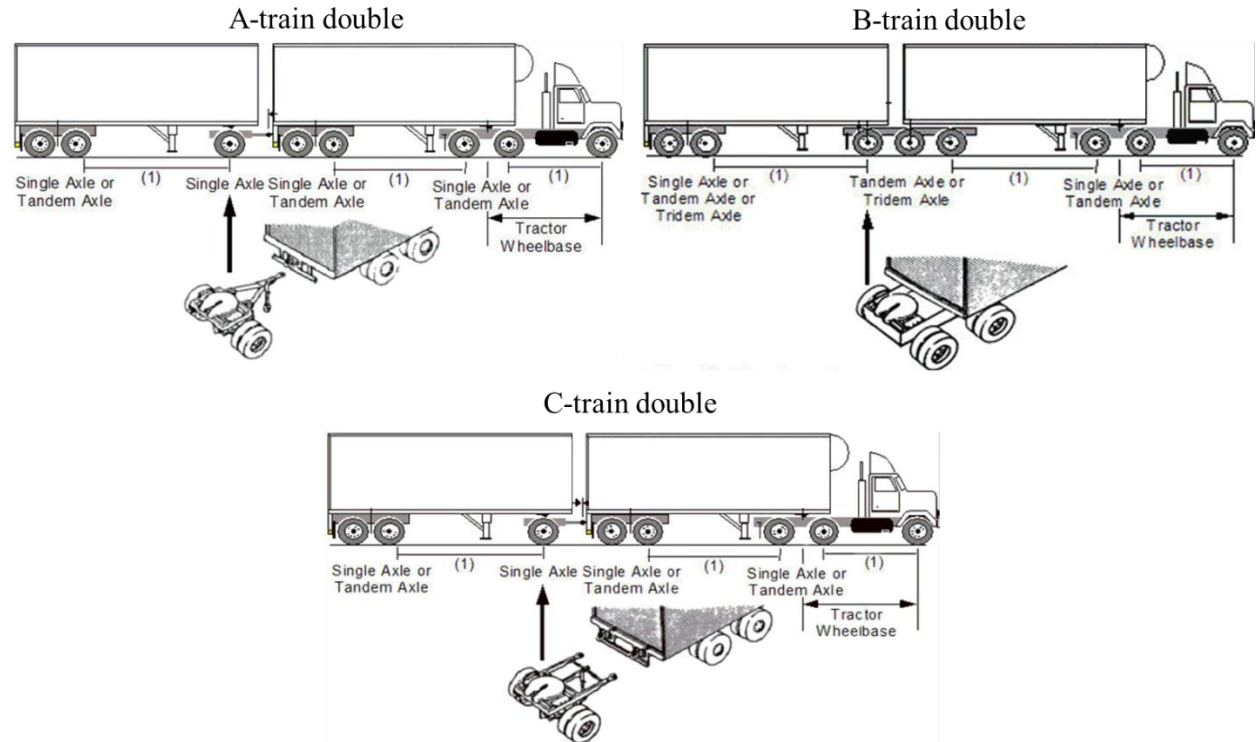


Figure 2-7. Common types of double-trailer configurations [37].

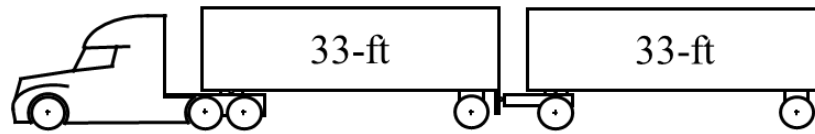
A common challenge with long-combination vehicles is that the added points of articulation in yaw motion can induce rearward amplification – the increase in lateral acceleration of the rear trailer compared to the leading vehicle unit – especially during evasive maneuvers. Because A-train doubles have an additional yaw articulation point as compared to the B-train and C-train doubles, A-trains may be more susceptible to rearward amplification than B- and C-trains. However, A-trains provide logistical benefits in that the front and rear semitrailers can be identical, allowing any trailer to be the front or rear unit. This is also achievable with C-train doubles. However, because the C-trail converter dolly's axle is far away from the front semitrailer's axle but is fixed in yaw motion, there is significant tire wear on the dolly axles, especially in low-speed tight-turn maneuvers. To counter this, some C-train converter dollies incorporate

passive, self-steering axles to help reduce tire scrub, but these can be costly and sometimes challenging to maintain. Furthermore, steerable axles may help low-speed maneuverability at the expense of high-speed directional stability. B-trains do not have this issue because the tandem axles are relatively close together, reducing tire scrubbing and resulting wear. However, in a B-train configuration the front and rear semitrailers are unique, resulting in logistical complications when coordinating a large fleet of trailers. Additionally, the front semitrailer of a B-train may require a different loading dock than the A- and C-train trailers to accommodate the forward position of the cargo area relative to the rear bumper.

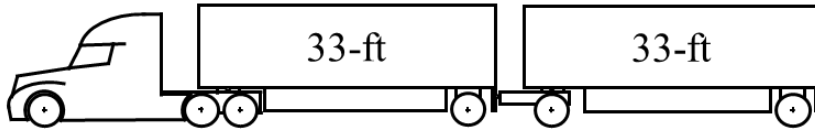
In addition to different ways to combine multiple trailers, there are a variety of trailer geometries such as box trailers, flatbeds, tankers, and so on. Box trailers are the most common in the United States and can typically be categorized into two general forms: straight-rail and drop-frame. The floor of a straight-rail trailer is flat, level with the bulkhead, and above the axles, making it easy to load and unload cargo with assistive equipment like forklifts and pallet dollies. Drop-frame trailers include a lowered deck between the rear axle and front bulkhead. This increases the total useable cargo volume and lowers the loaded trailer's CG height compared to a straight-rail trailer carrying the same payload. However, the varying floor height in the drop-frame trailer creates challenges in loading and unloading cargo. Figure 2-8 illustrates a drop-frame and straight-rail trailers.

Each trailer type has a wide range of standard sizes. The most common heavy-truck trailer in the United States is the 53-ft single straight-rail (box) semitrailer. These are typically equipped with a pair of tandem axles with dual tires. Another popular trailer configuration is the 28-ft A-double. Current federal regulations restrict long-combination vehicles to 28-ft doubles; however, some states and turnpikes permit larger combinations such as 33-ft doubles or even 28-ft triples on specific routes. Within the studies detailed in this document, four box trailer configurations are considered: 53-ft single (straight-rail), 33-ft double (straight-rail), 33-ft double (drop-frame), and 28-ft double (drop-frame). All tested double trailer configurations are A-train type. These four trailer configurations are chosen for their popularity throughout many U.S. roads [25] and for the project sponsor's interest. Figure 2-8 illustrates the four tested trailer configurations. Trailer lengths are drawn approximately to scale.

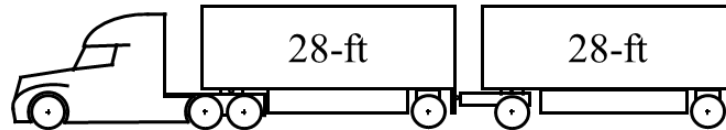
33-ft double (straight-rail)



33-ft double (drop-frame)



28-ft double (drop-frame)



53-ft single (straight-rail)

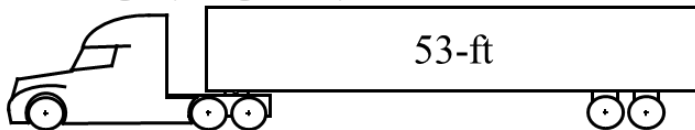


Figure 2-8. Tested trailer configurations.

2.3 Low-Speed Maneuverability

Due to their long wheelbases and trailer lengths, LCVs require significantly more space to maneuver through short radius turns than passenger vehicles. There are two common metrics that describe a vehicle's forward maneuverability: off-tracking and swept-width. Off-tracking is defined as the distance of the rearmost trailing axle path perpendicular to the steer axle path as illustrated in Figure 2-9. These paths are determined by the axles' centerlines. Swept width is the distance of the inner most body path perpendicular to the outer most body path as illustrated Figure 2-9. Swept width incorporates the natural dynamics of an LCV as well as body and cargo overhangs. Off-tracking only captures the vehicle's dynamics. Because this dissertation focuses on the dynamics of LCVs, off-tracking is the metric of choice. Determining a vehicle's swept width or off-tracking is trivial once the axle paths and vehicle geometry are known. Models and studies on off-tracking demonstrated here are only for vehicles with one steered axle that is the first axle in the lead unit (tractor or truck). All other axles on the vehicle, including its trailers, are non-steered. At low speeds, each axle's path is primarily governed by wheelbase, number of articulation points, and the steer axle path. Large amounts of off-tracking can impede the ability to maneuver the vehicle in confined areas. Chapter 3 provides a brief review of past kinematic models detailing this relationship and

proposes new numerical algorithms for LCV motion. These models are used to compare off-tracking and axle paths for a variety of trailer configurations in multiple maneuvers.

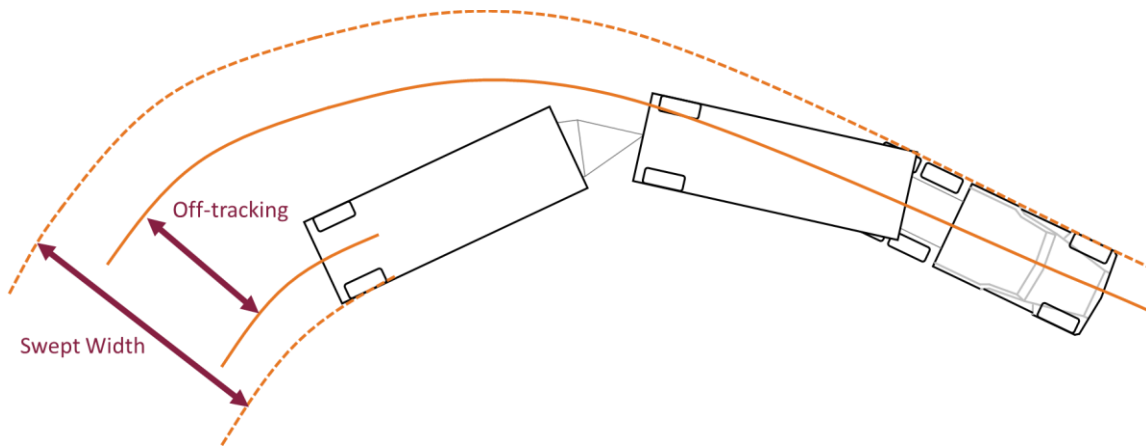


Figure 2-9. Illustration of off-tracking and swept width for an A-train double-trailer configuration.

Although there have been many instances where models were claimed to accurately capture vehicle motion, few were compared to physical tests in real-world conditions with full-scale equipment. Studies that included field tests typically omitted details on the test procedures themselves. This lack of transparency raises questions on the validity and accuracy of the field test data. One outlier paper that details the model and test procedure is provided by Erkert, et. al. [4]. In this paper, the model and algorithm provided by [5] was adopted to accommodate a trailer with varying length that is often found in logging trucks through forest roads. Physical tests were conducted with the logging tractor-trailers on flat, soft soil common to forest roads. Because the tires left tracks within the soil, off-tracking was easily measured for a single pass through a curve. Good agreement was found between the tractrix-based model and the physical test data. However, the equipment (logging trailers) and surface (soft soil) are very different than the studied combinations (53-ft single, 28-ft double, 33-ft double) and surface (asphalt) presented in Chapter 4. A more recent study by Saxe and Cebon [2] implement visual odometry to measure vehicle trajectory and off-tracking, but measurement accuracy had mixed results and was dependent on the test environment. Because past studies have not experimentally compared off-tracking of on-highway LCVs in real-world maneuvers nor provided sufficient details in test measurement and procedure uncertainties, there is a need to conduct physical tests on LCV off-tracking maneuvers and trailer configurations that replicate the project sponsor's use case.

2.4 Overview of Uncertainty Analysis for Field Test Data

One of the off-tracking study's objectives is to provide transparency in measurement uncertainty propagation through the test procedure and instrumentation into the final result, peak off-tracking. As part of this study, it is important to understand a few concepts listed below regarding measurement uncertainty. A useful guide with significantly more detail is provided in reference [17].

- Uncertainty quantifies the quality of a result with respect to measurement errors.
- Uncertainty analysis reports the propagation of measurement and model uncertainties into the result report.
- Uncertainty is defined at a specific confidence level, and all uncertainties that are combined within an uncertainty analysis must be at the same confidence level and are assumed to have normal distributions of error. For most engineering applications, 95% (or about 2 standard deviations) is an appropriate level.

If a dependent variable y is determined by some function of x that has uncertainty u_x and sample mean $\bar{x}$, then the true value of y is assumed to fall within the interval described by Equation (2-2) [17].

$$y \pm \delta y = f(\bar{x} \pm u_{\bar{x}}) \quad (2-2)$$

This can be re-written by Taylor expansion shown as Equation (2-3). The Taylor expansion is often dominated by the first derivative and may be further simplified as Equation (2-4) [17]. In short, the uncertainty in the dependent variable may be estimated by the uncertainty of the independent variable multiplied by the partial derivative of the dependent variable with respect to the independent variable. Numerical methods such as dithering – introducing small changes to the independent variable and recording the corresponding change in the dependent variable – are common for estimating uncertainty propagation through a complex system about a specific state. The result uncertainty due to a combination of multiple elemental uncertainties is determined by the root-sum-of-squares (RSS) of the elemental uncertainties, u_x , and the sensitivity index, λ_x , of the result to each uncertainty as shown in Equation (2-5). Chapter 4 details all uncertainties specific to the low-speed off-tracking field tests and Chapter 5 discusses measurement uncertainties associated with the high-speed testing instrumentation.

$$\bar{y} \pm \delta y = f(\bar{x}) \pm \left[\left(\frac{dy}{dx} \right)_{x=\bar{x}} u_x + \frac{1}{2} \left(\frac{d^2y}{dx^2} \right)_{x=\bar{x}} u_x^2 + \dots \right] \quad (2-3)$$

$$\delta y \approx \left(\frac{dy}{dx} \right)_{x=\bar{x}} u_x \quad (2-4)$$

$$u_y = \sqrt{\sum_{i=1}^N (\lambda_i u_i)^2} \quad (2-5)$$

2.5 Roll- and Directional-Stability of LCVs

Heavy vehicles are notorious for being dynamically unstable in roll motion due to their high CGs. On-road, maneuver induced (un-tripped) rollovers are of most interest because they are dependent on vehicle dynamic properties and driver behavior. Tripped or off-road slide-off induced rollover events are not discussed in this dissertation because they can be largely affected by factors outside the driver or vehicle's control. However, if a vehicle is more susceptible to un-tripped, on-road maneuver induced rollovers, then it will also be more apt to rollover in a tripped event or dramatic changes in the road surface such as when a vehicle slides off the road.

Due to the vehicle's size and mass, an LCV rollover event poses severe safety consequences to the driver and nearby traffic as well as cause catastrophic property damage. Data collected and organized by the U.S. Department of Transportation (USDOT) Federal Motor Carrier Safety Administration (FMCSA) shown in Table 2-1 clearly demonstrates the impact of LCV rollover events. For example, in 2018, 1.8% of crashes where a rollover was the first harmful event resulted in fatality. This 1.8% represents 4.2% of all fatal crashes involving large trucks [1].

Table 2-1. Severity of large truck rollovers in 2018 [1].

	Fatal Crashes		Injury Crashes		Property Damage Only Crashes		Total
	Number	Percent	Number	Percent	Number	Percent	
General	4,415	0.9%	107,000	21.4%	388,000	77.7%	499,415
Rollover	186	1.8%	4,000	39.3%	6,000	58.9%	10,186

A good first-order approximation of a heavy vehicle's roll-stability is the static stability factor (SSF). This is also commonly referred to as the stability factor or roll-stability factor. The SSF is defined by Equation (2-6) and compares the track width (T) to the CG height (H). A vehicle with higher SSF is generally less likely to rollover than a vehicle with lower SSF [39].

$$SSF = \frac{T}{2H} \quad (2-6)$$

The SSF is easy to obtain or estimate and quickly demonstrates a vehicle's susceptibility to rollover without incorporating suspension and tire compliance or dynamics of a specific maneuver. However, two vehicles with the same SSF may have different dynamic rollover thresholds [38]. The dynamic rollover threshold presented in [38] is based on energy equations and can incorporate suspension and tire compliance. Both the SSF and the dynamic rollover threshold discussed are only valid for a single vehicle unit. For tractor-semitrailers connected via a fifth-wheel hitch, the fifth-wheel provides some roll motion constraint between the two units, however, in a curve, the articulation between the two units allows some relative roll motion of one unit compared to the other due to the fifth-wheel's freedom in yaw and pitch rotation [38]. This compliance, however, is small and may be negligible compared to the freedom in roll motion within a pintle hitch. Thus, an A-train double may have two different SSFs, one for the tractor and front semitrailer and one for the rear full-trailer. The tractor-semitrailer SSF is typically taken from the semitrailer because, when loaded, it is likely to have a lower SSF and thus more susceptible to rollover than the tractor.

Another model to describe LCV rollover is the roll equation shown by Equation (2-7) [40]. Unlike the SSF, the roll equation includes the dynamics involved as the vehicle leans outward during a steady-state turn. A simple free-body diagram of an LCV trailer for which Equation (2-7) is derived is shown as Figure 2-10.

$$ma_{yh} = (F_2 - F_1) \left(\frac{T}{2} \right) - mg\Delta y \quad (2-7)$$

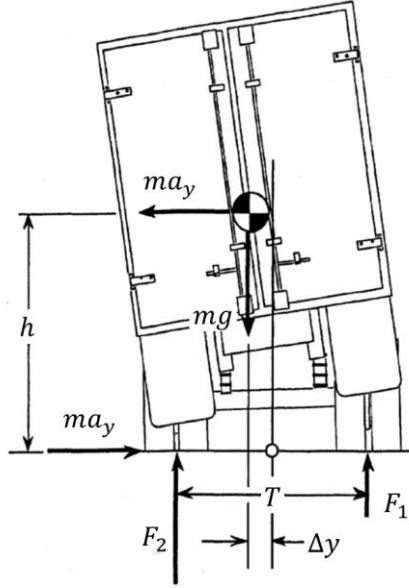


Figure 2-10. Free-body diagram for roll equation of LCV trailer [40].

Equation (2-7) may be rearranged and adapted to estimate the steady-state roll angle described by Equation (2-8) assuming small angles such that the roll angle is $\varphi = \Delta y/h$.

$$\varphi = \frac{(F_2 - F_1)T}{2hmg} - \frac{a_y}{g} \quad (2-8)$$

When evaluating the roll-dynamics of LCVs, couplings between vehicle units must be considered. Some couplings, such as a fifth-wheel hitch, have very limited freedom in roll-motion, effectively coupling two vehicle units in roll. However, pintle hitches like those used in A-trains allow for roll motion of the rear trailer relative to the front trailer. This effectively prohibits the transfer of roll-moments between the vehicle units. Consequently, the roll dynamics of the rear trailer are often not felt by the driver [34]. This can be especially dangerous when a driver is performing an evasive maneuver that may induce rearward amplification and the driver cannot appropriately correct their path to accommodate the rear trailer's instability.

Rearward amplification is the ratio of peak lateral accelerations of a trailing unit relative to a lead unit as shown by Equation (2-9). Rearward amplification greater than 1 is a common occurrence for doubles and triples traveling through evasive maneuvers such a fast lane change to avoid an obstacle. When rearward amplification is high, the rear trailer is more apt to rollover than the front trailer or lead unit. As mentioned, it can be quite challenging for a driver to feel the roll dynamics of the trailer(s) due to the isolation of the

cab from the trailers, particularly for double and triple trailers [34]. For an evasive maneuver, the rear trailer may swing out beyond the tractor's path. This is especially susceptible to occur when rearward amplification is greater than one and/or when the road surface is slick such as in snowy conditions. This swingout is a directional instability.

$$RA_i = \frac{a_{yi}}{a_{y1}} \quad (2-9)$$

Directional instability caused by rearward amplification can also result in jackknifing. Jackknifing is a phenomenon unique to vehicles with trailers and occurs when the yaw articulation angles between vehicle units reaches the geometric limitations of the equipment. LCVs are especially sensitive to jackknifing in slippery road conditions where there is limited lateral force between the tires and ground to ensure directional control. For LCVs, jackknifing can occur between any vehicle unit pair. Because the converter dolly connecting the front and rear trailer of an A-double has two articulation points in yaw, jackknifing can occur at either point. This results a total of three possible types of jackknifing on an A-train double as illustrated in Figure 2-11. Jackknifing events can be dangerous and pose serious safety concerns to truck drivers and nearby traffic. In 2018, jackknifing of large vehicles (LCVs) resulted in 188 fatal crashes, 2,000 crashes with serious injuries, and 7,000 property damage only crashes [1]. Figure 2-12 pictures an A-train double jackknifed between the front trailer and tractor, and between the rear trailer and dolly. In the situation shown below, the jackknifing event likely occurred from braking and losing directional stability on the snowy road. An increase in articulation angle between vehicle units increases the chance of jackknifing.

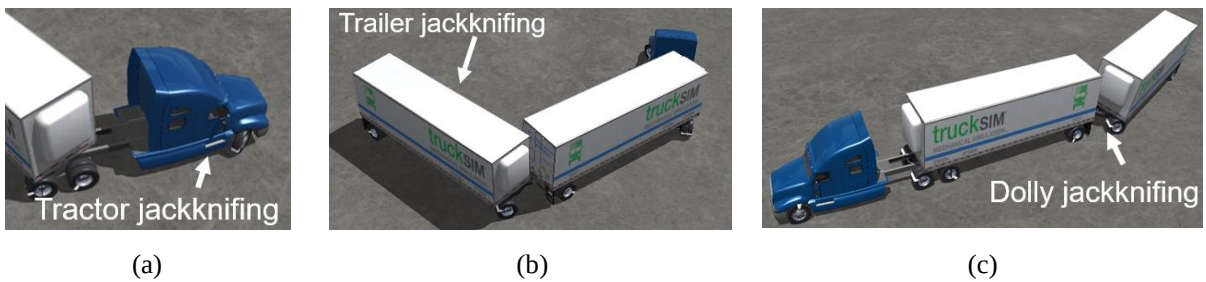


Figure 2-11. Potential jackknifing events: (a) tractor, (b) trailer, and (c) dolly.



Figure 2-12. Jackknifed A-train double (Massachusetts State Police, 2019).

Directional stability may also refer to the deviation of the tractor-trailer's path from the driver's intention. As mentioned in Section 2.1, this is observed as either understeer, where the tractor does not rotate as much as the driver's steering input, or oversteer, where the tractor rotates more than the driver's steering input.

2.6 Stability Control Systems

Before the advent of electromechanical roll-stability control systems in LCVs, vehicle manufacturers and road designers trialed several different approaches to warn the driver of a high probability of rollover. One of the earlier on-vehicle methods was proposed in 1990 and utilized lateral acceleration of the tractor or trailer and roll angle of the trailer axle [59]. In 1994, the USDOT tested active road signs that flashed warnings indicating to the driver that their truck was likely to rollover through an up-coming exit ramp as determined by sensors that estimated the vehicle's weight and speed. This system was believed to be cost effective in reducing rollovers for exit ramps with a history of at least one rollover event for every five years [42]. Prototypes of a similar system from the same engineers were later tested at three different sites along the Capital Beltway in Virginia and Maryland as part of a study for the Federal Highway Administration. Speed warnings were successful in reducing heavy vehicle speeds before rollover-prone exit ramps or curves, and, in the 3 years that the systems were tested, no rollover accidents occurred [43]. Another approach to warn drivers of impending vehicle instability was demonstrated in a 1998 report to NHTSA that detailed the Roll Stability Advisor, an on-board visual guide to instruct the driver whether they were nearing the roll-stability limits of the vehicle. The Roll Stability Advisor used several sensors on the tractor and trailer to estimate the rollover threshold with approximately 0.02 g accuracy after driving for approximately 1 min over 40 mph. The system would then provide a needle gauge indicating the vehicle's lateral acceleration and its proximity to the rollover threshold in either the left or right direction [44].

In the 1990s and early 2000s, industry and academia pursued a variety of active electromechanical control systems aimed to improve the roll-stability of an LCV without requiring driver intervention like that needed within a system that only provides a warning. Popular active electro-mechanical control systems for LCVs can be categorized into two groups: suspension-based and brake-based. The suspension-based systems aim to reduce roll compliance, lean the vehicle into the turn or distribute load transfers across an axle. These systems effectively stiffen the suspension system in roll motion and provide a counter moment to try to keep the vehicle upright. As illustrated in [40], reducing roll compliance in a vehicle's suspension and tires increases roll-stability. Brake-based systems adopt electronic brake systems used for anti-lock braking (ABS) to provide differential braking between the left and right tires. Differential braking can provide a counter yaw moment to ensure directional stability that can reduce rearward amplification while simultaneously slow the vehicle. Vehicle speed is one of, if not the most influential factor that causes vehicle rollover. This is roughly illustrated by Equation (2-10) describing the lateral (centripetal) acceleration of a particle moving at tangential speed v along a circular path with radius R . If the speed is doubled, the lateral (centripetal) acceleration quadruples. Extending this to vehicle dynamics, this means that the likelihood of rollover in a steady-state turn is approximately proportional to the square of the vehicle's speed.

$$a_y = \frac{v^2}{R} \quad (2-10)$$

The following subsections provide an overview on the general design of different stability control systems. The pros and cons of each approach are briefly reviewed.

2.6.1 Active and Passive Suspension to Improve Trailer Roll-Stability

In academia, there have been countless studies on active anti-roll bar suspension systems to reduce rollover propensity of heavy trucks and LCVs. However, many of these studies focused on defining theoretical limits of control strategies without considering practical limitations for equipping vehicles with these technologies. For example, Lin *et al* evaluated active anti-roll bar control strategies based on load transfer, roll angle, and lateral acceleration without accounting for the actuator mechanics and physical implementation [45]. In some of their later work, they pursued optimal roll control for a single-unit heavy truck but did not conduct physical tests [52]. Similarly, Dunwoody and Froese proposed an active roll-control for a semitrailer that utilized hydraulic actuators on the axles as well as above the fifth-wheel plate.

In simulation, this system proved to increase the roll-stability by actively leaning the trailer body into the turn. However, without a physical prototype to test with real drivers, the design theory has its limitations for application [48]. Likewise, Huang *et al* evaluated different control strategies for active anti-roll bars on a tractor-semitrailer and determined which axles were most appropriate to include such equipment. This study was conducted solely in simulation and demonstrated that active anti-roll bars were most effective in improving the vehicle's roll stability when installed on the trailer axles as opposed to the tractor's [51]. Yu and Özgüner suggest a rollover detection system and expanded the system to determine control parameters for an active anti-roll suspension [57]. In some of his earlier work, Sampson *et al* developed linear models of yaw-roll dynamics and designed a state feedback roll control system for a tractor semitrailer [46]. The physical limitations were not pursued until later when Sampson designed an active anti-roll bar that utilized hydraulically actuated frame mounts to control the forces shared by the air springs on either side [47,49]. This system was tested in simulation while accounting for actuator limitations and proved to increase roll-stability. It wasn't until 2005 where physical test results were published by Sampson's colleagues Miede and Cebon [50,58]. There have been numerous academic pursuits for an active suspension system to improve roll-stability of LCVs, however many of these investigations did not thoroughly evaluate the practical limitations for including such systems on tractor trailers. Many proposed methods would require a hydraulic pump and circuit to power hydraulic actuators. These systems would inherently be extraordinarily expensive to purchase, install, and maintain. Additionally, the added weight of these systems would reduce payload capacities of the vehicles. For these reasons, suspension-based methods to improve trailer roll-stability are typically passive and based on balanced pneumatic circuits for air springs as described in the following.

Many LCVs have pneumatic suspensions to accommodate the dramatic changes in sprung mass for different cargo loads. A load leveling valve that is connected to an axle opens and closes an orifice connecting the air supply tank to the suspension air spring and an orifice from the air spring to the atmosphere. When the vehicle is heavily loaded, the suspension compresses, opening the orifice between the supply tank and air spring to ultimately increase the air pressure and volume inside the air spring until the vehicle is raised to a level position. When the vehicle is unloaded, air is exhausted from the air spring until the vehicle becomes level. The load leveling valve, control arm, and connection rod on an air spring equipped trailer is shown in Figure 2-13.

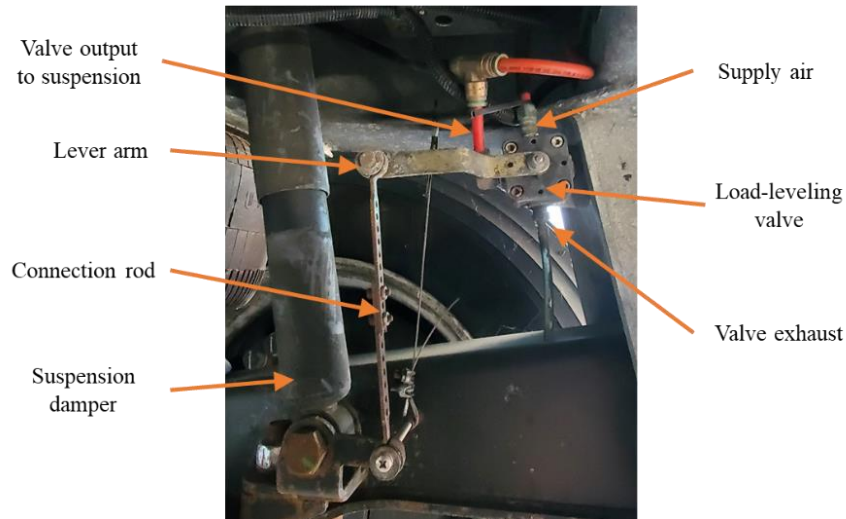


Figure 2-13. Picture of load-leveling valve for trailer air-spring suspension.

Since the pneumatics needed to support air springs and air brakes already exist within most LCV trailers, balanced pneumatic circuits are a cost-effective method to improve roll-stability without requiring significantly more equipment. Furthermore, balanced pneumatic suspensions are purely mechanical, unlike the electromechanical, hydraulically actuated active anti-roll suspension systems discussed previously. A balanced pneumatic suspension system differs from standard suspension systems in that there are multiple load-leveling valves for a given axle set rather than just one. Figure 2-14 illustrates the plumbing arrangement of tandem trailer axles with air springs in both standard and balanced pneumatics. Similar setups are installed on tractor tandem drive axles. Most tractor steer axle suspensions are leaf-springs and thus do not use a balanced pneumatic circuit. Simulation studies and experimentation have shown that balanced pneumatic suspension systems can provide better roll-stability than standard suspension systems [53,54,55]. However, because these pneumatic systems are inherently slow due to large air spring volume and flow restrictions through valves, a heavy truck may rollover before the pneumatics can affectively increase suspension roll stiffness [55].

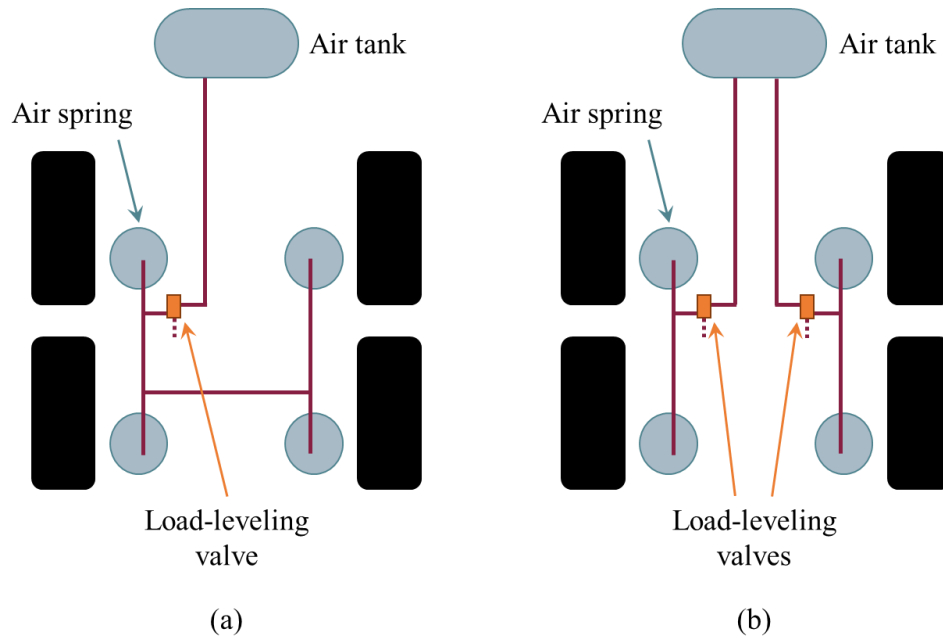


Figure 2-14. Plumbing circuit for (a) standard (single load-leveling valve) and (b) balanced (two load-leveling valves) pneumatic suspensions.

2.6.2 Electronic Stability Control (ESC) and Roll-Stability Control (RSC)

Differential braking stability control systems utilize electronic brake systems to supply and control fluid pressure to actuate individual brakes independently. These were first introduced in passenger vehicles as an extension of existing anti-lock brakes (ABS) that became popular in the late 1980s to early 1990s. Heavy vehicles later adopted similar technology but with slightly differing characteristics to accommodate the air brakes in place of hydraulic brakes. On heavy trucks, the control strategies for these systems are slightly different from passenger vehicles due to differing dynamic characteristics. For example, heavy vehicles are much more susceptible to rollover than passenger vehicles, and passenger vehicles are more susceptible to directional instability than roll instability. Additionally, most passenger vehicles, aside from light trucks, typically do not have to worry about dynamics introduced by towing heavy trailers. This section will highlight brake-based stability control system operations specific to LCVs.

Brake-based stability control systems may be referred to as either RSC (roll-stability control) or ESC (electronic stability control). In this dissertation, ESC is designated for the tractor or truck and incorporates control strategies to improve directional- and roll-stability. RSC is specific to trailers and is designed to primarily target roll-stability. RSC in LCVs may also provide some directional stability to the trailers as a result of their braking action. In short, ESC and RSC refers to the brake-based stability enhancement

systems implemented on the tractor and trailers, respectively, where RSC may be regarded as a sub-category of ESC. In literature, RSC and ESC are also referred to as RSP and ESP where the word “control” is replaced with “program”. Not all RSC and ESC systems are the same, and some provide braking action on vehicle units other than the one the system is attached to. For the systems described and tested herein, RSC is only on a trailer, ESC is only on a tractor, and there is no communication between stability systems of connected vehicle units nor does one system provide braking action on another unit.

Typical inputs for an ESC control algorithm include lateral acceleration, yaw rate, steer angle and rate, wheel speeds, vehicle mass, and driver throttle. Most of these inputs can be directly measured with sensors, however, the vehicle mass is typically estimated by either air pressure supplied to suspension air springs or from engine torque and transmission data provided by the SAE J1939 CAN bus [62]. Lateral acceleration and yaw rate relative to driver inputs are the dominate factors used within the ESC control algorithm. Once these values reach or surpass some threshold, ESC commands the electronic braking modules to brake specific wheels with a tailored amount of pressure. A preview lateral acceleration is estimated by Equation (2-11) below. This value provides a threshold below the normal ESC activation thresholds to accommodate delay in vehicle inertial and component mechanical responses [63]. In Equation (2-11), δ is the steering angle (*rad*), R_δ is the vehicle roll coefficient (*rad · s²/m*), v_x is the vehicle longitudinal speed (*m/s*), and L is the vehicle wheelbase (*m*). The preview acceleration can improve ESC performance by triggering ESC intervention as much as 200-300 ms before the lateral acceleration threshold is achieved [34]. As of 2005, all newly manufactured tractors must be equipped with ESC according to FMVSS 136 [64].

$$a_{y_{preview}} = \frac{\delta v_x^2}{R_\delta v_x^2 + L} \quad (2-11)$$

Brake-based RSC systems perform similarly to electronic stability control systems; however, RSC systems typically do not have as many inputs. RSC inputs typically include lateral acceleration from one or more locations on the trailer body, trailer wheel speeds, and air spring pressure. The suspension air spring pressures are used to estimate the trailer mass including the cargo load. Lateral acceleration is the predominate factor for triggering RSC intervention. At vehicle startup, estimated rollover lateral accelerations are typically set to a low, conservative value. Then, as the vehicle travels at moderate speeds and experiences a range of lateral accelerations through turns, RSC initiates test pulses on the trailer brakes and alters the lateral acceleration threshold according to the vehicle response. It is best for RSC systems to estimate trailer mass and CG height for each vehicle startup because trailer masses can vary

tremendously from changes in cargo payload. If control thresholds are too low, RSC will intervene too frequently possibly annoying the driver, and worse, cause the brakes to heat up and increase risk to brake fade. If control thresholds are too high, RSC will not be as effective at preventing rollover. Fancher, Winkler, Ervin, and Zhang were among the first to consider and simulate a brake-control system to reduce undesirable dynamics of LCVs by targeting rearward amplification of full-trailers found in A-train doubles [60]. Svenson *et al* introduced brake-based RSC within the National Advanced Driving Simulator and found that RSC improved roll-stability in simulation, and the simulation results closely matched experimental tests [61]. Many suppliers have their own RSC systems with proprietary control algorithms, and there are currently no regulatory requirements for LCV trailers to have RSC. Chapter 6 will further detail ESC and RSC operation through the comparison of their roll-stability benefits in 33-ft straight-rail double trailers.

2.7 Heavy Duty Vehicle Braking Systems

This section introduces the pneumatic braking circuit on LCVs and the brake types common to LCVs: drum brakes and disc brakes. All mechanical friction brakes slow a vehicle by converting kinetic energy into heat. Drum and disc brakes utilize different mechanism to accomplish this task, and consequently, have slightly different performance characteristics. In nearly all LCVs in the U.S., both brake types are pneumatically driven. Figure 2-15 provides a detailed illustrates of the complete pneumatic circuit for a common 6x4 tractor-semitrailer. Air is pressurized by an engine-driven compressor and stored in multiple reservoirs. A governor regulates the air pressure, and a dryer removes moisture that has entered the system. The driver's foot brake actuates the treadle valve that then delivers air from the treadle valve supply port to the delivery port. This air then reaches the relay valves (also known as service valves or service relay valves), supplying pressure from a reservoir tank to the brake chambers. Gladhands couple two vehicle units together, and each vehicle unit has its own reservoir (supply tank) that is ultimately pressurized by the tractor's compressor. Check valves throughout the system ensure that upstream leaks do not immediately affect pressure in reservoirs or components downstream. See ref. [69] for more information.

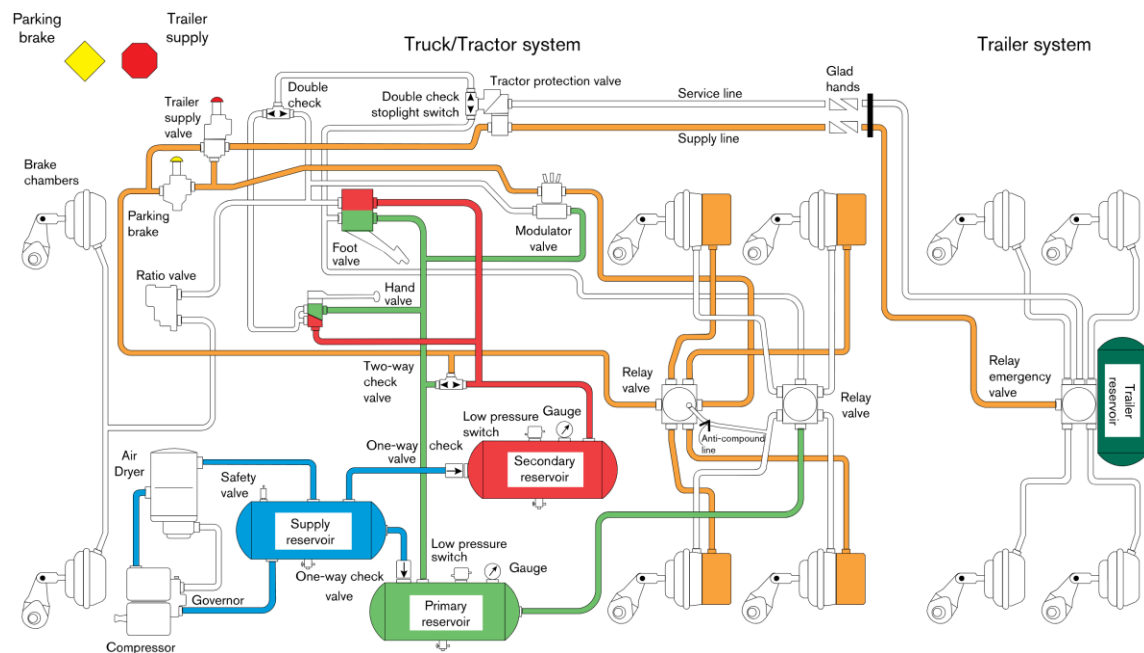
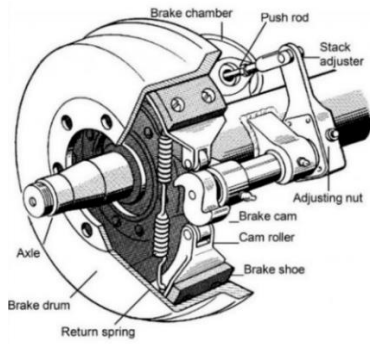


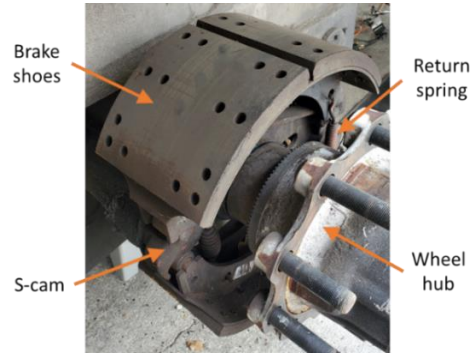
Figure 2-15. Complete air brake circuit of a tractor-semitrailer [65].

2.7.1 Trailer Drum Brakes

Drum brakes currently dominate the U.S. LCV market and have a long history throughout the commercial vehicle industry. Figure 2-16 provides a cutaway illustration and picture of a trailer drum brake. Pressurized air within the brake chamber extends the push rod rotating the s-cam that forces brake shoes outward against the inside wall of a rotating brake drum. The drum is connected directly to and rotates with the wheel hub. Friction between the non-rotating brake shoe and inner wall of the brake drum slows the wheel axle's rotation and thus the vehicle. The s-cam's profile pushes against the cam rollers that force the brake shoes outward against the drum's inner wall. The return spring keeps the cam roller in contact with the s-cam so that the brake shoes disengage from the brake drum when air pressure is released from the brake chamber. As shown in the picture, brake shoes are very large and cover a significant portion of the drum inner wall surface. For extended braking durations, thermal expansion causes the drum's radius to increase. In this condition, the brake shoes must extend further to achieve the same normal force between the brake shoes and drums as before the drums expanded. At very high temperatures, the friction coefficient at the interface between the friction material (brake shoe) and rotating member (drum) reduces. These two factors, thermal expansion and reduced friction coefficient, can result in brake fade – the decrease in braking torque after extended brake application duration and/or frequency. Brake fade can occur with any type of friction brake.



(a)



(b)

Figure 2-16. (a) Illustration of pneumatic drum brake mechanism [66] and (b) internals of drum brake assembly.

2.7.2 Trailer Disc Brakes

Rather than pushing brake shoes outward against a spinning drum, disc brakes push brake pads against the sidewalls of a metal rotor that rotates with the wheel axle. Figure 2-17 provides a picture of a trailer disc brake. Pistons within the brake caliper extend the brake pad outward to collide with the rotor sidewall. The brake pad's working surface is considerably smaller than the drum brake's shoe. This typically results in the disc brake pads requiring more frequent replacement than drum brake shoes. However, replacing disc brake pads is typically less laborious than drum brake shoes. Although the brake pad's working surface is smaller than the drum brake shoes, disc brakes tend to outperform drum brakes, especially during extended braking periods where brakes are susceptible to fade. This is partly because as the disc brake's rotor expands with increase in temperature, it expands into the brake pad rather than away like the drum from the drum brake shoe. Disc brakes are popular on LCVs in Europe and are growing in popularity within the U.S. commercial vehicle industry.



Figure 2-17. Trailer air disc brake.

Chapter 3

Low-speed Off-Tracking of LCVs – Comparison of Numerical Tractrix Models

Off-tracking of LCVs has been a popular topic for many decades and was first considered as early as the 1940s to provide guidelines for road designers. Since the U.S. Interstate Highway System (IHS) was signed into action in 1956, vehicle off-tracking became a more prevalent topic with rapid road construction through the 1960s [7]. Without access to today's modern technologies and computers, studies on vehicle motion were relatively crude and typically used graphical techniques such as very small physical models or a tractrix integrator to physically plot axle paths on paper through a hypothetical maneuver [6,8,9]. This approach was limited to the plotted maneuver and its accuracy could be influenced by the user's skill and experience with the tools. Figure 3-1 shows a tractrix integrator.

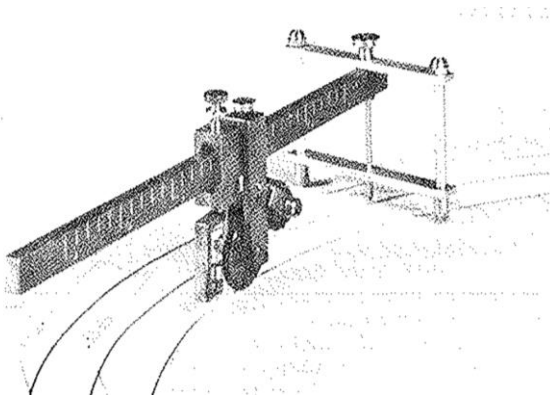


Figure 3-1. Tractrix integrator [9].

This chapter reviews kinematic models used to estimate tractrix curves describing axle paths of LCVs. Two new numerical geometric models are presented and compared to existing models. Differences in the models' accuracy, ease of implementation, and computational expense are discussed. The average path defined by the two new numerical models is decidedly the most accurate and least sensitive to anomalies in the input (steer axle) path. This model is then compared to a high-order dynamic model from TruckSim® and demonstrates very good agreement in the axle paths and resulting off-tracking. Lastly, the new numerical model is used to compare common LCV configuration on U.S. roads: 53-ft single, 28-ft double, and 33-ft double. This model is then used in Chapter 4 as part of processing field test data.

3.1 Kinematic models

Kinematic models estimate vehicle motion or off-tracking without accounting for inertial properties of the vehicle nor a vehicle's compliance (tire slip, roll steer, etc.). Additionally, all wheels for an axle group are typically represented as a single wheel at the geometric center of the axle group. This is part of the bicycle model – the foundation for describing the geometric nature of vehicle motion through kinematic models. Kinematic models are accepted as first-order approximations for vehicle motion at all speeds and can be very accurate at low-speeds where tire slip is negligible. In this context, because kinematic models are defined by the vehicle's geometry, they are also commonly called geometric models.

3.1.1 Bicycle Model and Vehicle Trajectory

The bicycle model gets its name from modeling a vehicle's geometry and wheels with only two wheels like a bicycle – one steered at the front and one non-steered in the rear. The bicycle model for a 6x4 tractor with a single steered axle and tandem drive axle pair is illustrated in Figure 3-2. In this application, the model's rear wheel is located at the geometric center of the pair of tandem axles. Because the model assumes no tire slip, it also assumes the vehicle's steering geometry maintains perfect Ackermann throughout its steering range. Additionally, the bicycle model assumes that the vehicle is travelling along a perfectly flat surface and does not experience any external forces. The bicycle model estimates the vehicle's global trajectory by integrating the wheel speeds and vehicle yaw rate. For the example 6x4 tractor shown, we may assume the drive axle speed and steer angle are known from driver inputs. From these two values alongside the tractor's geometric wheelbase, the tractor's yaw rate and steer axle speed are determined by Equation (3-1) and (3-2), respectively. In the figure and equations, v_r is drive axle speed, v_f is steer axle speed, δ is the steer angle, L is the tractor wheelbase, R_f is the turn radius of the steer axle, R_r is the turn radius of the drive axles' geometric center, and ω is the yaw rate of the vehicle.

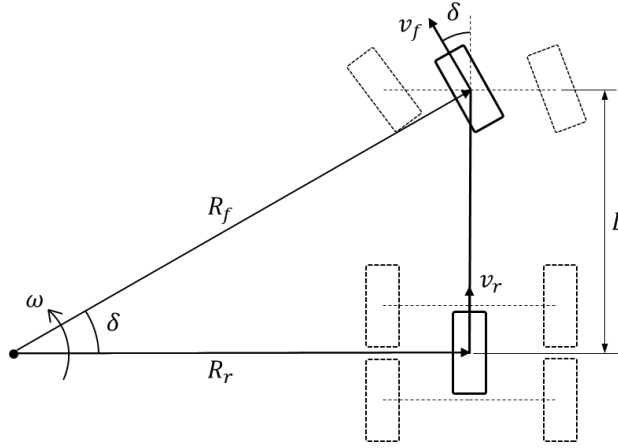


Figure 3-2. Bicycle model of 6x4 tractor.

$$\omega = \frac{v_r}{R_r} = \frac{v_r}{\sqrt{\left(\frac{L}{\sin \delta}\right)^2 - L^2}} \quad (3-1)$$

$$v_f = R_f \omega = \left(\frac{L}{\sin \delta}\right) \frac{v_r}{\sqrt{\left(\frac{L}{\sin \delta}\right)^2 - L^2}} \quad (3-2)$$

3.1.2 Steady-State, Maximum Off-Tracking Model

Trailers may be added to the bicycle model to estimate steady-state motion through a turn of constant radius as illustrated in Figure 3-3 for a tractor-semitrailer. Notice that the trailer axles' turn radii intersect the same point as the tractor steer and drive axle turn radii. This occurs at the steady-state condition where the trailers reach their maximum off-tracking for a given steer axle path of constant curve radius. The same variables in Figure 3-2 are used in Figure 3-3. For vehicle unit i , L_i is the wheelbase and R_i is the rear axle turn radius. OT_{ss} is the steady-state (maximum) off-tracking of the entire vehicle. The forward or backward offset from the i th vehicle unit's rear axle to the hitch that connects the $i+1$ trailing unit is called the hitch offset and is labeled as HO_i . Typically, HO_i is close to and may be approximated as zero for the tractor and converter dolly fifth-wheel hitches but should be included if known [9,12]. The hitch offset from the dolly's pintle hitch to its leading unit's rear axle, however, is sufficiently large for common equipment that it must be included. The general form of the Western Highway Institute (WHI) formula for estimating maximum off-tracking as illustrated in Figure 3-3 is described by Equation (3-3) [6,9,10,11,12]. This equation is based on right-triangles bound by each vehicle unit length and their respective front and rear axle (or kingpin) turn radius.

$$OT_{ss} = R_f - \sqrt{R_f^2 - \sum_{i=1}^n L_i^2 + \sum_{i=1}^{n-1} HO_i^2} \quad (3-3)$$

Figure 3-3 also provides some additional variables of interest. Particularly, the articulation angle θ_1 between the two vehicle units is the difference in the two units' orientations φ_1 and φ_2 . From basic trigonometry, the articulation angle between successive vehicle units can be found as Equation (3-4). By convention throughout this document, a positive articulation angle occurs during a left-hand turn and a negative articulation angle occurs during a right-hand turn. Inboard off-tracking, however, is always a positive quantity.

$$\theta_i = \varphi_i - \varphi_{i+1} = \sin^{-1} \left[\frac{L_{i+1}}{R_{H_i}} \right] + \sin^{-1} \left[\frac{HO_i}{R_{H_i}} \right] \quad (3-4)$$

Equation (3-4) assumes dimensions L_{i+1} and HO_i are known and that R_{H_i} may be calculated as Equation (3-5) below.

$$R_{H_i} = \sqrt{R_f^2 - \sum_{k=1}^i L_k^2 + \sum_{k=1}^i HO_k^2} \quad (3-5)$$

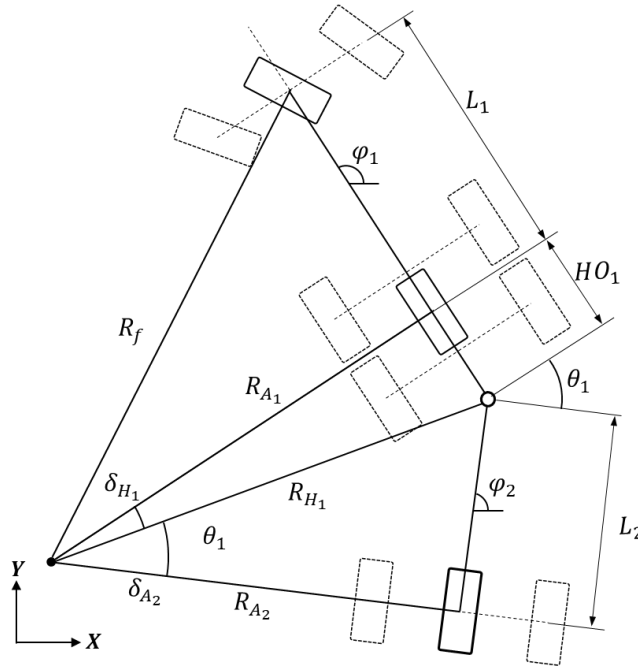


Figure 3-3. Bicycle model of tractor-semitrailer in steady-state turn.

3.1.3 Tractrix Model of Articulated Vehicle Motion

The maximum, steady-state off-tracking as calculated by Equation (3-3) provides a quick and easy comparison between differing vehicle dimensions or curve radii, but it fails to capture the off-tracking transients as the vehicle enters and exits a curve. These transients are critical for roadway designs and autonomous vehicle path planning algorithms. In 1963, Jindra was among the first to model off-tracking transients by implementing tractrix equations [13]. This model, however, was limited to a single-unit vehicle (no trailers) and constant radius curves with straight-line entry and exit. In 1985, Woodrooffe *et al.* presented a model to determine transient development of off-tracking for a multi-unit vehicle entering or exiting a curve of constant radius [14]. Although this was a step forward from Jindra because it could be used for articulated vehicles and closely matched field test data, it was still limited to straight-line entry/exit and a curve with constant radius. In 1986, Sayers points out the need for a numerical method to model vehicle motion for any maneuver and proposed an algorithm for use on an Apple II computer [12]. Then, in 1991, Sayers provided a continuous model to describe vehicle motion of any number of trailing units for any given lead unit steer axle path [5]. For small step size along an input steer axle path, the numerical model converges to the continuous equations. Like other kinematic models mentioned, Sayers' method assumes that the lead unit's front axle is the only steered axle and that there is no lateral tire slip. The steer axle path is taken in terms of arc radii and arc lengths and provides continuous differential equations of motion based on tractrix curves. Sayers' numerical method is detailed below. In his algorithm, the steer axle is stepped through the input path in very small distance increments (≤ 0.1 m). For each step along the steer axle path, each vehicle unit's axle and hitch positions are determined sequentially, starting from the tractor and ending with the rearmost trailer. Sayers assumes each axle's trajectory from one step to the next follows a circular path of constant curvature, where the curvature is defined by the corresponding vehicle unit's wheelbase and the preceding vehicle unit's hitch path. Figure 3-4 presents the algorithm flowchart. For details on the algorithm's equations, please see reference [12] or Appendix A for the corresponding MATLAB function. A few typos were found in the reference paper, so it is recommended to follow the MATLAB function if you wish to replicate the model.

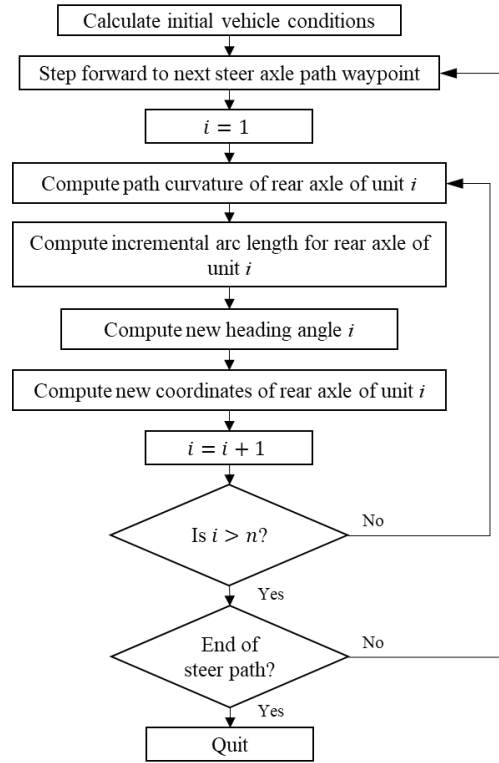


Figure 3-4. Sayers' numerical tractrix algorithm flowchart for LCVs. Adapted from [12].

Although Sayers' numerical method closely matches tractrix curves for long combination vehicles, it has limitations and challenges for implementation. In general, the algorithm is noticeably more computationally expensive than a simpler geometric model. With modern computers, this may not be an issue unless used to predict a vehicles motion for a given path in real-time. Additionally, the algorithm requires a steer axle path defined by path curvature and arc lengths between waypoints. If the steer axle path is provided as a series of cartesian coordinates, a conversion to path curvature and arc length is required. This adds computational expense and is susceptible to numerical challenges when the path is irregular. Lastly, there must be very little difference in arc length from one waypoint to the next along the steer axle path for the algorithm. This is because, in part of calculating the arc length of a trailer axle path from waypoint i to $i+1$, the arc length from point $i-1$ to i is assumed to be the same as from i to $i+1$. In summary, significant variations in the input path or large step sizes can significantly impair the algorithm's accuracy. This will be discussed further in comparison to the proposed models in following sections.

3.1.4 Simple Geometric Models of Long Combination Vehicle Motion

In preparation for analyzing field test data detailed in Chapter 4, two simpler geometric models were developed to estimate tractor-trailer motion by emulating tractrix curves numerically. Both new algorithms

are similar to Sayers' numerical approach in that the steer axle path is the input and the trailing axle paths are determined sequentially from the tractor to the rearmost trailer at each input path waypoint index. The proposed numerical algorithms for LCV motion at low-speeds are called simple geometric models – SGMs for short – because they provide a simple and approachable method to model vehicle motion based solely on the vehicle and steer axle path geometries. The SGMs assume the following: the vehicle is traveling forward at a low-speed such that tractrix curves are sufficiently accurate representations of axle paths; only the tractor's front axle is steered; and each axle travels along a straight line from one step to the next. The SGMs converge to continuous tractrix curves for fine input path resolution, accept an input path as a series of cartesian coordinates, and unlike Sayers' model, do not require consistent spacing in the input path waypoints and distances traveled by each axle from one waypoint to the next. The axle paths are determined sequentially from the input steer axle path to the rearmost trailing axle. The SGMs differ in where tire slip is neglected as described in the following paragraphs. The three proposed models are referred to as SGM-A, SGM-B, and SGM-C. SGM-C is the average path of independent SGM-A and SGM-B path calculations; therefore, only SGM-A and SGM-B are detailed. In both SGM-A and SGM-B as illustrated in the following figures, points A_i and P_i are the axle and pivot, respectively, of the i th vehicle body where $i = 1$ is the steer axle and $i = n$ is the rearmost trailer. For these models, an LCV has $n - 1$ vehicle units plus one steer axle resulting a total of n bodies. The coordinates of an axle's previous and current positions are written as vectors $\mathbf{r}_{A_i}$ and $\mathbf{r}'_{A_i}$, respectively.

Simple Geometric Model – No Tire Slip Just After Current Position (SGM-A)

SGM-A assumes no tire slip just after the vehicle's previous position such that the current axle position is along the longitudinal axis of the vehicle unit's previous position – i.e.: A'_2 exists along the line connecting A_2 and P_1 in Figure 3-5. δ_i is the angle between body i and $i-1$; φ_i is the global orientation of body i ; and β_i is the difference in the past and present orientations of body i . The vector connecting a hitch's previous position $\mathbf{r}_{P_i}$ to its current position $\mathbf{r}'_{P_i}$ is $\mathbf{d}_i$. The steer axle is the first body, the tractor second, the first trailer third, and so forth. Figure 3-6 provides the SGM-A algorithm flowchart corresponding to the illustration in Figure 3-5. L_i is the i th body's wheelbase – distance from the lead pivot to the rear axle(s) representation – and H_i is the i th body's hitch length – the distance from the lead pivot to the rear hitch or equivalently the summation of wheelbase and hitch offset.

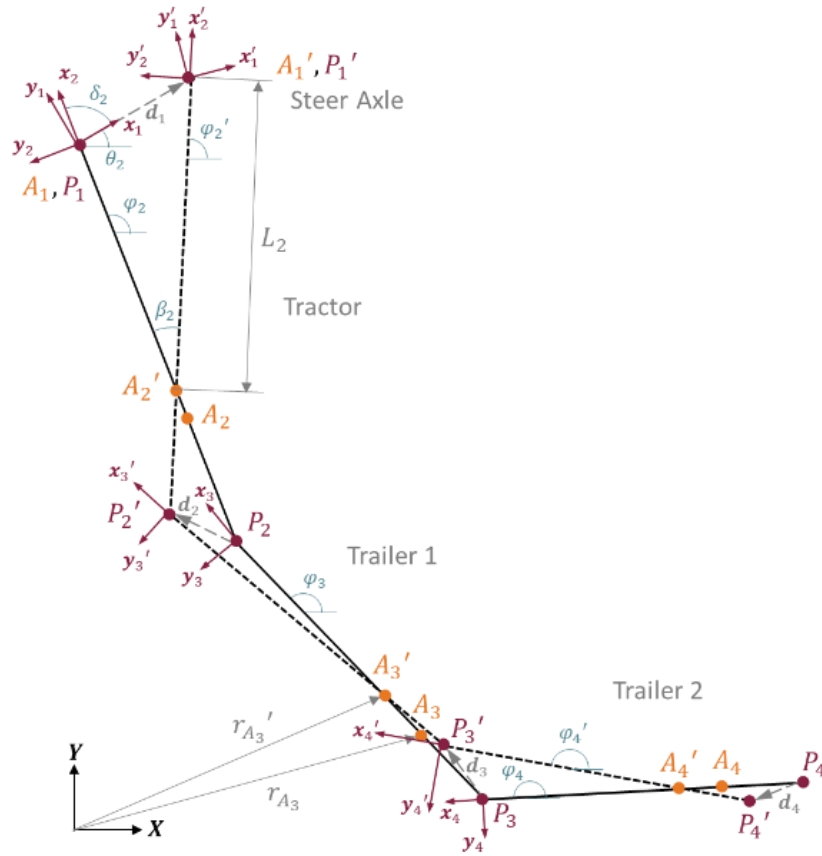


Figure 3-5. Illustration of SGM-A algorithm for a multi-unit vehicle.

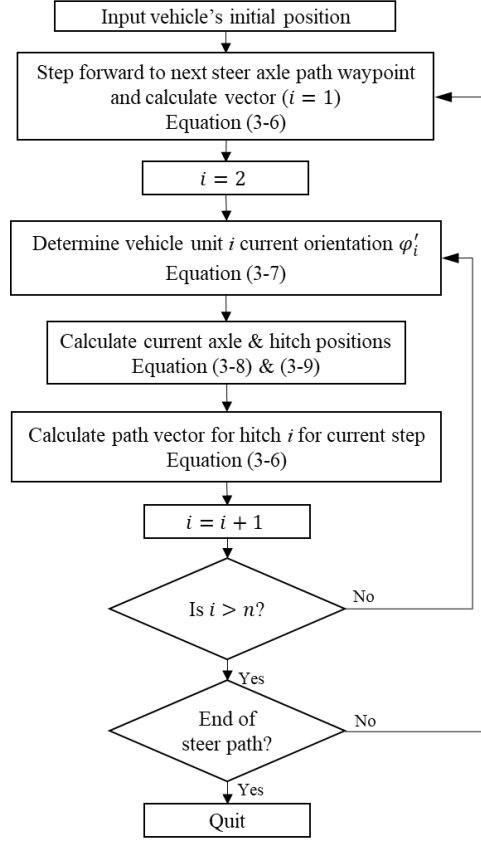


Figure 3-6. SMG-A algorithm flowchart.

Once the initial position of each axle and orientation of each vehicle unit is provided, the algorithm steps the steer axle forward from its previous position $\mathbf{r}_{A_1}$ to its current position $\mathbf{r}'_{A_1}$. The pivot the tractor articulates about the steer axle is defined as $\mathbf{r}_{P_1} = \mathbf{r}_{A_1}$ such that the vector $\mathbf{d}_1$ is determined by Equation (3-6). Next, the tractor's current orientation ϕ'_2 is determined by Equation (3-7). Using the current vehicle orientation ϕ'_i , the leading hitch position $\mathbf{r}'_{P_{i-1}}$, and the vehicle unit's wheelbase L_i , the current axle and hitch position are found by Equations (3-8) and (3-9), respectively. Lastly, the vector connecting the previous and current axle positions is computed by Equation (3-6). The algorithm moves to the next vehicle body $i+1$ and repeats Equations (3-6) through (3-9) in appropriate order. After the axle positions of all n vehicle bodies ($n-1$ vehicle units plus the steer axle) have been determined, the algorithm steps to the next steer axle path waypoint. This process is repeated until the steer axle has reached the end of its path and all axle positions have been computed.

$$\mathbf{d}_i = \mathbf{r}'_{P_i} - \mathbf{r}_{P_i} \quad (3-6)$$

$$\varphi'_i = \varphi_i - \sin^{-1} \left[\frac{\|\mathbf{d}_{i-1}\|}{L_i} \sin(\pi + \angle \mathbf{d}_{i-1} - \varphi_i) \right] \quad (3-7)$$

$$\mathbf{r}'_A = \mathbf{r}'_{P_{i-1}} - L_i \begin{bmatrix} \cos(\varphi_i) \\ \sin(\varphi_i) \end{bmatrix} \quad (3-8)$$

$$\mathbf{r}'_P = \mathbf{r}'_{P_{i-1}} - H_i \begin{bmatrix} \cos(\varphi_i) \\ \sin(\varphi_i) \end{bmatrix} \quad (3-9)$$

Simple Geometric Model – No Tire Slip Just Before Current Position (SGM-B)

Unlike SGM-A, SGM-B negates tire slip just before the vehicle's current position such that an axle's current position is along a line connecting its previous position and the current leading pivot position – i.e.: A'_2 exists along the line connecting A_2 and P'_1 in Figure 3-7. As shown in Figure 3-7, SGM-B uses the same nomenclature and coordinate frames as SGM-A in Figure 3-6, except $\mathbf{d}_i$ connects the i th unit's axle's previous position $\mathbf{r}_{A_i}$ to the $i - 1$ current hitch position $\mathbf{r}'_{P_{i-1}}$. Because $\mathbf{r}_{P_1} = \mathbf{r}_{A_1}$ and the steer axle path is the known input through the entire maneuver, the algorithm computes each vehicle unit's position sequentially from the tractor to the rearmost trailer for each step along the steer axle path. The vector $\mathbf{d}_i$ is determined by Equation (3-10), and the current axle and hitch positions $\mathbf{r}'_{A_i}$ and $\mathbf{r}'_{P_i}$ are determined by Equations (3-11) and (3-12), respectively, for each vehicle unit. The i th vehicle body's current orientation is $\varphi'_i = \angle \mathbf{d}_i$. The algorithm then steps to the next steer axle path waypoint and repeats Equations (3-10) through (3-12) for all vehicle units. The algorithm concludes once the steer axle has reached the end of the input path and all axle and hitch positions have been determined through the entire maneuver. Figure 3-7 illustrates the SGM-B algorithm for an LCV, and Figure Y presents the SGM-B flowchart with corresponding equations for each step.

$$\mathbf{d}_i = \mathbf{r}'_{P_{i-1}} - \mathbf{r}_{A_i} \quad (3-10)$$

$$\mathbf{r}'_{A_i} = \mathbf{r}_{A_i} + (\|\mathbf{d}_i\| - L_i) \begin{bmatrix} \cos(\varphi_i) \\ \sin(\varphi_i) \end{bmatrix} \quad (3-11)$$

$$\mathbf{r}'_{P_i} = \mathbf{r}_{A_i} + (\|\mathbf{d}_i\| - H_i) \begin{bmatrix} \cos(\varphi_i) \\ \sin(\varphi_i) \end{bmatrix} \quad (3-12)$$

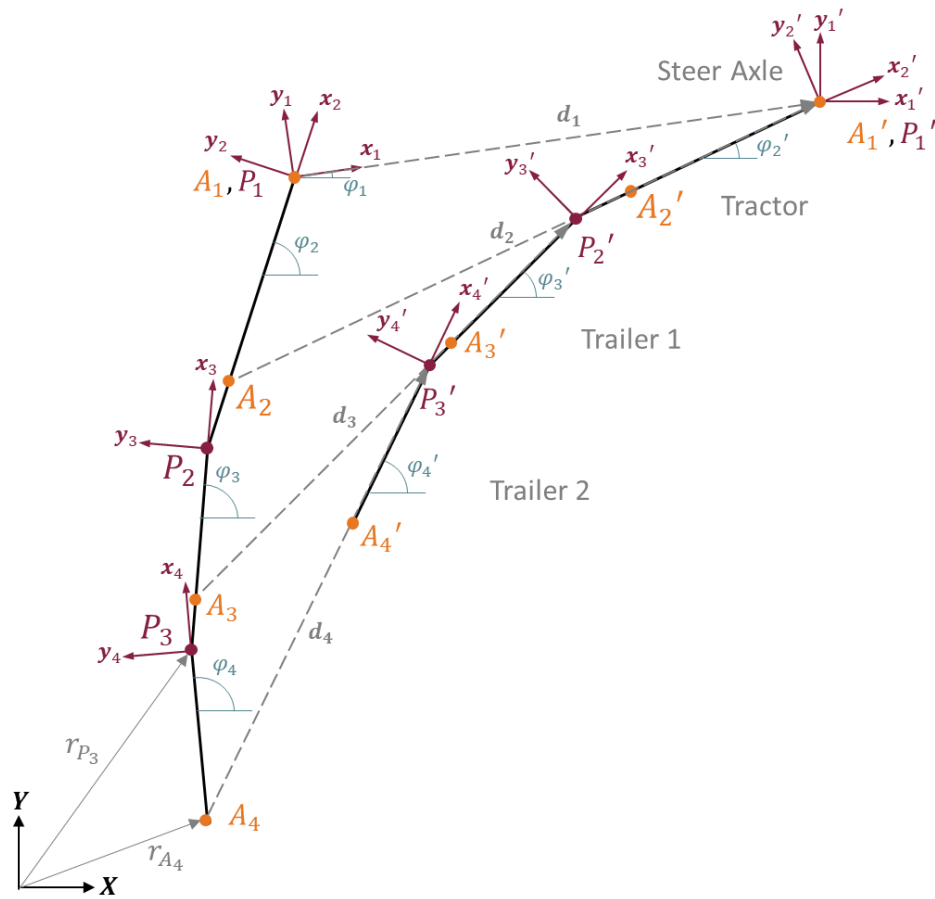


Figure 3-7. Illustration of SGM-B algorithm for a multi-unit vehicle.

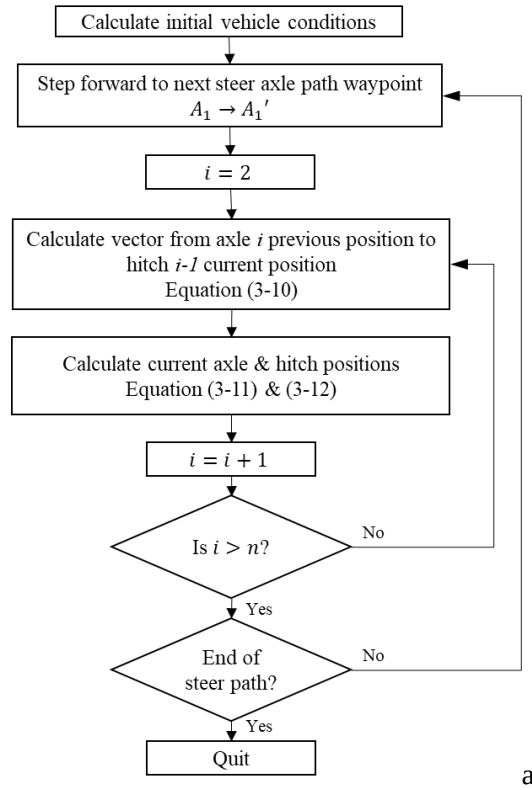


Figure 3-8. SGM-B algorithm flowchart.

3.1.5 Model Sensitivity to Input Path Resolution

Trailing axle path estimate accuracy of any of the numerical models – Sayers, SMG-A, SGM-B, SGM-C – is subject to the step size along successive waypoints. Peak off-tracking error from the continuous tractrix model’s estimate with respect to path step size is compared here among the three models by simulating a 33-ft A-double and sleeper-cab tractor through four unique maneuvers. The 33-ft double LCV configuration was selected because it incorporates multiple vehicle units, results in large amounts of off-tracking, and is of primary interest in comparison to other trailer configurations as detailed in Chapter 4. Table 3-1 lists the wheelbase – the distance from the trailer kingpin to the trailer axle center – and hitch length – the distance from the trailer kingpin to the hitch that the following trailer connects – for the simulated 33-ft A-double LCV. The maneuvers are a 50-ft radius J-turn, a 75-ft radius J-turn, a 50-ft radius circle, and compounded left-then-right 50-ft radius J-turns (S-turn). The 50-ft J-turn maneuver was selected because J-turns introduce significant amount of off-tracking and the 50-ft radius is commonly used as a minimum on U.S. roads [8,15]. The 75-ft J-turn maneuver was selected because it is geometrically similar to the 50-ft J-turn, but with a larger radius. The 33-ft A-double approaches but does not reach its steady-state (maximum) off-tracking in the 75-ft J-turn maneuver. The 50-ft radius circular

path was simulated to ensure steady-state (maximum) off-tracking was reached. In the circular maneuver, the vehicle had to pass through the full rotation of the maneuver more than once to reach maximum off-tracking. Figure 3-9 plots each maneuvers' steer axle path. The continuous tractrix model is approximated by Sayers' numerical model with step size of 0.4 in (0.01 m). For this step size, peak off-tracking is within 0.1% of the continuous model, making it acceptable to use the numerical algorithm with very small step size as the baseline representative of the continuous tractrix curves [5,12].

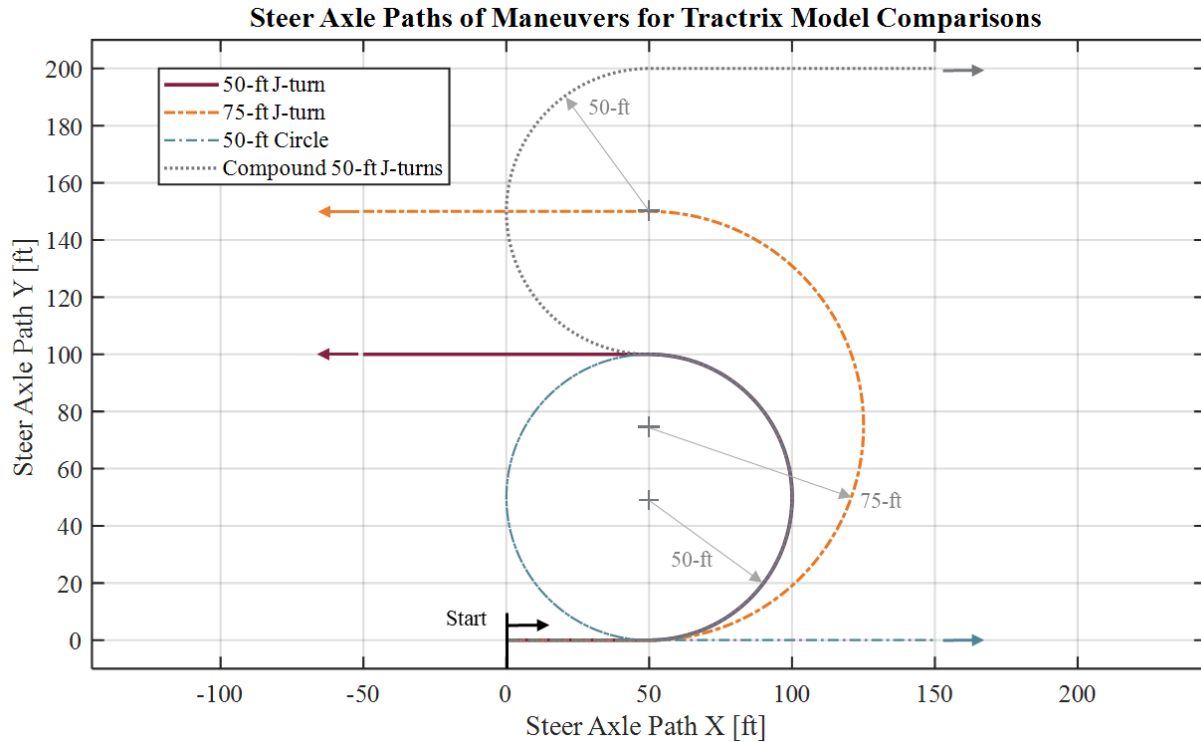


Figure 3-9. Simulated steer axle paths for tractrix model comparisons.

Table 3-1. Wheelbase and hitch length of simulated 33-ft A-double with sleeper cab tractor.

	Tractor	33-ft Trailer	A-Dolly
Wheelbase [ft]	21.25	28.29	6.04
Hitch Length [ft]	21.89	30.69	6.04

Figure 3-10 presents each numerical model's peak off-tracking percent error from the continuous tractrix model with respect to the input path step size for each of the four maneuvers. The compounded J-turns maneuver is divided into its 1st and 2nd turn. Unsurprisingly, SGM-A and SGM-B are most sensitive to

waypoint resolution for nearly all maneuvers. The SGM-A model under-estimated peak off-tracking and the SGM-B model over estimated peak off-tracking. Peak off-tracking found from the average path of the two simple geometric models is largely unaffected by the input path step size. Sayers' numerical tractrix model was sensitive to the path input resolution, but less than either SGM-A or SGM-B, except during the second turn of the compounded J-turns maneuver. This may be due to significant changes in arc length for a trailing axle between steer path waypoints as mentioned in Section 3.1.3. Sayers' numerical tractrix model also tends to under-estimate off-tracking.

All four numerical tractrix models – Sayers, SGM-A, SGM-B, SGM-C – exhibited a near linear relationship of peak off-tracking to input path resolution for each maneuver. The figure also includes percent error in peak off-tracking of the WHI formula – Equation (3-3). The WHI formula is independent of the steer axle path input because it represents the maximum off-tracking for a continuous curve and does not account for transient development of off-tracking from the beginning to end of a maneuver. For very large distances between input path waypoints (>5.5 ft), the WHI formula is more accurate than either SGM-A and SGM-B. During the 75-ft J-turn maneuver, the vehicle approaches steady-state off-tracking, and so the WHI formula estimates peak off-tracking within 1% error.

Because the 75-ft J-turn is geometrically similar to the 50-ft J-turn, the numerical models' peak off-tracking error in the 75-ft J-turn presented in Figure 3-10 trends similarly to the 50-ft J-turn. However, the percent error is slightly larger for the 75-ft J-turn than the 50-ft J-turn maneuver for all simulated numerical models. This is due to lower peak off-tracking for the larger radius turn that makes a small error a larger fraction of the peak off-tracking value.

Similar to the 50-ft J-turn and 75-ft J-turn maneuvers, SGM-A and SGM-B exhibited significant sensitivity to input path resolution for the continuous circular input path. SGM-C deviates slightly from the continuous tractrix model for large steer axle path step size, but was within 5% error even for a step size greater than 6 ft. Unlike the 50-ft J-turn and 75-ft J-turn, Sayers' numerical tractrix model was largely unaffected by the input path resolution. This coincides with one of the model's primary assumptions: that the path traveled between successive waypoints is circular. In this maneuver, the trailing axle paths are approximately circular once the vehicle nears and reaches its steady-state, maximum off-tracking. Lastly, because the steady-state, maximum off-tracking was reached, the WHI formula's off-tracking estimated is indistinguishable to the continuous tractrix model's peak off-tracking.

The last simulated maneuver is the compounded J-turns – also referred to as S-turn – maneuver. The peak off-tracking for this maneuver is evaluated for the first and second turn separately. The first turn is identical to the 50-ft J-turn maneuver except that the exit path curves outward away from the turn rather than straightaway tangent to the curve. The second turn is also similar to the 50-ft J-turn, except in opposite direction and with different initial conditions – no straight entrance and the vehicle is already off-tracking in the opposite side. These conditions effectively reduce peak off-tracking for the two turns in the S-turn as compared to the single 50-ft J-turn. Similar to the other maneuvers, each model's peak off-tracking percentage error is approximately proportional to the input path resolution. The magnitude for all data points is just less than the magnitude of their corresponding data points in the 50-ft J-turn maneuver. SGM-C proved to be the least sensitive to input path resolution. Interestingly, SGM-C outperformed Sayers' numerical tractrix model for medium and large input path resolution. However, the other models exhibited similar relationships as found in the first turn and the 50-ft J-turn and 75-ft J-turn maneuvers.

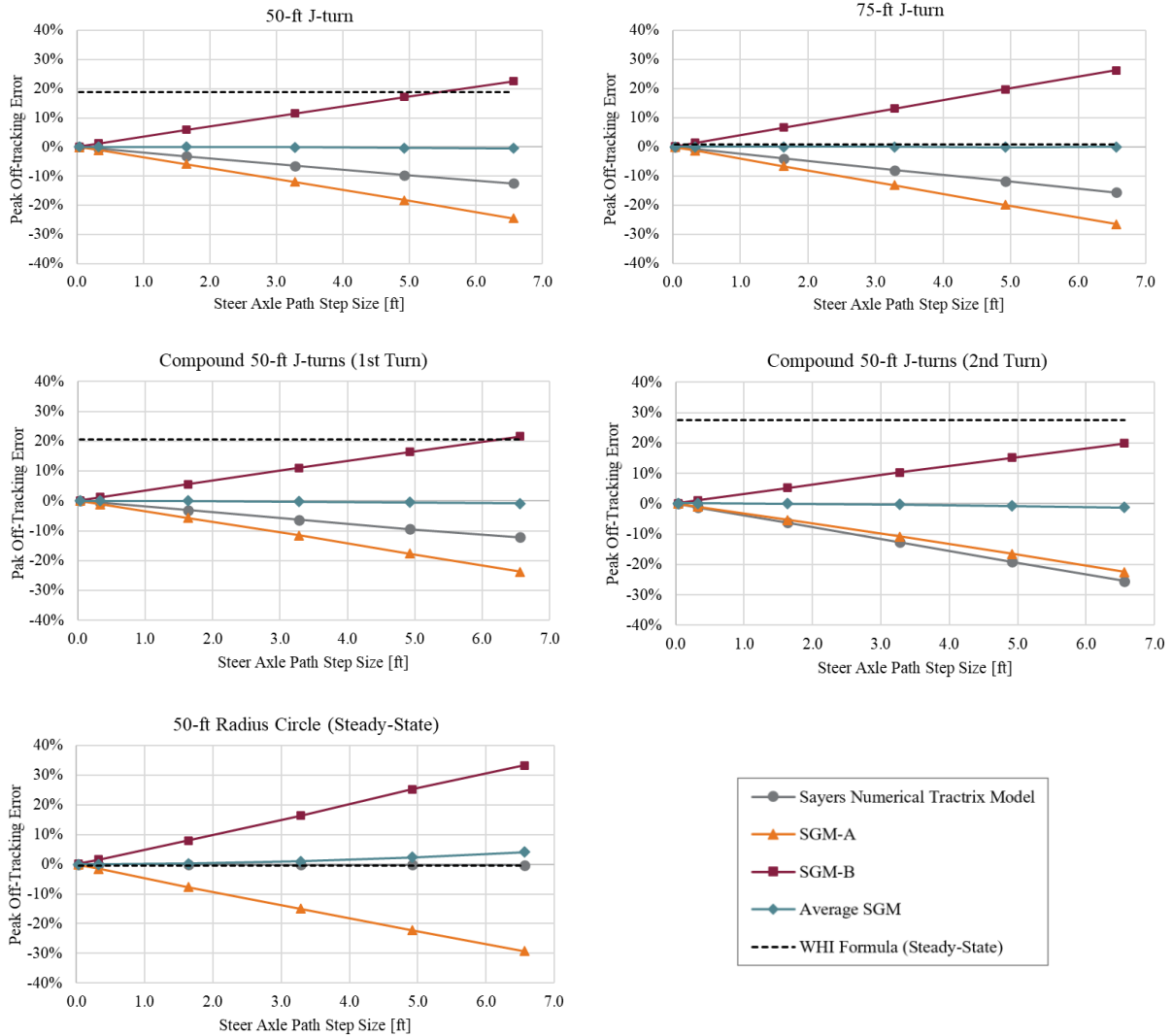
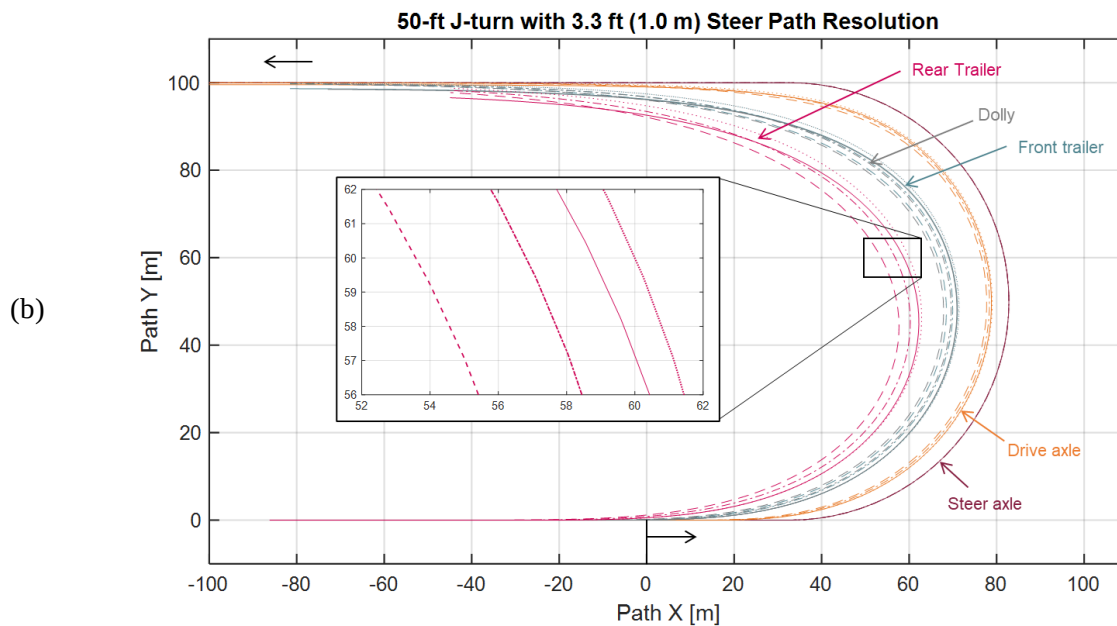
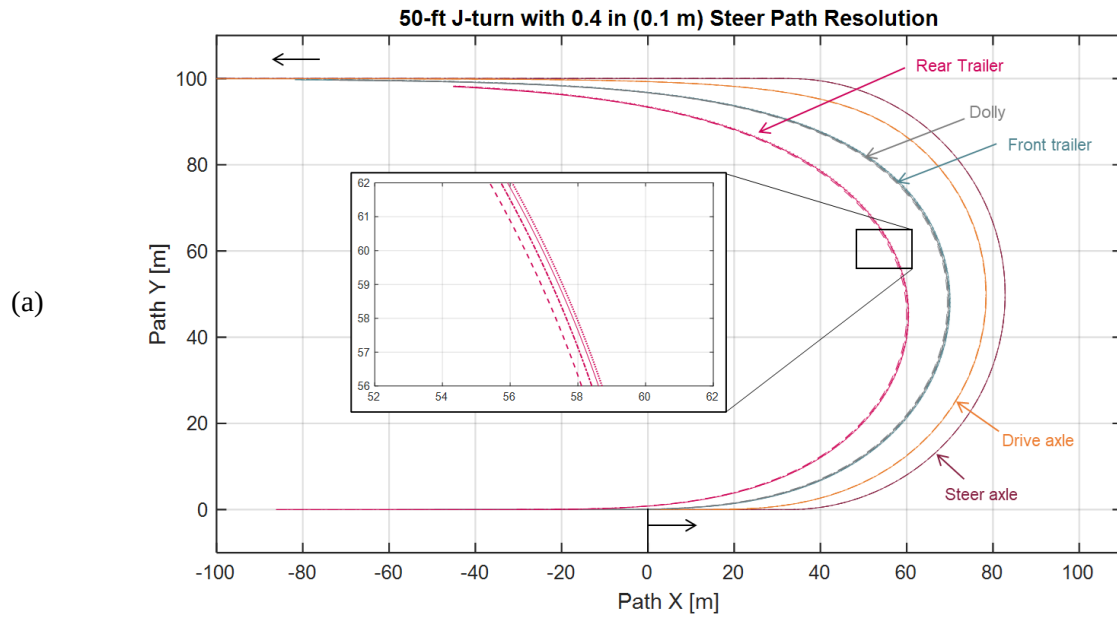


Figure 3-10. Percent error of each kinematic model's peak off-tracking estimate from the continuous tractrix curves for each of the simulated maneuvers with the 33-ft double trailer configuration.

Figure 3-11 demonstrates the models' axle paths for two example input path resolutions – (a) 4 in (0.1 m) and (b) 3.3 ft (1.0 m) – for the 50-ft J-turn maneuver. With 4 in input path resolution, the difference of rearmost axle path among the four numerical models is almost indistinguishable from each other unless examined closely. For the 3.3 ft input path resolution, the differences are clearly observed for all trailing axles, including the tractor drive axle. Differences between models are generally amplified by an increase in off-tracking, or equivalently, trailing axles further from the steer axle. Figure 3-12 presents off-tracking through the 50-ft J-turn maneuver for each model and the two example input path resolutions (4 in and 3.3 ft). Differences in off-tracking are more clearly visible than differences in the axle paths for low-input path resolution.



— Sayers' tractrix model - - SGM-A SGM-B — · — Average SGM

Figure 3-11. Kinematic model comparison for A-double 33-ft trailers axle paths through 50-ft radius J-turn with (a) 4 in (0.1 m) and (b) 3.3 ft (1.0 m) steer axle path resolution.

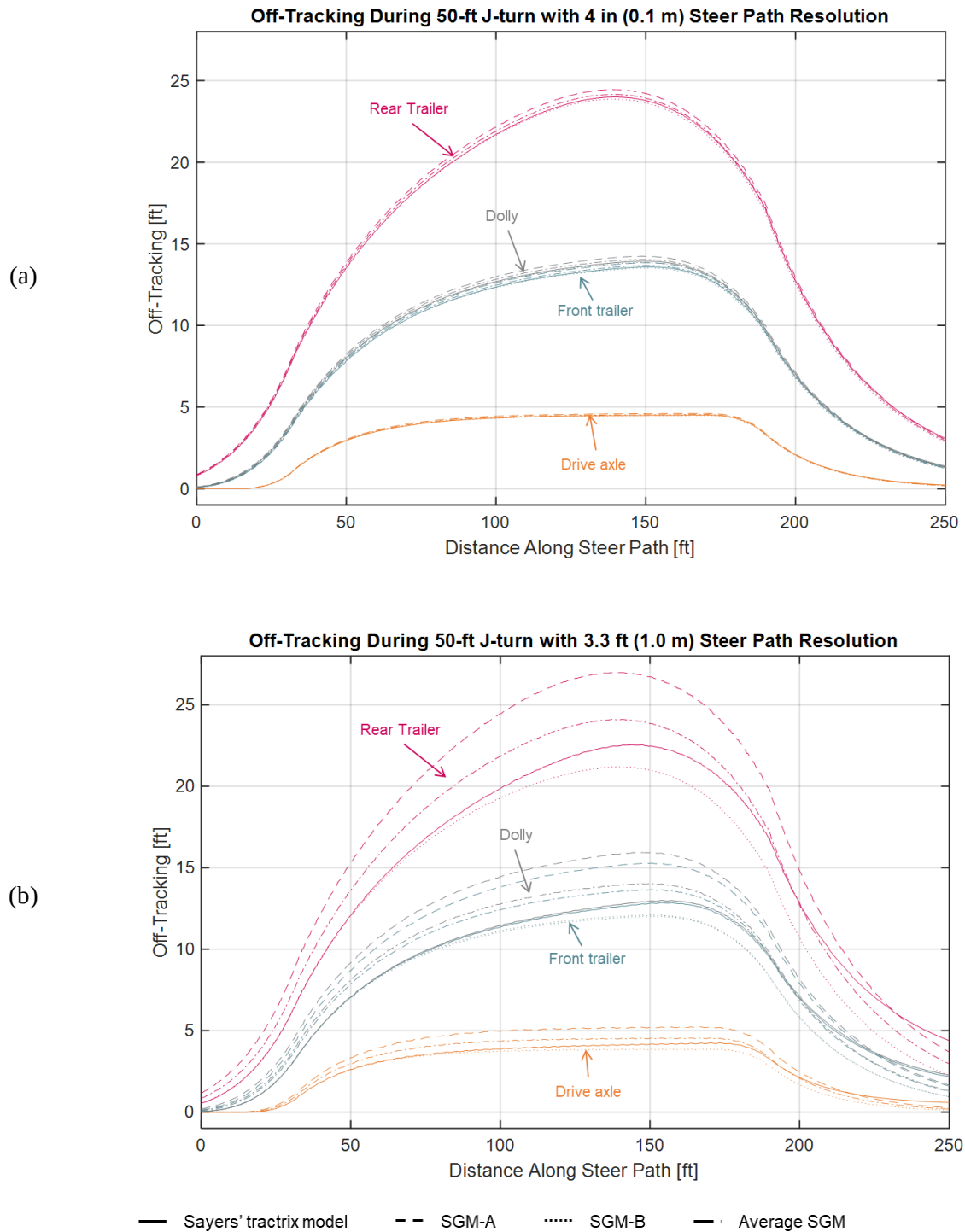


Figure 3-12. Kinematic model comparison for A-double 33-ft trailers off-tracking through 50-ft radius J-turn with (a) 4in (0.1m) and (b) 3.3ft (1.0m) steer axle path step size.

Figure 3-13 and Figure 3-14 demonstrate the models' axle paths and off-tracking through the S-turn maneuver, respectively. Off-tracking is always a positive quantity, and where it becomes zero in the middle of the compound J-turn maneuver is where the trailers move from the left to the right of the steer axle path.

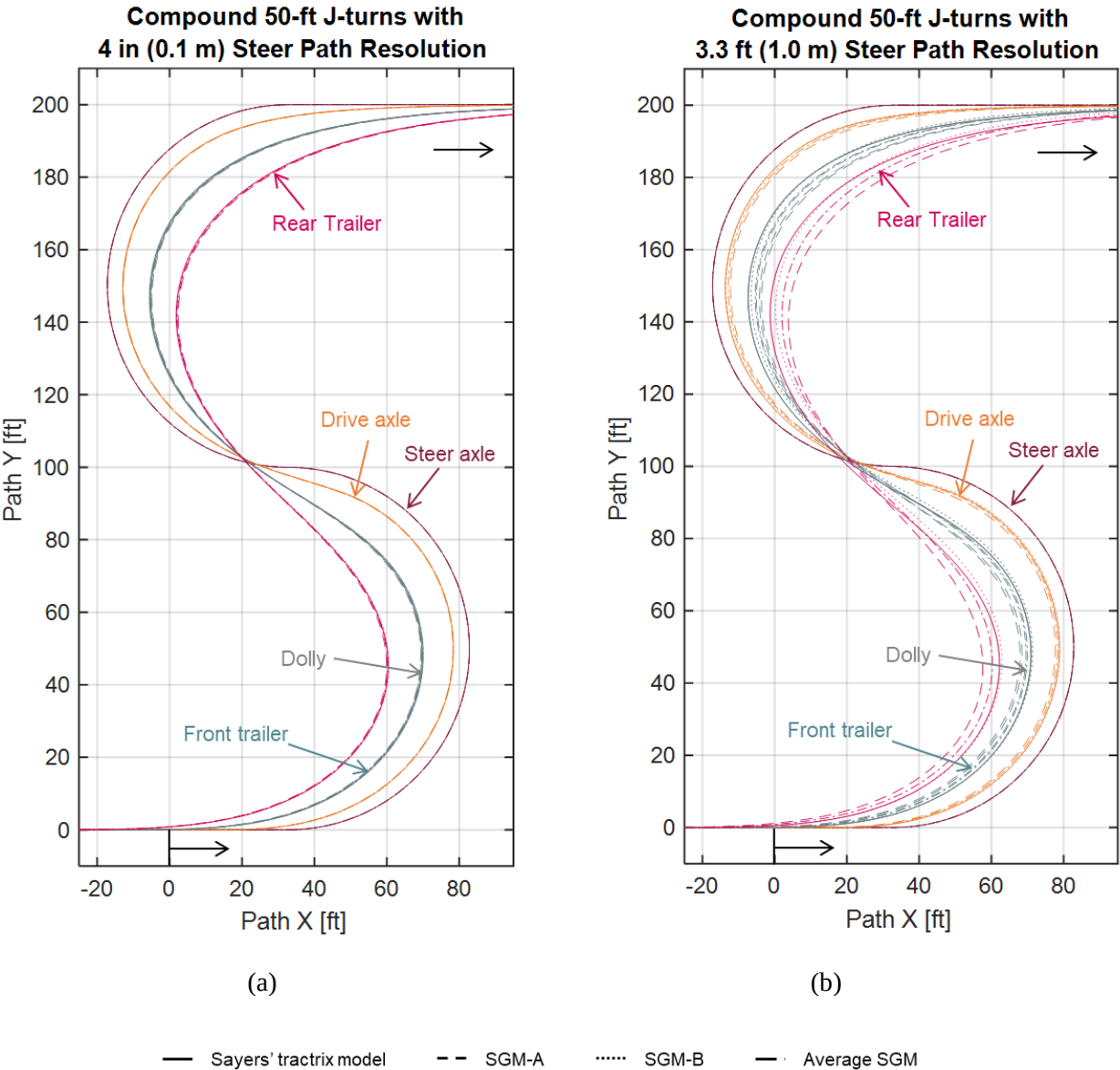


Figure 3-13. Kinematic model comparison for A-double 33-ft trailers axle paths through compound 50-ft radius J-turns with (a) 4 in (0.1 m) and (b) 3.3 ft (1.0 m) steer axle path resolution.

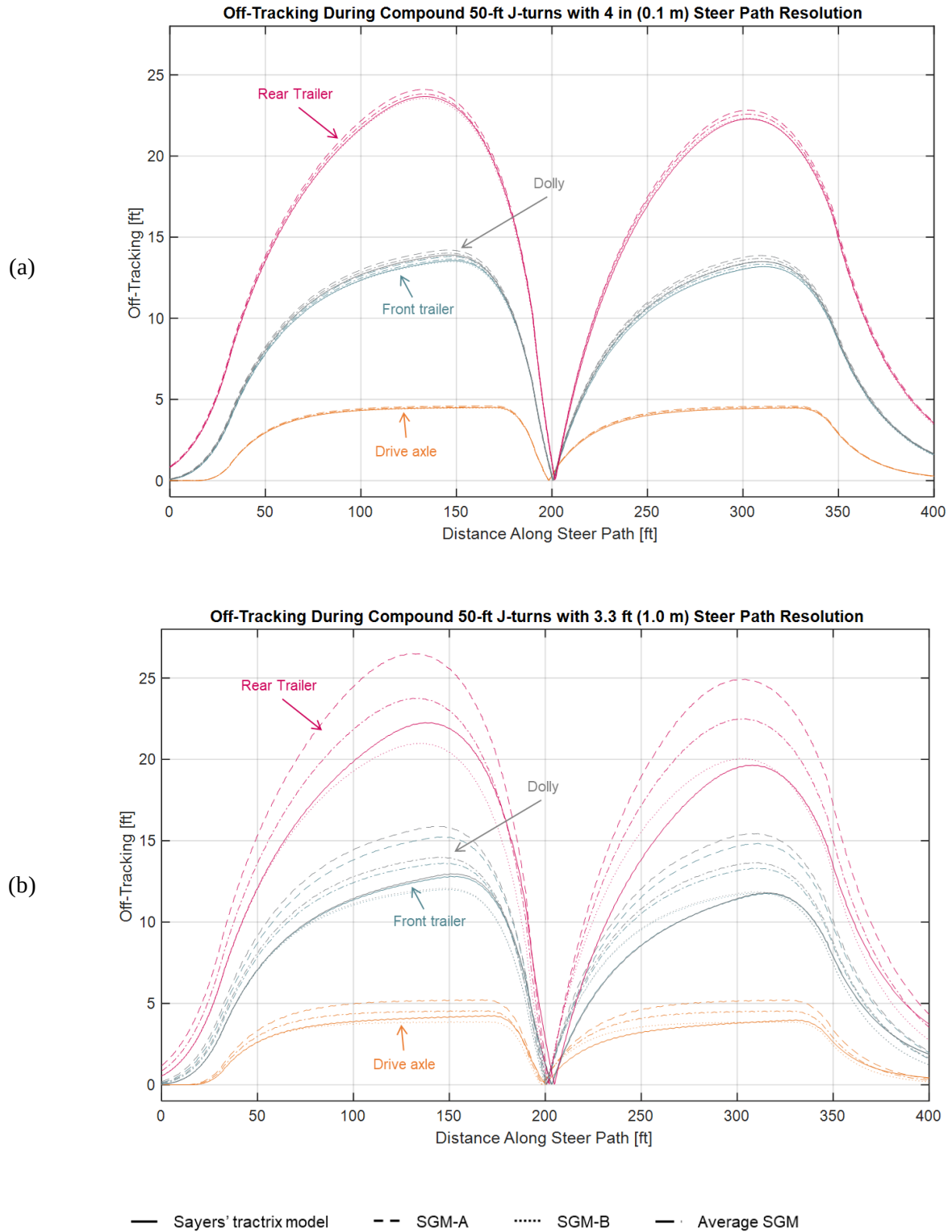


Figure 3-14. Kinematic model comparison for A-double 33-ft trailers off-tracking through compound 50-ft radius J-turns with (a) 4 in (0.1 m) and (b) 3.3 ft (1.0 m) steer axle path step size.

In summary, the proposed models closely match the continuous tractrix model for fine resolution of the input steer axle path. As the input steer axle path resolution increases (larger step size), all three geometric models deviate from the tractrix curves, two of which (SGM-A and SGM-B) deviate more than Sayers' model. For a single turn and initially zero articulation angle, SGM-B over-estimates off-tracking while the SGM-A under-estimates off-tracking. SGM-A matches Sayers' numerical tractrix model slightly better than SGM-B for all simulated input path resolution. SGM-C is largely unaffected by the input path resolution. The input path resolution does not impact whether a model over- or under-estimates off-tracking for a single curve, instead, the maneuver itself will play a greater role in whether a model over- or under-estimates off-tracking, as exemplified by Sayers' numerical tractrix model's path through the S-turn. The WHI formula matches the continuous tractrix model well only for the steady-state off-tracking condition and the 75-ft J-turn where the simulated vehicle configuration nears its maximum off-tracking for that curve radius.

3.1.6 Trailer Configurations with Equivalent Steady-State Off-Tracking

Section 3.1.5 demonstrated that SGM-A, SGM-B, and SGM-C are good approximations of tractrix curves for estimating LCV axle paths and off-tracking so long as the input path resolution is sufficiently small. Additionally, the SGM-C was least sensitive to the input path resolution. Three different hypothetical tractor-trailer configurations will be introduced here for comparison among a range of vehicle units: a single semitrailer, an A-train triple, and a tractor without trailers. This comparison is conducted to inspect if the path of an LCV may be represented by some other LCV with more or fewer vehicle units and different dimensions. Reducing an LCV to a single truck within the model is of particular interest because it significantly reduces the computational effort of the numerical tractrix models. The hypothetical vehicle configurations are dimensioned to result the same steady-state off-tracking as the 33-ft A-double according to the WHI formula under the following assumptions: configurations with trailers are towed by the same tractor, converter dollies are unchanged, and trailer hitch offsets are the same as the 33-ft trailers. The WHI formula may be re-arranged into Equation (3-13) to determine the trailer length for a given input path radius R_f , steady-state off-tracking OT_{ss} , and number of equivalent trailers N_t . The subscripts tcr and tlr correspond to the tractor and trailers, respectively.

$$L_{tlreq} = \left[\frac{1}{N_t} \left(R_f^2 - L_{tcr}^2 + HO_{tcr}^2 + (N_t - 1)(HO_{tlr}^2 + HO_{dolly}^2 - L_{dolly}^2) - (R_f - OT_{ss})^2 \right) \right]^{\frac{1}{2}} \quad (3-13)$$

Similarly, the theoretical wheelbase of an equivalent tractor not towing any trailers may be determined to result the same steady-state off-tracking as another tractor-trailer configuration. This equivalent tractor wheelbase L_{tcreq} is found as Equation (3-14).

$$L_{tcreq} = [2R_f OT_{ss} - OT_{ss}^2]^{\frac{1}{2}} \quad (3-14)$$

According to Equation (3-14), an equivalent tractor wheelbase representative of the A-double 33-ft steady-state off-tracking is 45.10 ft. From Equation (3-13), a single semitrailer with wheelbase 40.01 ft, an A-triple with 22.67 ft trailer wheelbases, and an A-double with 28.03 ft trailer wheelbases all result the same steady-state – maximum – off-tracking for any input path radius while towed by the same tractor. Although not directly apparent in Equations (3-13) and (3-14), the equivalent wheelbases are independent of the input path radius. All four configurations – single tractor, single semitrailer, A-double, and A-triple – were simulated through the four test maneuvers detailed previously – 50-ft J-turn, 75-ft J-turn, constant-radius circle, and S-turn – for comparison in transient off-tracking. The input path resolution was 0.4 in (0.01 m) and SGM-C was used to best represent the continuous tractrix curves. At this input path step size, however, any numerical tractrix model would be appropriate as demonstrated in the previous section. Table 3-2 lists the vehicle dimensions for each simulated vehicle configuration. The summation of all vehicle unit wheelbases and all hitch offsets that link two vehicle units is also included. This value is indicative of the total vehicle length of interest for evaluating off-tracking at the rearmost axle. In all simulated configurations, the hitch offsets are positive because the hitch is located behind the rear axle of the towing unit.

Table 3-2. Equivalent tractor-trailer dimensions for steady-state off-tracking that matches 33-ft A-double with sleeper-cab tractor.

	Tractor		Trailer		Dolly		Summation of Wheelbases & Hitch Offsets [ft]
	Wheelbase [ft]	Hitch Offset [ft]	Wheelbase [ft]	Hitch Offset [ft]	Wheelbase [ft]	Hitch Offset [ft]	
Tractor only	45.10	-	-	-	-	-	45.10
Single	20.83	0.63	40.01	-	-	-	61.47
A-double	20.83	0.63	28.03	2.61	6.04	0.0	86.16
A-triple	20.83	0.63	22.67	2.61	6.04	0.0	106.75

Figure 3-15 lists the percent difference in peak off-tracking for each hypothetical configuration compared to the A-double with 33-ft trailers. As expected, the percent difference is lowest for maneuvers where the vehicle combinations reach or approach their steady-state off-tracking, such as in the 50-ft radius circle (not shown because there is 0% difference for all configurations) and 75-ft J-turn maneuvers. Peak off-tracking differences among the configurations were most evident in the compound J-turn’s second turn. This is not surprising because off-tracking is constantly changing significantly throughout the maneuver and the initial conditions of the second turn differ between configurations due to having different off-tracking at the first turn’s exit.

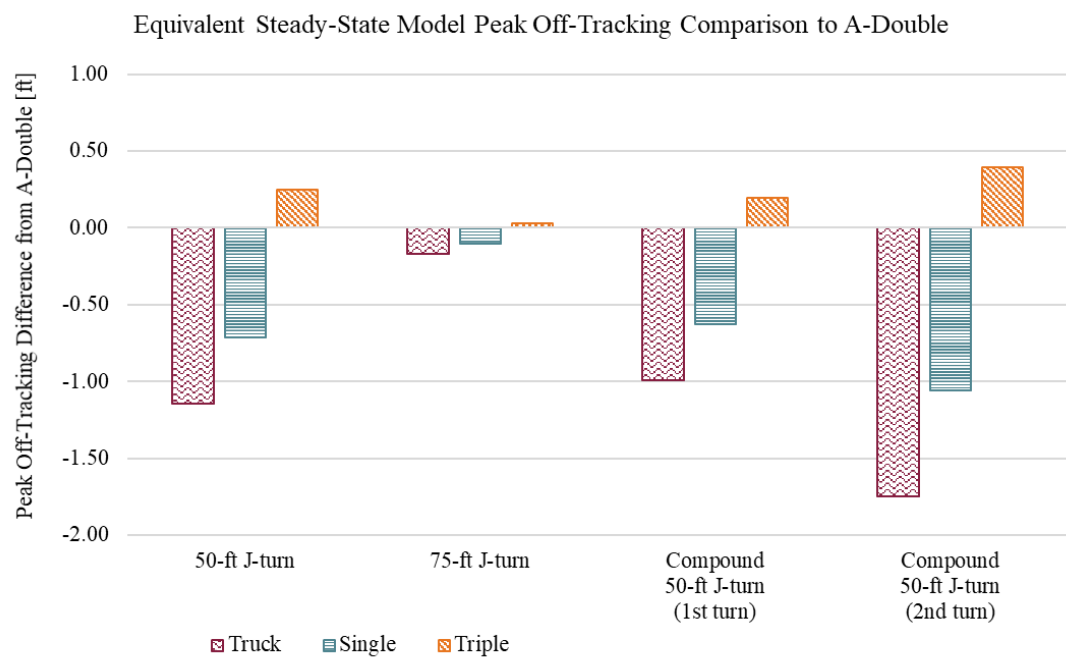


Figure 3-15. Equivalent steady-state model peak off-tracking difference from A-double.

Interestingly, the hypothetical A-triple configuration resulted in peak off-tracking within just 2% of the A-double for all simulated maneuvers. It is expected that as the number of trailers increases while wheelbases decrease to accommodate equal steady-state off-tracking, the peak off-tracking percent difference between configurations of successively more trailers converges until the model effectively represents a continuous rope as the number of trailers approaches infinity. For the data shown, trailers on the A-triple configuration included the same hitch offset as the trailers on the A-double configuration. The simulations were also run with an A-triple configuration where the trailers did not have a hitch offset. These trailers had a wheelbase of 22.57 ft to match the A-double steady-state off-tracking, and their peak off-tracking was within 0.1% of the A-triples shown in Table 3-2 and Figure 3-15. The hitch offset is sufficiently small relative to the

trailer wheelbase such that either configuration may be used for comparison. For the remainder discussion, the A-triple configuration includes a hitch offset as detailed earlier in Table 3-2.

Figure 3-16 and Figure 3-17 plot the rearmost axle path and off-tracking of each hypothetical configuration for the 50-ft J-turn maneuver, respectively. Both figures illustrate that as the number of vehicle units increase, the rearmost axle off-tracking becomes more sensitive to changes in the steer axle path. For example, at the maneuver entry, the A-triples have the most off-tracking. They are also the first to reach peak off-tracking through the turn and are first to reduce off-tracking as the lead unit exits the maneuver. The order of peak off-tracking magnitude and position along the steer axle path is as follows: A-triple, A-double, single semitrailer, tractor only.

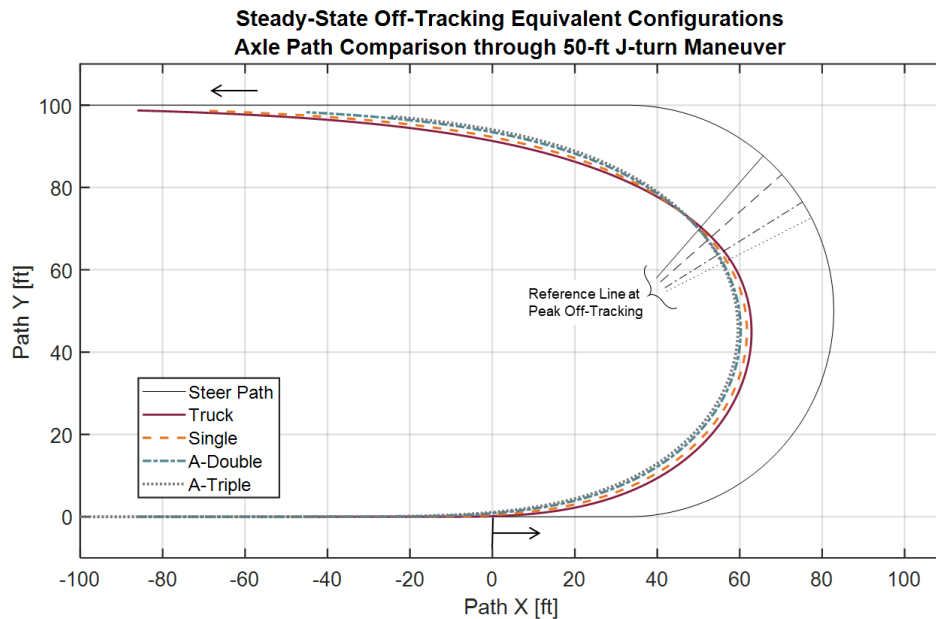


Figure 3-16. Axle paths of steady-state off-tracking equivalent configurations through the 50-ft J-turn maneuver.

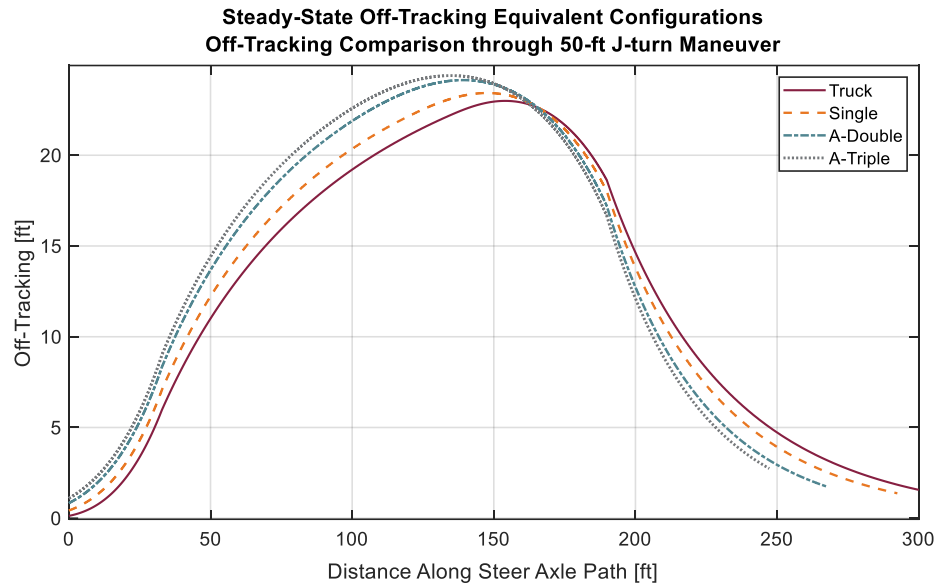


Figure 3-17. Off-tracking of steady-state off-tracking equivalent configurations through the 50-ft J-turn maneuver.

Figure 3-18 and Figure 3-19 plot the axle paths and off-tracking of each hypothetical configuration for the compound J-turn maneuver, respectively. Similar findings from the 50-ft J-turn are demonstrated. A higher number of vehicle units and articulation points result quicker changes in off-tracking than fewer vehicle units. Notice that the truck and single semitrailer off-tracking continue to deviate from the A-double and A-triple configurations through the maneuver. In short, the longer wheelbases found in the single truck and tractor-semi-trailer effectively act as a low-pass filter to the steer axle path.

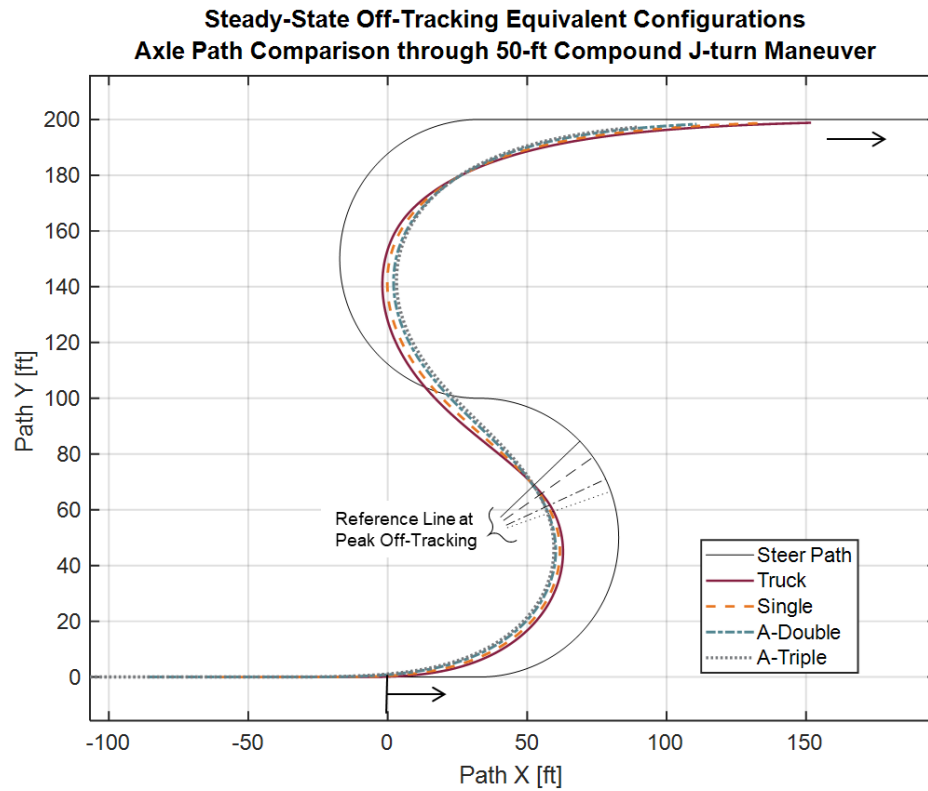


Figure 3-18. Axle paths of steady-state off-tracking equivalent configurations through the 50-ft Compound J-turn maneuver.

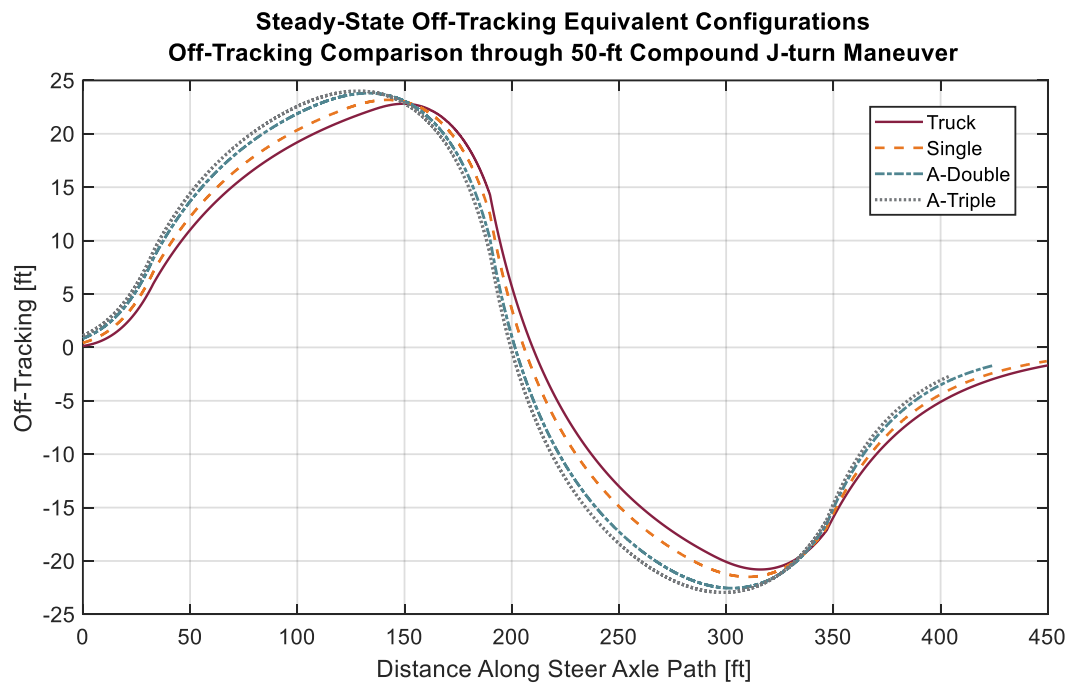


Figure 3-19. Off-tracking of steady-state off-tracking equivalent configurations through the 50-ft Compound J-turn maneuver. Needs to be positive.

Although all simulated vehicle configurations result the same maximum (steady-state) off-tracking for a given steer path radius, the configurations with fewer units are slower to approach and reach the peak off-tracking than the configurations with more units. However, the configurations with more units have a total length greater than the configurations with fewer units as presented by the combined wheelbase and hitch lengths in Table 3-2. Therefore, although the A-double and A-triple may reach more off-tracking with respect to a given point along the steer axle path as compared to the single semitrailer and single truck, the longer vehicle units require the steer axle to travel further along the steer path for the trailers to reach that point in space. The difference in location of peak off-tracking and the corresponding steer axle position along the steer axle path among the different trailer configurations is related to the steer axle path trajectory. There are an infinite number of possible steer axle paths and hypothetical vehicle configurations to compare, hence the presented investigation may not necessarily provide a general solution. The maneuvers selected and compared previously, however, are sufficient to demonstrate how the number of vehicle units and articulation points can influence the rate of change of off-tracking with respect to the steer axle's distance traveled along its path.

3.2 Kinematic Models Compared to TruckSim® High-Order Dynamic Model

This section compares the SGMs to the TruckSim® dynamic model for four unique maneuvers. The goal of this section is to determine if the tractrix model assumptions are acceptable for the desired maneuvers, test equipment, and test speed, and which of the two models, or their average, should be used for processing field test data detailed in Chapter 4. Sayers' tractrix model is omitted here because it will not be considered as part of analyzing field test data. This is because it requires the steer path input to be a series of arc lengths and curvature rather than cartesian coordinates. For the smooth curves resulting from simulations, this is trivial. But, when using noisy test data the conversion would result in numerical errors. The same vehicle dimensions listed in Table 3-1 were used in the simulation. The simulated vehicle parameters were chosen to match the field test equipment. Default component models were used if more specific models could not be determined for the test equipment. The implemented tire model was taken from [28]. In each simulated maneuver, the vehicle held a constant 2.8 mph speed, the simulated roadway was flat, and the assumed road surface friction coefficient was 0.85.

3.2.1 Maneuvers

Four unique maneuvers were simulated in TruckSim® to compare axle paths to those determined by the proposed simple geometric models. The four maneuvers are a 50-ft radius J-turn, 75-ft radius J-turn, left turn through a roundabout (~60-ft radius), and a right turn through an intersection (~50-ft radius). The

steer axle path for both J-turn maneuvers simulated in TruckSim® differ from those used in the geometric model comparisons. In TruckSim®, the steer axle path was determined such that a point ahead of the steer axle traced out the simple J-turns discussed in the previous subsection. These paths differ to facility ease of laying and following a path during field tests as well as provide an appropriate steer axle path for data processing that will be detailed in Chapter 4. Just as before, the 50-ft radius J-turn was chosen because it is representative of commonly used minimum radius for designing roadway turns [8,15]. The 75-ft radius J-turn is geometrically similar to the 50-ft J-turn and is expected to result noticeably less off-tracking and allow the 33-ft A-double tractor-trailer to approach its maximum off-tracking for that curve radius. The steer axle path for both maneuvers is shown in Figure 3-20. The roundabout and urban intersection maneuvers introduce more complex steering as they require the tractor to steer both left and right. With approximately 60-ft radius, the roundabout is considered as a medium sized urban roundabout [15,16]. The intersection maneuver mimics a driver turning right through an intersection while being cognizant of the vehicle's off-tracking to prevent the rearmost trailer from colliding with some object (traffic light post, sidewalk, etc.) inside of the turn. Figure 3-20 and Figure 3-21 present the steer axle path for each of the maneuvers.

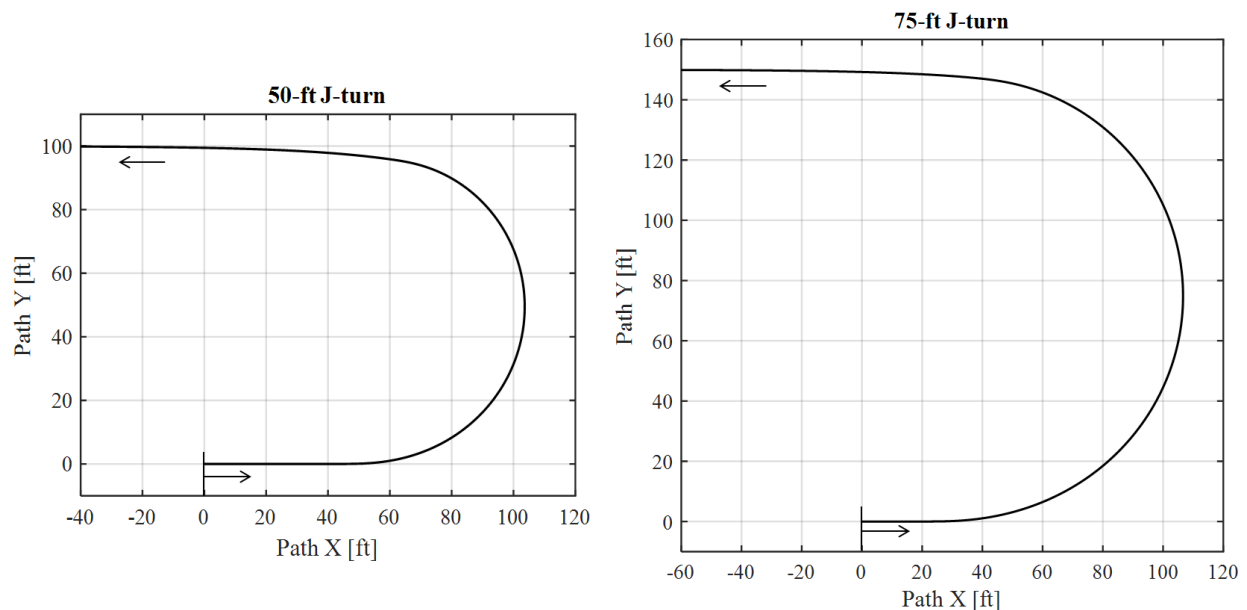


Figure 3-20. Steer axle path for the 50-ft and 75-ft J-turn maneuvers.

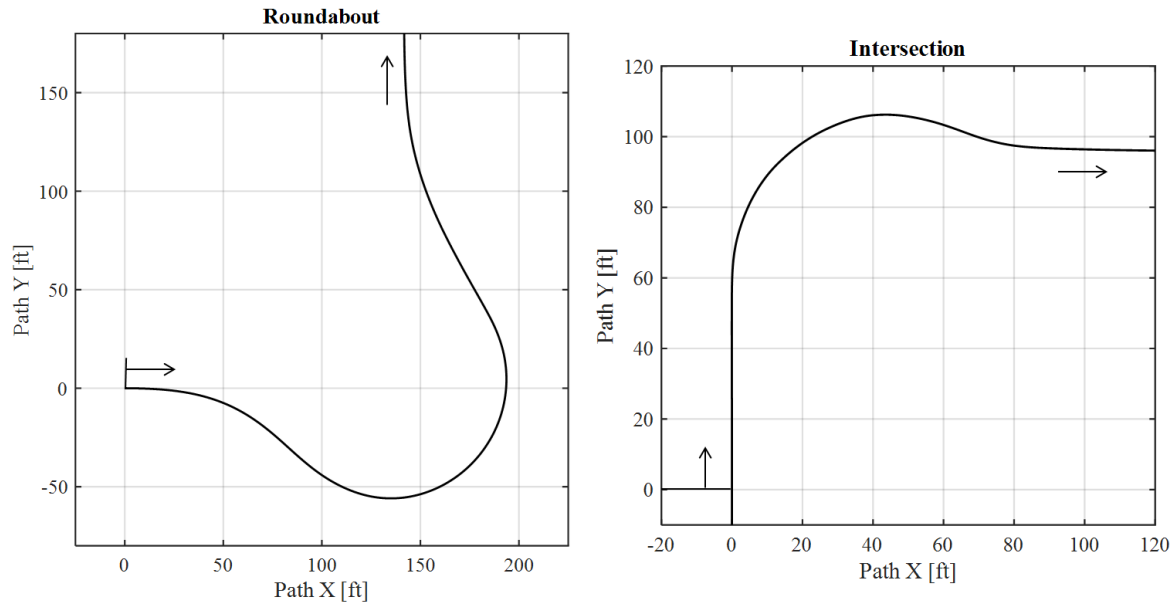


Figure 3-21. Steer axle path implemented in TruckSim simulations and field testing.

3.2.2 Peak Off-Tracking Comparison

Using the same steer axle path from the TruckSim® simulations as the input path to the simple geometric models, peak off-tracking of the rearmost trailer axle is compared from each model relative to the TruckSim® simulation for each maneuver. As demonstrated earlier, the geometric models converge as the step size decreases, but a small step size may not be representative of experimental data from field tests. Thus, a step size must be chosen to bridge the gap from simulation to field testing in Chapter 4. The steer axle speed during field tests ranged from approximately 2.4 to 3.0 mph, but was most often around 2.8 mph. The steer axle position and speed were sampled at 5 Hz by a GPS receiver. For a 5 Hz sample rate at 2.8 mph, the distance between each sample is approximately 9.8 in. The blend of 15 unique trials conducted for each field test maneuver permits assuming significantly smaller distances between successive steer axle path waypoints. Thus, a 4 in (0.1 m) step size was selected for the simple geometric models as it is acceptably large to correlate with field test data while sufficiently small such that the geometric models are representative of tractrix curves. Of course, the TruckSim® model's temporal and spatial sampling had significantly better resolution for accurate dynamics simulation.

Figure 3-22 presents the peak off-tracking error of each model relative to the TruckSim® simulation for the four maneuvers from the rearmost trailer axle. For all trailing units – the front trailer, dolly, and rear trailer – SGM-C's path resulted in a peak off-tracking within 0% to -0.5% difference from the TruckSim

model for all four maneuvers as shown in Figure 3-23. To put this in another context, the maximum peak off-tracking difference of the average SGM to the TruckSim® model was within ¼ in.

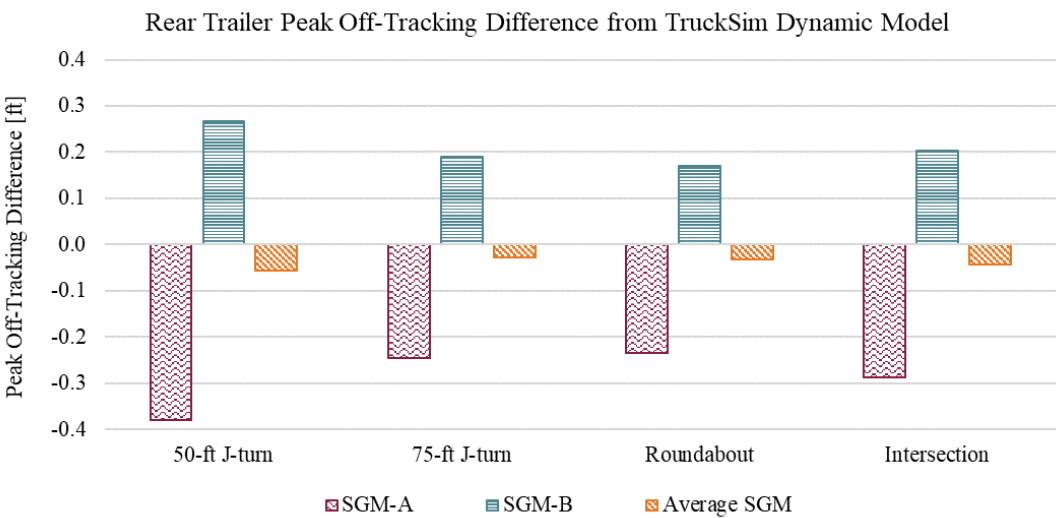


Figure 3-22. Peak off-tracking difference from numerical tractrix models to TruckSim® dynamic model.

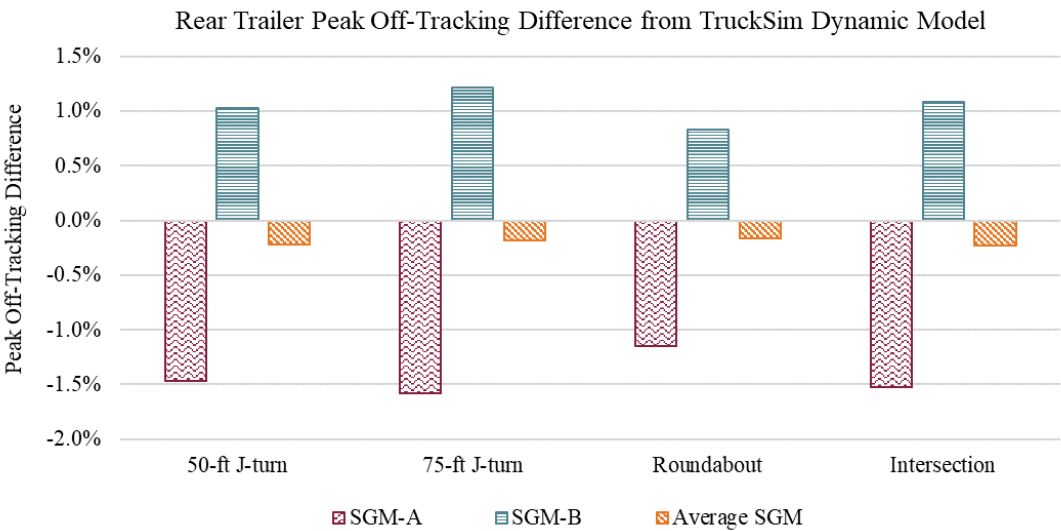


Figure 3-23. Percent of peak off-tracking difference from numerical tractrix models to TruckSim® dynamic model.

3.3 Off-Tracking Comparison of Common Trailer Configurations

As demonstrated in the previous sections, SGM-C is the least sensitive to steer path discretization and matched the dynamic model simulation best through the real-world low-speed maneuvers. In this section,

SGM-C will be applied to two additional vehicle configurations common to U.S. roads – a single 53-ft semitrailer and an A-double with 28-ft trailers – through the same four test maneuvers – 50-ft J-turn, 75-ft J-turn, roundabout, and intersection. Figure 3-24 and Table 3-3 provide the tractor-trailer dimensions for each test configuration, including the A-double with 33-ft trailers discussed in previous sections. Figure 3-24 illustrates the different combinations approximately to scale in length to provide a visual comparison in the overall vehicle lengths. The same tractor is used for each configuration.

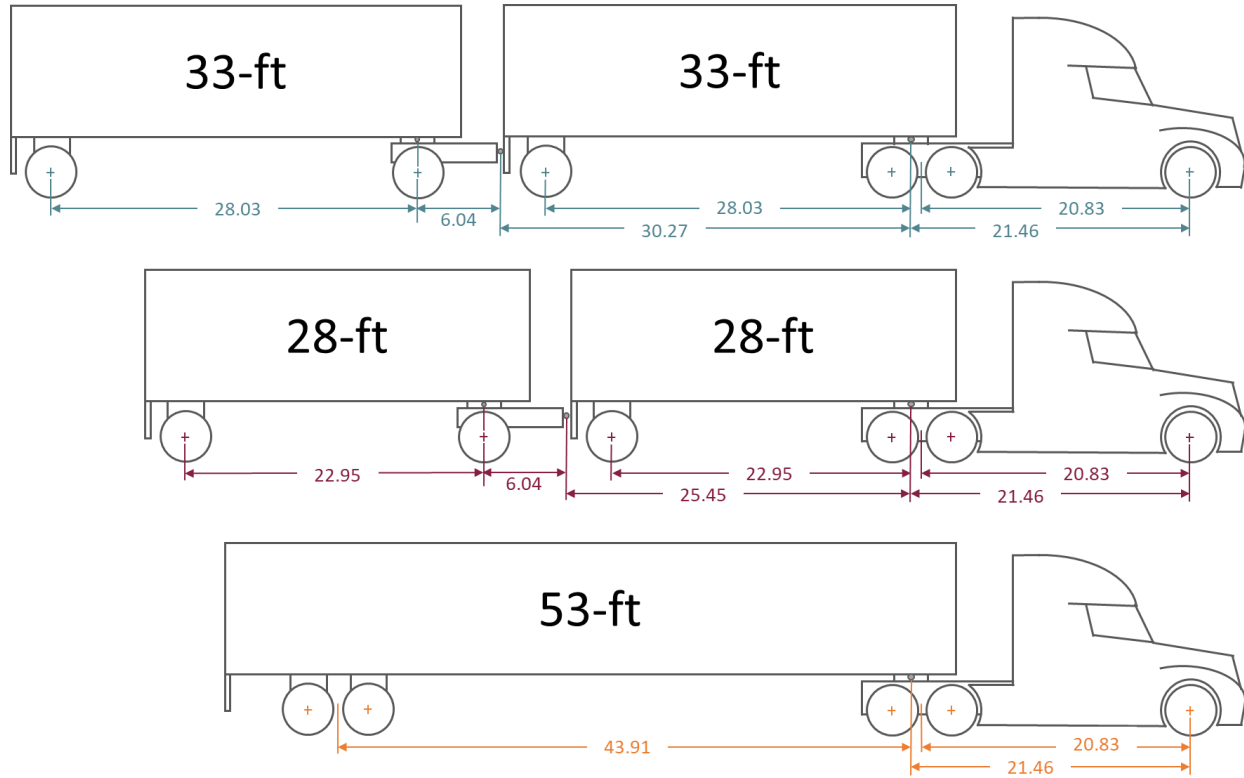


Figure 3-24. Illustration of three tested trailer configurations. The trailer lengths are scaled relative to each other to provide a visual representation of their relative lengths.

Table 3-3. Test tractor, trailers, and dolly wheelbases and hitch lengths.

	Tractor	28-ft Trailer	33-ft Trailer	53-ft Trailer	Dolly
Wheelbase [ft]	20.83	22.95	28.03	43.91	6.04
Hitch Length [ft]	21.46	25.45	30.64	-	6.04

Figure 3-25 presents peak off-tracking for each trailer configuration through all four maneuvers. Unsurprisingly, the order from least to most off-tracking in all maneuvers was: 28-ft double, 33-ft double,

53-ft single. Although the A-double trailer configurations are longer than the single 53-ft trailer, the additional articulation points reduce off-tracking. The 33-ft double has more off-tracking than the 28-ft double because the 33-ft trailers have longer wheelbases while still having similar hitch offsets and are coupled by the same converter dolly as the 28-ft trailers. These results coincide with the findings from Section 3.1.6 and will be used for comparison to field test data discussed in Chapter 4. Figure 3-26 and Figure 3-27 present the rearmost axle paths of each trailer configuration for the four maneuvers. The line perpendicular to the steer path where peak off-tracking occurs for each trailer configuration is also shown. Because the single 53-ft trailer has tandem axles, the rearmost axle is taken as the geometric center of the tandem axles. Figure 3-28 and Figure 3-29 present the off-tracking through each maneuver. Interestingly, the 33-ft double off-tracking is quicker to rise at the beginning of the maneuver than the 53-ft single configuration. This coincides with data presented in Section 3.1.6, but is not as prominent because the two trailer configurations have very different steady-state (maximum) off-tracking. All three trailer configurations cross the steer axle path along the roundabout maneuver near the same locations. This also coincides with the findings in Section 3.1.6 with the compound J-turns, however significant off-tracking differences through the maneuver result more variation in crossing the roundabout near the exit than near the entrance of the maneuver. In summary, differences between the trailer configurations' axle paths and off-tracking through each of the four maneuvers of interest were clearly demonstrated. The peak off-tracking values will be considered in comparison to experimental tests detailed in Chapter 4.

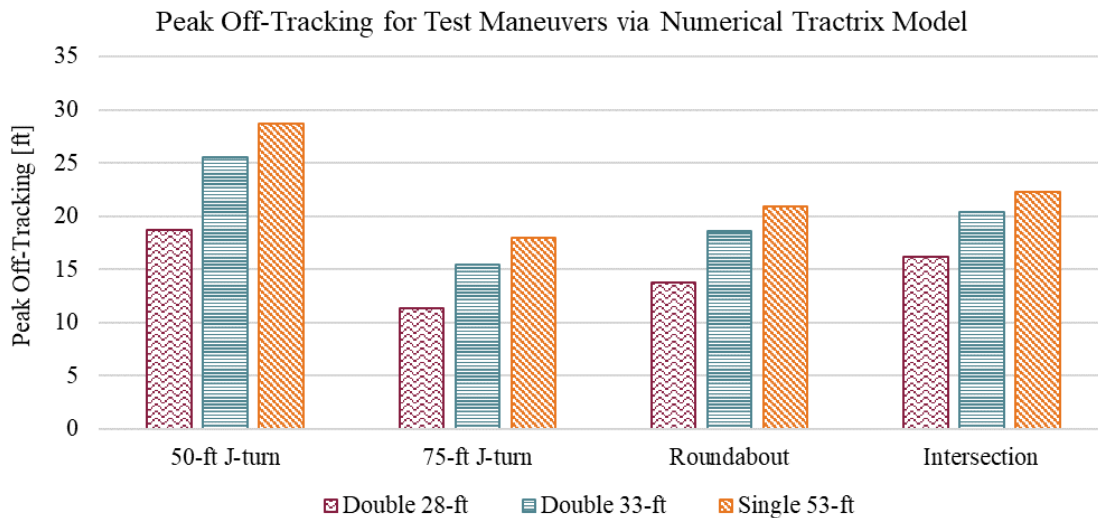
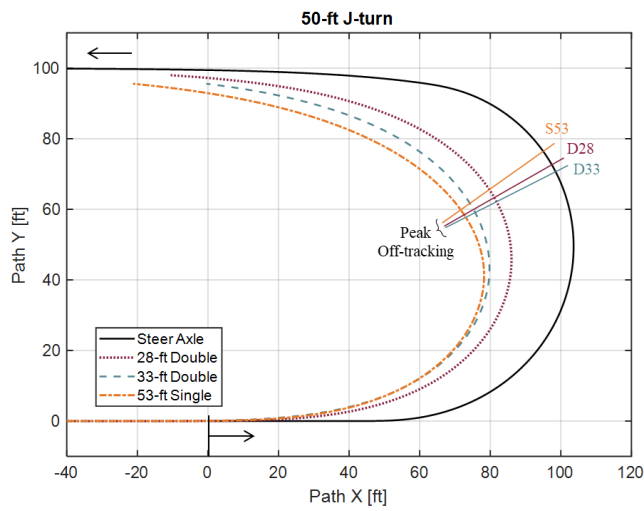
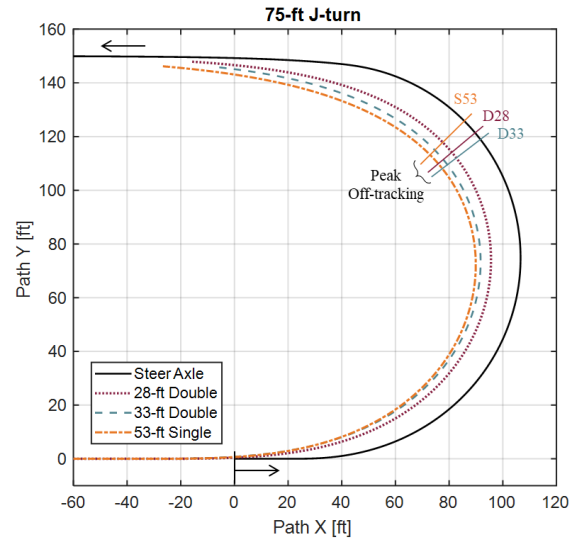


Figure 3-25. Peak off-tracking determined by the numerical tractrix model for each trailer configuration through the four test maneuvers.

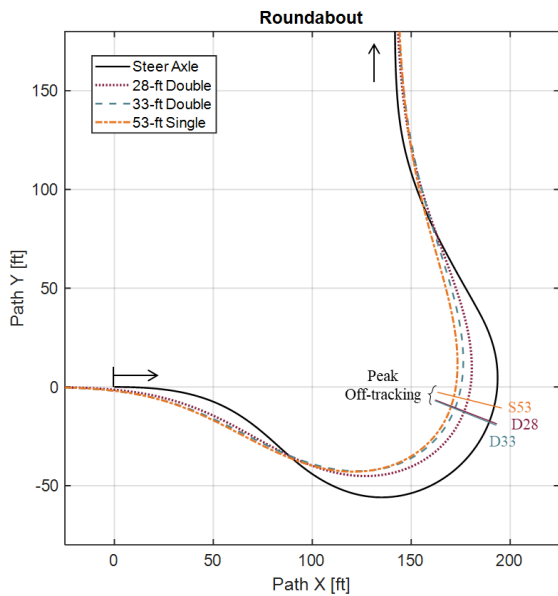


(a)

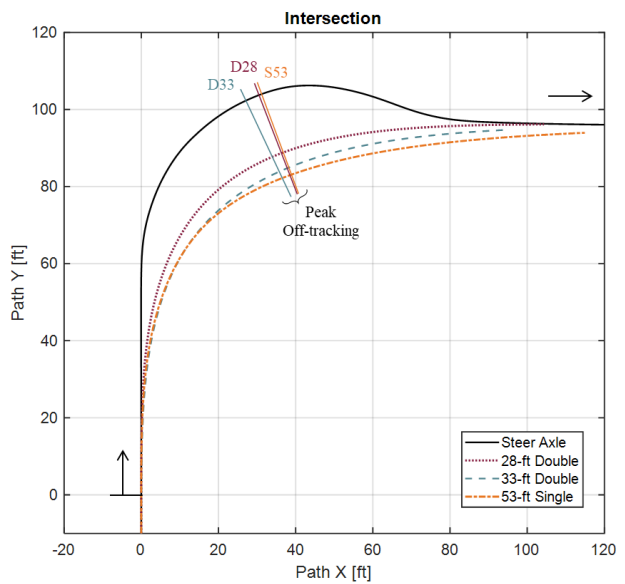


(b)

Figure 3-26. Steer axle and rearmost axle paths of compared trailer configurations through a (a) 50-ft J-turn and (b) 75-ft J-turn.

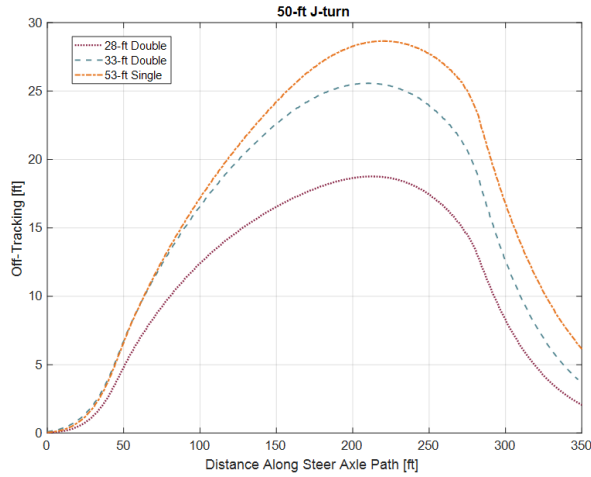


(a)

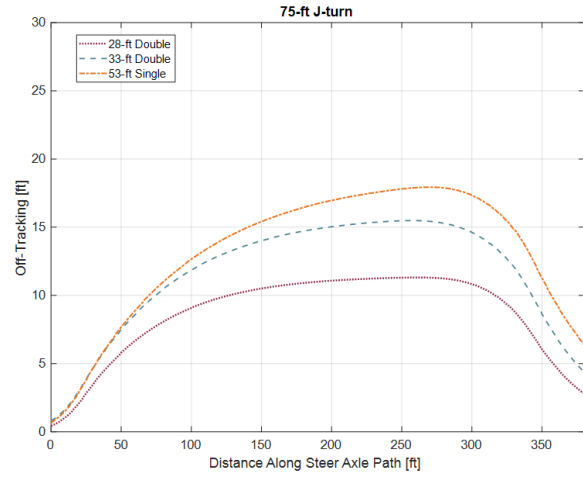


(b)

Figure 3-27. Steer axle and rearmost axle paths of compared trailer configurations through a (a) roundabout and (b) right-turn at an intersection.

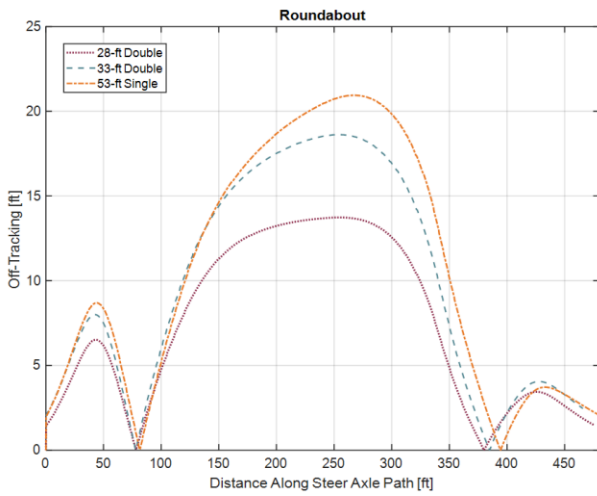


(a)

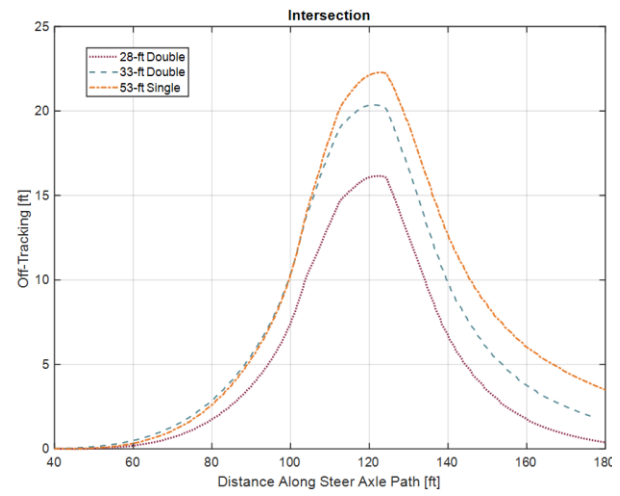


(b)

Figure 3-28. Rearmost axle off-tracking through the (a) 50-ft J-turn and (b) 75-ft J-turn.



(a)



(b)

Figure 3-29. Rearmost axle off-tracking through the (a) roundabout and (b) intersection maneuvers.

Chapter 4

Low-speed Off-tracking Field Test and Instrumentation Calibration

For any field test, understanding and acknowledging measurement uncertainty is paramount to validate test results. Past physical studies on LCV off-tracking have either been crudely setup or did not clearly document the test procedure and measurement uncertainty. Furthermore, physical tests comparing double 33-ft trailers to other trailer configurations in real-world maneuvers have not been conducted before. This chapter aims to provide these missing gaps in literature. Testing for and measuring off-tracking can be accomplished in a number of different methods including the use of GPS-based systems [27], local positioning, visual odometry [2], path traces [4], and so forth. Each method has its own pros and cons; however, the most precise and accurate methods typically require the largest capital investment in instrumentation. The method detailed here was chosen to prioritize the following criteria:

1. Employ sensors already owned by the lab facility;
2. If absolutely necessary, only purchase sensors that will directly benefit known future heavy vehicle tests to be conducted by the lab facility;
3. Easily integrate sensors into existing data acquisition unit cohesively; and
4. Design test procedure for speed and repeatability to accommodate limited access at desired test site.

The first considered method to measure vehicle motion through each test maneuver was by way of real-time kinematic (RTK) GPS systems like that shown in [27]. This method uses a base station staged at the test site and communicates with the GPS receiver to correct for GPS measurement errors. Although these systems claim very high accuracy and precision, their cost was beyond the project's budget, and they would create additional challenges to synchronize with the data acquisition unit used for other sensors. Local positioning systems (LPS) work similarly to GPS, but instead of the receiver communicating with satellites, LPS estimates a receiver's distance only from staged anchors. To triangulate the receiver's position, several anchors must be scattered about the test area. Local positioning systems typically provide high accuracy and precision, but the accuracy is influenced by the test site size and number of anchors. Unfortunately, affordable local positioning systems were only suitable for small area, indoor use. Larger scale local positioning systems exist but are designed for more permanent installations and are well beyond the project's budget. These are typically found in shipyards to track shipping containers or inside large manufacturing facilities to track material and goods.

Some example past studies on LCV motion and off-tracking used completely different methods to measure the tractor and trailer paths. Saxe and Cebon installed several stereo cameras on the test vehicle and designed computer vision algorithms to capture key elements in the environment that were then used to estimate the relative position of the vehicle to those elements [2]. This method did not provide consistently accurate results and modifying the computer vision algorithm is well beyond the scope of this study. Erket, Sessions, and Layton implemented a significantly simpler approach and conducted off-tracking tests on soft soil so that the vehicle's tires left definitive traces on the ground [4]. These traces were then measured after the vehicle passed through the maneuver. This method was acceptable for the tested equipment (logging trucks) and could easily be adapted for on-highway vehicles using chalk attached to an axle to mark the axle's trajectory. However, measuring each axle path of interest for all configurations and maneuvers (and ideally multiple trials per combination) would require a significant amount of time to conduct the study. Furthermore, tight measurement resolution along each path is required to capture peak and transient off-tracking through a maneuver. The total number of individual manual measurements increases with resolution, resulting in significant time increase to conduct the study.

The designed method to estimate vehicle motion and off-tracking is summarized in Figure 4-1. First, paths were laid out using reflective highway tape and white rope on the test cite asphalt. A downward-facing camera was positioned low on the tractor bumper and the camera's view was projected in real time to a custom dash-mounted monitor. The driver traveled through each maneuver while maintaining the laid path within the camera's field-of-view. GPS speed was integrated along the target steer axle path to synchronize all data spatially. Articulation angles between each vehicle unit were measured by multi-segment LiDAR sensors. The SGM-C algorithm estimated the tractor's orientation and dolly's contribution to trailer-trailer articulation angle measurement at each step along the steer axle path. The combination of the tractor steer axle path, estimated tractor orientation from a tractrix model, measured articulation angles, and known vehicle dimensions determined each axle's path through the maneuver. Off-tracking was then determined for each axle path.

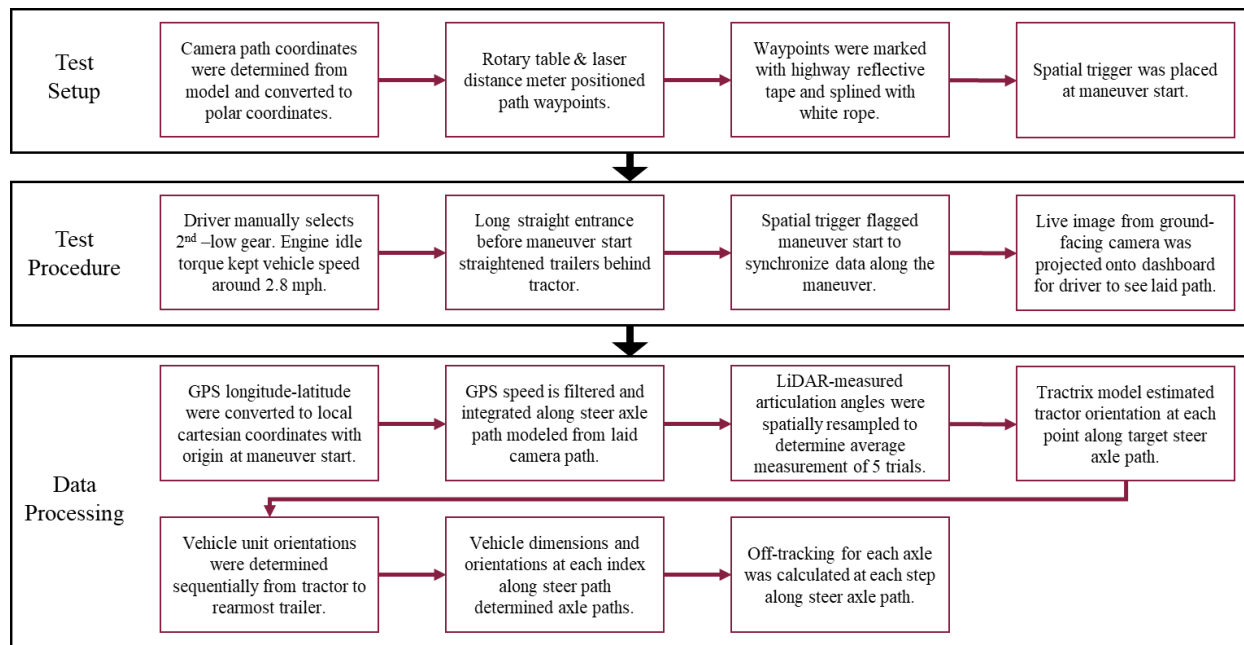


Figure 4-1. Low-speed off-tracking field test procedure. All key steps are shown from test setup to test procedure and data analysis.

This chapter details the setup, instrumentation, data processing, and results of the field tests on low-speed off-tracking of common LCVs in the U.S. – 28-ft double, 33-ft double, and 53-ft single – through four real-world maneuvers – 50-ft J-turn, 75-ft J-turn, left-turn in a roundabout, and right-turn in an intersection. The dimensions of each test vehicle configuration were listed previously in Table 3-3 and the maneuvers’ steer axle paths were presented previously in Figure 3-21. The following subsections are listed in order of the test flow. The first section details peak off-tracking sensitivity to measurement error within each step of the test process. Next, sensor calibrations and uncertainty are detailed for each measurement. The chapter concludes with test results and a discussion on measurement and result uncertainties for each maneuver and vehicle configuration.

Although transient motion of each trailing axle relative to the steer axle for the tested vehicle configurations is of interest, the most concise metric for comparing trailer configurations and maneuvers is peak off-tracking. Peak off-tracking reduces the overall vehicle configuration and maneuver characteristics to a single data point. By reducing the analysis to a single data point result, the complexity required to estimate uncertainty propagation from individual measurements and the test procedure is significantly reduced.

Uncertainty of measured peak off-tracking is determined by the root-sum-of-squares of each measurement's uncertainty multiplied by the result's (peak off-tracking) sensitivity to that measurement. As will be demonstrated in Section 4.1, peak off-tracking sensitivity to each measurement is dependent on the trailer configuration and maneuver. Section 4.1 provides the sensitivity indices λ and Sections 4.2 and 4.3 detail measurement uncertainties $u_{\bar{x}}$ for each parameter. The combined uncertainty in peak off-tracking from each measured value is found as Equation (4-1) below. Equation (4-1) may only be used if all measurement uncertainties $u_{\bar{x}}$ are determined at the same confidence level. Throughout this analysis, all uncertainties are calculated or assumed at the 95% level. This is common practice in engineering and is typically provided in sensor datasheets [17,26].

$$u_R = \left[\sum_{i=1}^N (\lambda_i u_{\bar{x}_i})^2 \right]^{\frac{1}{2}} \quad (4-1)$$

4.1 Peak Off-Tracking Sensitivity Analysis

Tractor-trailer motion is a nonholonomic system and therefore does not have a closed form solution to determine how each measured parameter – steer axle path, articulation angles, etc. – impacts the result, off-tracking. Thus, each trailer configuration must be simulated through all test maneuvers. This was accomplished using the SGM-C numerical tractrix model with 4 in (0.1 m) step size which, as shown in Chapter 3, provides an accurate representation of vehicle off-tracking. Once the baseline axle paths and peak off-tracking were established, each of the following parameters were altered independently to simulate measurement errors and the tractrix model was re-run, as illustrated by the block diagram in Figure 4-2 below.

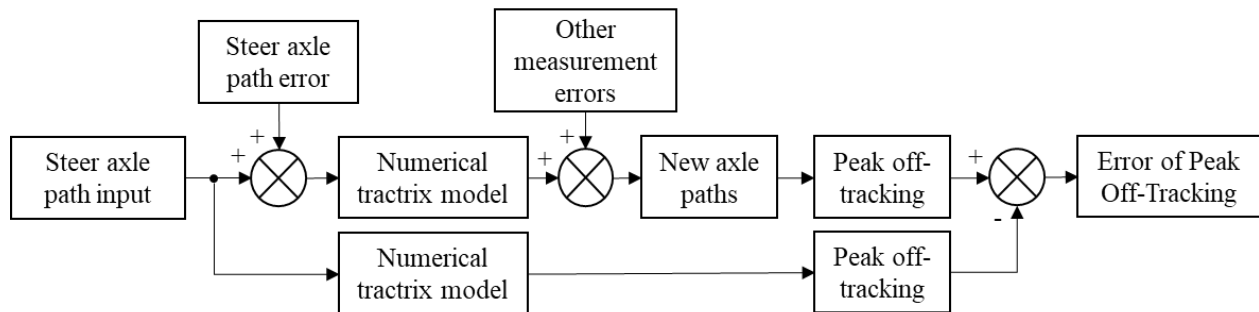


Figure 4-2. Block diagram illustrating process flow for simulating measurement errors.

From the scatter of peak off-tracking error with respect to the parameter with simulated error, a sensitivity index was determined according to Equation (4-2) where λ_i is the sensitivity index of the result R (peak off-tracking) to parameter x_i . The parameters' errors are grouped into two categories: spatial and temporal. A temporal error for a parameter may be equivalent to some spatial error of the same parameter, however, the relationship of peak off-tracking is unique to temporal and spatial errors and thus must be addressed separately.

$$\lambda_i = \frac{\partial R}{\partial x_{i_{x=\bar{x}}}} \quad (4-2)$$

4.1.1 Peak Off-Tracking Sensitivity to Spatial Errors

Spatial errors are errors in direct measurements such as articulation angles or steer axle path. Peak off-tracking sensitivity with respect to spatial errors has units of [ft/ft] or [ft/deg]. The spatial errors of interest are listed below.

1. **Steer axle position offset from the baseline axle path.** This is the position error of the estimated steer axle path relative to the actual steer axle path in the direction perpendicular to the target path. This may be the result of error from physically laying the path and/or error in the steer axle's position relative to the camera's position used to follow the laid path.
2. **Tractor orientation.** This is used to illustrate the impact of the model's estimated tractor orientation relative to what the actual orientation may be. Tractor orientation error is dominated by the driver's steering behavior while maintaining the laid path within the camera's field-of-view.
3. **Articulation angle between the tractor and front trailer.** This captures error in the articulation angle measurement between the tractor and front trailer and accounts for sensor performance, calibration, and installation.
4. **Articulation angle between the front and rear trailers.** This captures error in the articulation angle measurement between the front and rear trailers and accounts for sensor performance, calibration, and installation.

Although vehicle motion is nonlinear, the peak off-tracking sensitivity is approximately linear with respect to each of the four simulated spatial errors for the three tested trailer configurations and four maneuvers. As an example, Figure 4-3 presents the peak off-tracking error of the 50-ft J-turn maneuver with respect to articulation angle error between the tractor and front trailer for the three trailer configurations. Notice that as the articulation angle error increases, so does the peak off-tracking error.

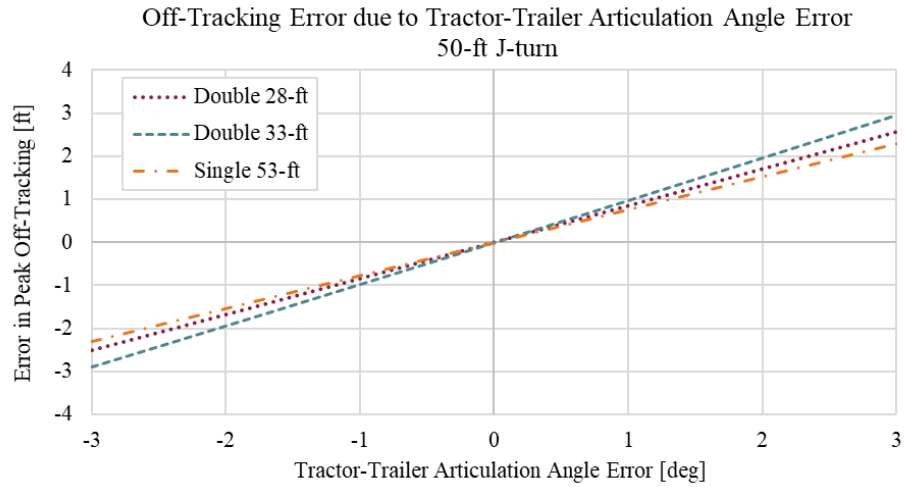


Figure 4-3. Error in peak off-tracking with respect to tractor-trailer articulation angle measurement error through the 50-ft J-turn.

Because the relationships are approximately linear and pass through the origin, the slope of each linear curve-fit is taken as the sensitivity index λ . The tractor orientation, tractor-trailer articulation, and trailer-trailer articulation sensitivity indices all have the same units of [ft/deg] and are compiled into Figure 4-4. As expected, the sensitivity increases with an increase in total vehicle length following the measurement. For example, peak off-tracking is more sensitive to tractor orientation than tractor-trailer articulation for all trailer configurations. For the A-doubles, peak off-tracking is more sensitive to tractor-trailer articulation than trailer-trailer orientation. Similarly, among the trailer configurations the order from least to most sensitive to tractor orientation is as follows: 53-ft single, 28-ft double, and 33-ft double. The summation of sensitivities increases with the total vehicle length and number of parameters, as demonstrated by the combined height of each peak off-tracking sensitivity bar in Figure 4-4 for each maneuver and trailer configuration. The summation of these sensitivities is approximately constant among the maneuvers for a given trailer configuration.

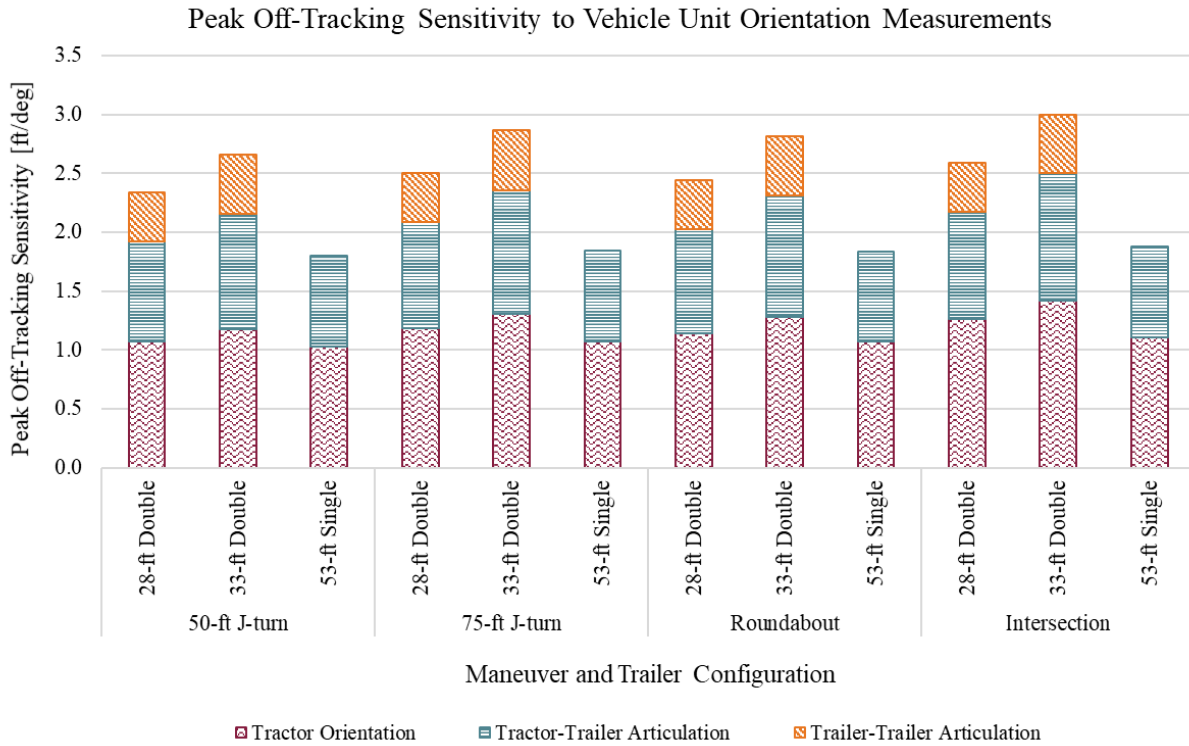


Figure 4-4. Peak off-tracking sensitivity to tractor orientation, tractor-trailer articulation, and trailer-trailer articulation measurements.

While peak off-tracking sensitivity to vehicle unit orientations typically increases with the total vehicle length, the percent peak off-tracking sensitivity decreases with an increase in baseline off-tracking, as demonstrated in Figure 4-5. For example, the 33-ft double's peak off-tracking is more sensitive to each vehicle unit's orientation than the 28-ft double; however, the percentage of peak off-tracking is less sensitive for the 33-ft double than the 28-ft double. This relationship among trailer configurations held for all four maneuvers and is likely to exist for most maneuvers where off-tracking is large. From this, it is suggested that, in general, the percent error in peak off-tracking is more sensitive to the vehicle orientation and articulation measurements from a vehicle configuration with less off-tracking than a vehicle with more off-tracking. Similar results are found when comparing maneuvers for a given trailer configuration. For example, the order of maneuvers from least to most peak off-tracking of the baseline model is roundabout, 75-ft J-turn, intersection, and 50-ft J-turn. This order is the same but from most to least percent peak off-tracking. These trends show that, in general for common U.S. road-going LCVs, the percent peak off-tracking is less sensitive to vehicle unit orientations for trailer configurations and maneuvers that have more off-tracking.

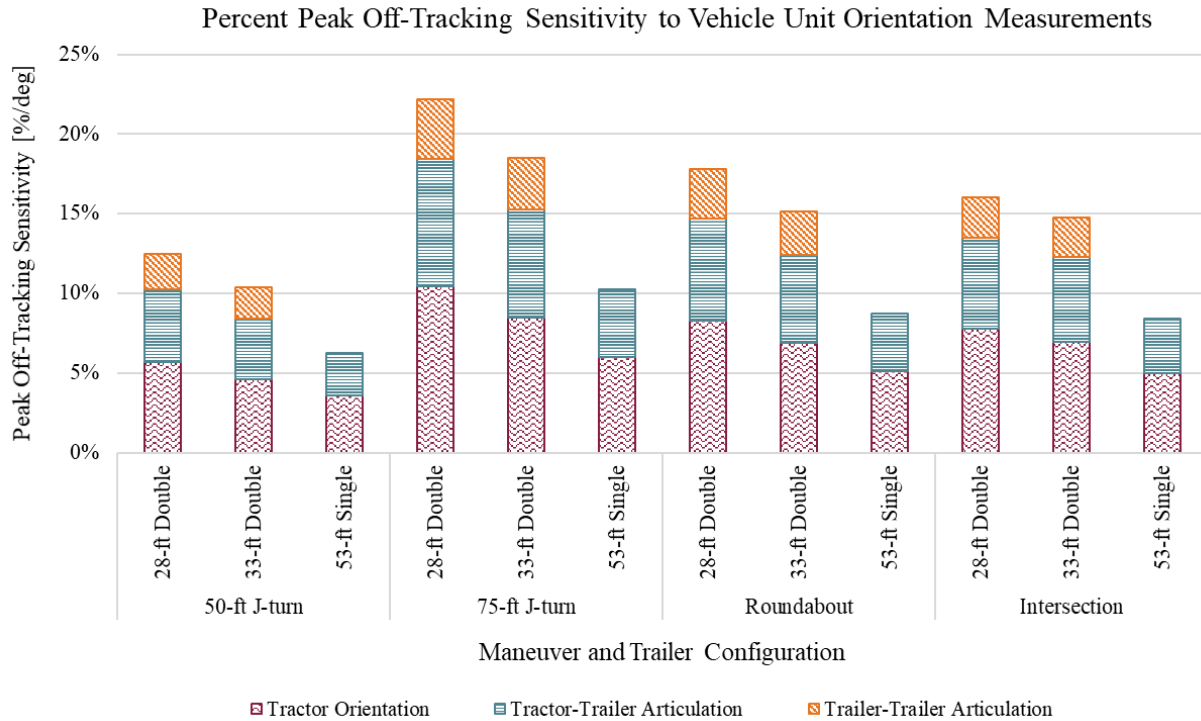


Figure 4-5. Percent peak off-tracking sensitivity to tractor orientation, tractor-trailer articulation, and trailer-trailer articulation measurements.

Peak off-tracking sensitivity to the steer axle path lateral position error has dimensions of [ft/ft] and thus are shown in Figure 4-6 and not included as part of Figure 4-4 and Figure 4-5. It is unsurprising that, generally, the sensitivity increases in order of increasing off-tracking of the three vehicle units. This occurs for the J-turn maneuvers and roundabout, but not the intersection. Although the intersection maneuver has the second highest peak off-tracking of the four maneuvers, it has the second lowest sensitivity to steer axle lateral position error for the double trailer configurations, and the lowest for the 53-ft single trailer configuration. The different results among the maneuvers exemplify how the sensitivity analysis must be conducted iteratively for each maneuver and trailer configuration of interest rather than determined by a closed-form, general solution.

Just as with sensitivity to vehicle unit orientations, the percent peak off-tracking sensitivity to the steer axle path lateral position error generally decreases with an increase in off-tracking. This trend was observed across all trailer configurations except for the 75-ft J-turn where the 33-ft double's percent peak off-tracking was more sensitive to the steer path lateral error than the 28-ft double. This data is presented as Figure 4-7.

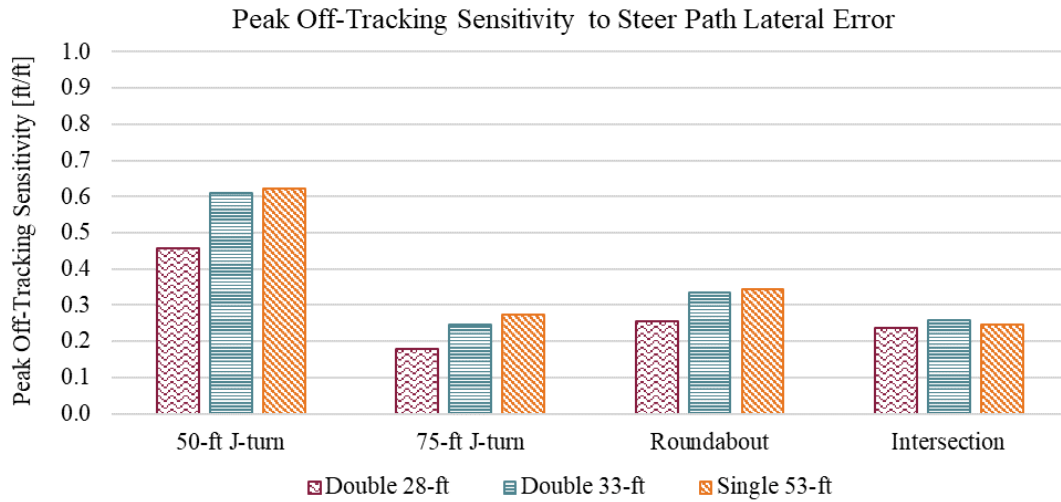


Figure 4-6. Peak off-tracking sensitivity to steer axle path lateral position error.

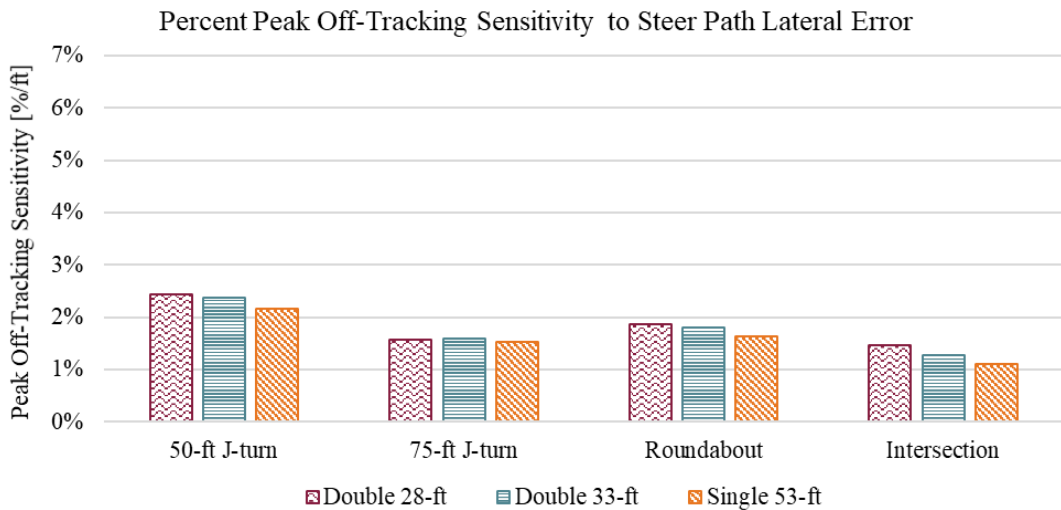


Figure 4-7. Peak off-tracking sensitivity to steer axle path lateral position error.

In addition to the four spatial parameters discussed, there is one other parameter incorporated as part of the low-speed off-tracking field tests: the dolly's contribution to the trailer-trailer articulation angle measurement. The dolly's contribution is represented as a percentage of the total articulation between the front and rear trailers and is determined by the tractrix model. This estimated percentage is directly applied to the measured data, even if the measured and modeled data do not match exactly. A sensitivity analysis on this percentage showed that the percent peak off-tracking sensitivity was less than 0.25%/ft for all trailer configurations and maneuvers. Because this value is significantly less than the other factors, it is assumed negligible and omitted from the sensitivity and uncertainty analyses.

4.1.2 Peak Off-Tracking Sensitivity to Temporal Errors

Temporal errors are the result of unsynchronized data. Because the tractrix model is independent of speed, it is easiest to simulate temporal errors by shifting data along the steer axle path. If speed is of interest, then this value may be used after converting speed to distance for a given time error. For example, a 1 second synchronization error while traveling at 3 mph effectively shifts the data by 4.4 ft along the steer axle path – if the tractor-trailer articulation angle has a 1 sec temporal error at this speed, then the articulation angle measured from 4.4 ft ahead of the current position is taken as the current measured articulation angle. By this definition, peak off-tracking sensitivity with respect to temporal errors has the units [ft/ft]. Temporal errors of interest are listed below.

1. **Steer axle position error along the target axle path.** This value replicates temporal error in synchronizing all data (tractor orientation estimate, articulation angles, etc.) with where the tractor is believed to be along the target path. This is equivalent to spatial error in the steer axle's path along the predefined target path. This type of error can occur from a misplaced target path and/or accumulated error from integrating GPS speed along the assumed path.
2. **Tractor-trailer articulation angle synchronization.** This replicates temporal error in synchronizing LiDAR data of the tractor-trailer articulation angle with respect to all other data.
3. **Trailer-trailer articulation angle synchronization.** This replicates temporal error in synchronizing LiDAR data of the trailer-trailer articulation angle with respect to all other data.

The tractor's orientation is not included here because it is estimated by the tractrix model for a given steer axle path input. Simulating only tractor orientation synchronization error effectively assumes that the steer axle position error along the axle path is equivalent to the distance for a given vehicle speed and temporal error in both tractor-trailer and trailer-trailer articulation angle measurements. This does not capture the true nature of the low-speed study procedure and measurements and is thus omitted.

Unlike the sensitivity to spatial errors, the relationship of peak off-tracking sensitivity to temporal errors is a single, approximately linear curve. The most extreme example is plotted in Figure 4-8 – the error in peak off-tracking with respect to tractor-trailer articulation angle temporal error for the 75-ft J-turn. For this maneuver, the relationship between the result and input temporal error may be best represented as a piecewise linear function. For easiest implementation and most conservative approach, the slope of the portion with greatest slope magnitude was taken as the sensitivity index λ . This is an acceptable approach because (1) the relationship is monotonic, and (2) the relationship is approximately linear within the range

where the slope is taken. This piecewise-like behavior of peak off-tracking error sensitivity was only observed for the following parameters, maneuvers, and trailer configurations:

- Tractor-trailer articulation temporal error
 - 75-ft J-turn: 28-ft double, 33-ft double, 53-ft double
 - Roundabout: 28-ft double
- Trailer-trailer articulation temporal error
 - 75-ft J-turn: 28-ft double, 33-ft double, 53-ft double
 - Roundabout: 28-ft double

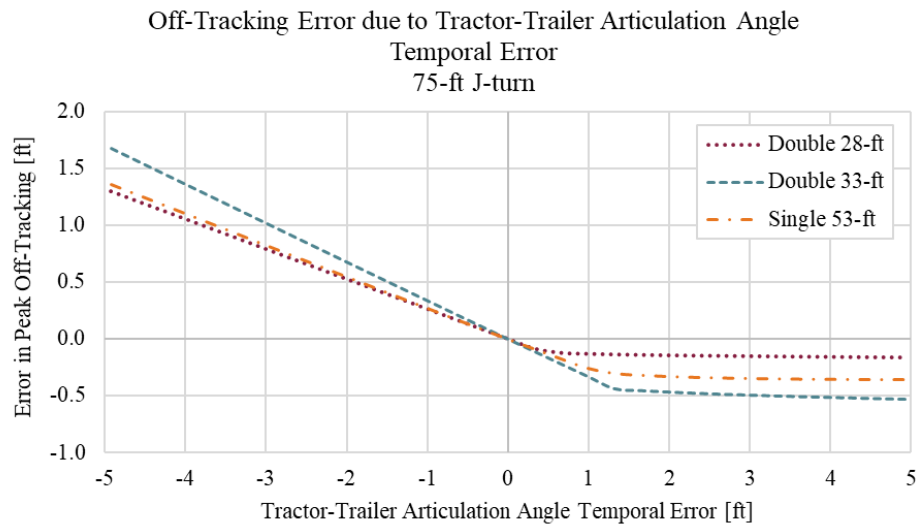


Figure 4-8. Error in peak off-tracking with respect to tractor-trailer articulation angle temporal error through the 75-ft J-turn.

Error in peak off-tracking with respect to temporal errors for all other maneuver, trailer configuration, and parameter combinations were approximately linear. Just as with the spatial errors, the sensitivity index for each temporal error is taken as the slope of the linear curve fit. For piecewise relationships like that shown in Figure 4-8, the sensitivity index was taken as the slope with greatest magnitude.

Unlike the sensitivity to spatial errors, peak off-tracking sensitivity to temporal errors was not consistent among the different trailer configurations and maneuvers. Figure 4-9 plots peak off-tracking sensitivity with respect to articulation angle temporal errors. This data clearly demonstrates the irregular relationships of peak off-tracking sensitivity with respect to temporal errors for a variety of trailer configurations and maneuvers. Figure 4-10 presents the percent peak off-tracking error with respect to articulation angle

temporal errors. Just as with the spatial parameters, the percent peak off-tracking sensitivity generally decreases as the off-tracking increases.

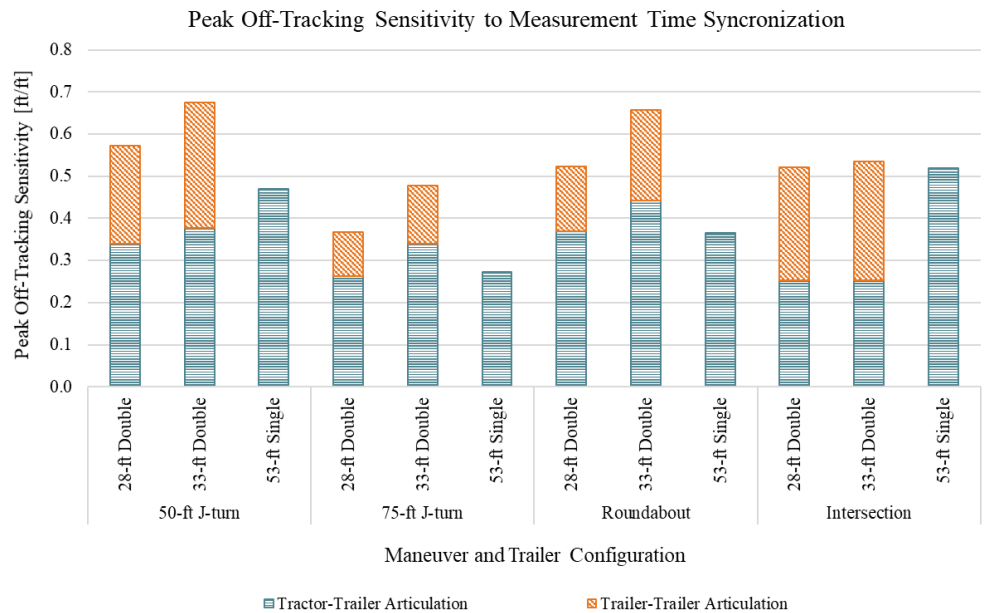


Figure 4-9. Peak off-tracking sensitivity to tractor-trailer articulation and trailer-trailer articulation temporal errors.

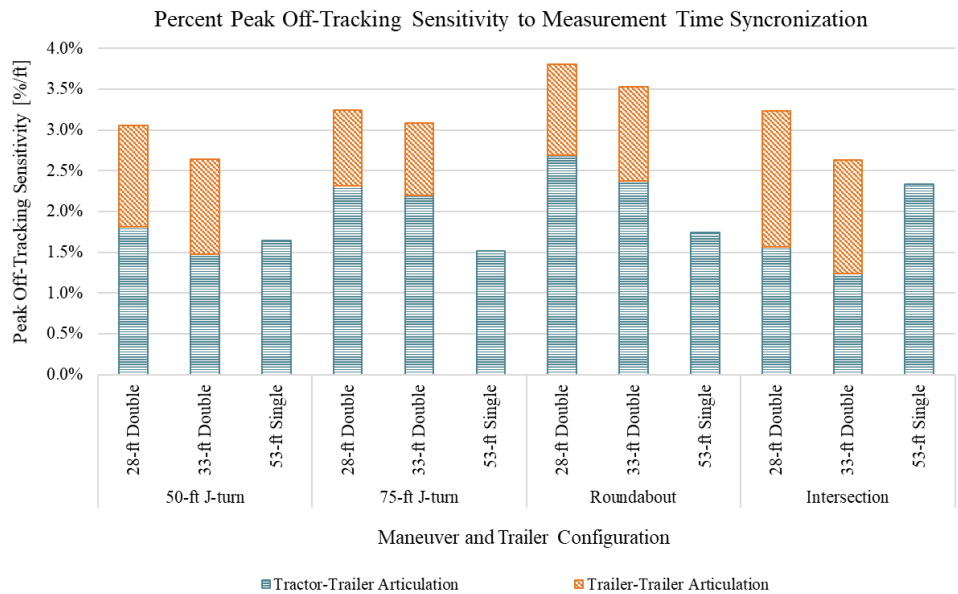


Figure 4-10. Percent peak off-tracking sensitivity to tractor-trailer articulation and trailer-trailer articulation temporal errors.

Figure 4-11 presents the peak off-tracking sensitivity to steer axle path temporal error. This sensitivity is equivalent to steer axle position error along the target axle path. For three of the four maneuvers, the order from least to most sensitive is 53-ft single, 28-ft double, and 33-ft double. Unlike the significant variation of sensitivity to articulation angle temporal error among trailer configurations, the sensitivity to the steer axle temporal error is approximately the same for the three trailer configurations for a given maneuver. Figure 4-12 plots the percent peak off-tracking sensitivity to steer axle path temporal error. This sensitivity exhibited clear relationships among the different trailer configurations. For all four maneuvers, the 53-ft single, 33-ft double, and 28-ft double had the smallest to largest percent peak off-tracking sensitivity, respectively, relative to the steer axle path temporal error.

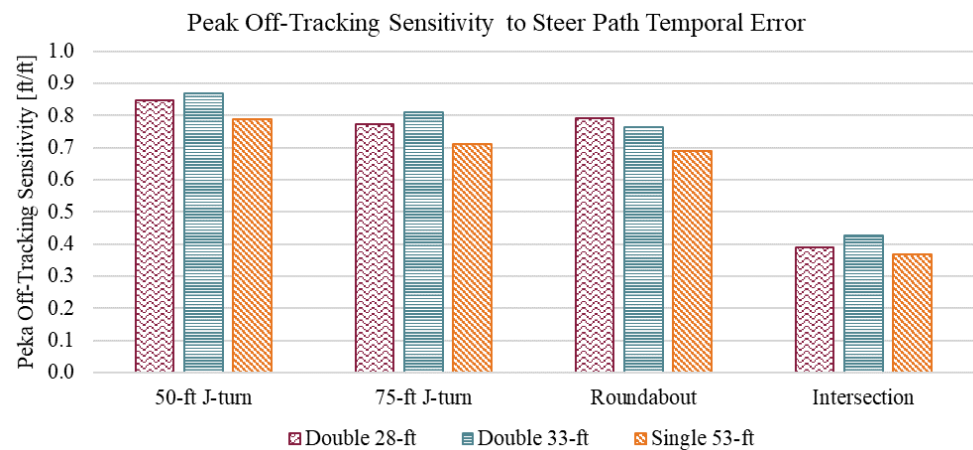


Figure 4-11. Peak off-tracking sensitivity to steer axle path temporal error.

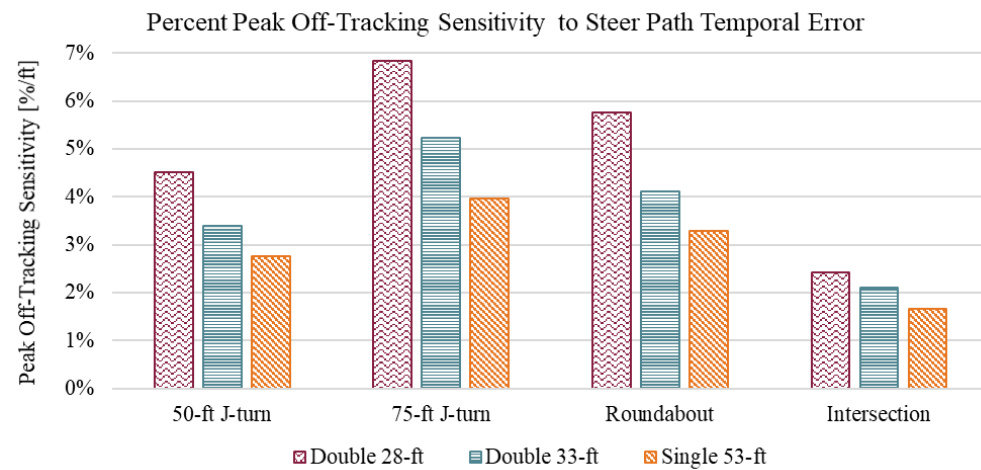


Figure 4-12. Percent peak off-tracking sensitivity to steer axle path temporal error.

4.1.3 Sensitivity Analysis Summary

In summary, peak off-tracking sensitivity to measurements is dependent on both the vehicle configuration and maneuver. Sensitivity to some parameters, such as tractor orientation, exhibited an approximately linear relationship as well as clear trends among vehicle configurations or maneuvers. However, sensitivity to some other parameters, such as tractor-trailer articulation angle time synchronization, did not demonstrate clearly defined trends among trailer configurations and maneuvers. Table 4-1 lists the peak off-tracking sensitivity indices for each parameter of interest for the three trailer configurations and four maneuvers.

Table 4-1. List of peak off-tracking sensitivity indices for each parameter of interest for the test vehicle configurations and maneuvers.

Maneuver	Trailer Configuration	Spatial Parameters				Temporal Parameters		
		Steer Axle Position Perpendicular to Path	Tractor Orientation	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle	Steer Axle Position Along Path	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle
		[ft/ft]	[ft/deg]	[ft/deg]	[ft/deg]	[ft/ft]	[ft/ft]	[ft/ft]
50-ft J-turn	28-ft Double	0.46	1.07	0.85	0.42	0.85	0.34	0.23
	33-ft Double	0.61	1.18	0.98	0.50	0.87	0.38	0.30
	53-ft Single	0.62	1.03	0.77	-	0.79	0.47	-
75-ft J-turn	28-ft Double	0.18	1.18	0.90	0.42	0.77*	0.26*	0.10
	33-ft Double	0.25	1.31	1.05	0.51	0.81*	0.34*	0.14
	53-ft Single	0.28	1.08	0.77	-	0.71*	0.27*	-
Roundabout	28-ft Double	0.26	1.14	0.88	0.42	0.79*	0.37*	0.15
	33-ft Double	0.34	1.28	1.03	0.50	0.79	0.44	0.21
	53-ft Single	0.34	1.07	0.77	-	0.69	0.37	-
Intersection	28-ft Double	0.24	1.26	0.92	0.41	0.39	0.25	0.27
	33-ft Double	0.26	1.42	1.08	0.50	0.43	0.25	0.28
	53-ft Single	0.25	1.11	0.77	-	0.37	0.52	-
* More conservative value taken from piecewise monotonic relationship.								

4.2 Test Setup and Procedure

All tests were conducted in a large, open parking lot at Virginia Tech in March 2020. For a given maneuver, fair comparison between trailer configurations is only achieved when the steered path is nearly identical for all tests. To accommodate this, each maneuver path was securely marked on the asphalt with highway reflective tape and white rope. The path was followed using a downward facing camera mounted low on the tractor's front bumper projecting a live video feed to a monitor in the driver's direct line of sight on top of the dashboard. The video camera used was a GoPro Hero 4 and was configured to its narrowest field-of-view (FOV) setting. The narrow field of view ensured that the driver stayed along the desired path if the laid path was within the camera's sight. Design criteria for the monitor were (1) large size to make it easy for the driver to see and (2) bright enough to not be washed out by direct sunlight. A self-contained 13" laptop display was chosen. A frame was built to house the display's controller, dedicated battery supply, voltage regulator, and encase the display itself. The monitor frame fit snug along the dashboard's curved surface to not move under vibration or driving, and the display was supported by a locking hinge so that it could be easily stowed away while not in use. The monitor and GoPro are pictured in Figure 4-13 below.

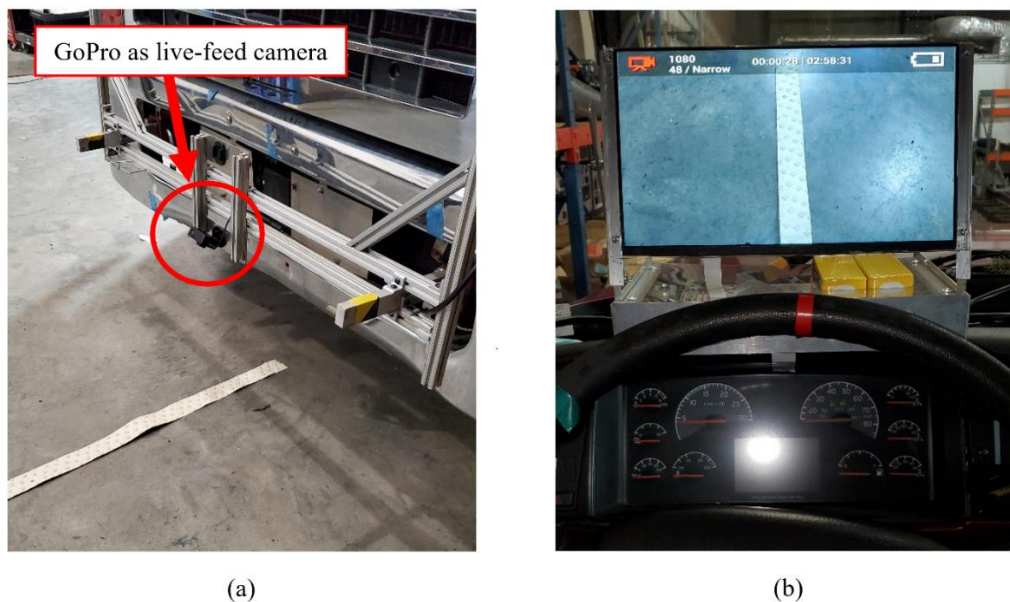


Figure 4-13. (a) GoPro installed on the front bumper as live-feed camera. (b) dashboard monitor for live-feed camera.

Because the GoPro was mounted forward of the steer axle, the coordinates for the laid paths of the roundabout and intersection maneuvers were preemptively adjusted to accommodate the difference between the driver's perspective and the desired steer axle path. The 50-ft J-turn and 75-ft J-turn

maneuvers' laid paths did not accommodate the camera's forward position from the steer axle because their simple geometry (half-circles of constant radius) eased the path installation process. Small adjustments like those made for the roundabout and urban intersection would add unnecessary complexity to the J-turn maneuvers without providing significant benefit to analyze the different trailer configurations' off-tracking. For all maneuvers, the difference in the steer axle path and camera path was accounted for during data processing and analysis. The difference between the camera path and the steer axle path is illustrated in Figure 4-14. The camera path was taken from the camera's field-of-view (FOV) center at the ground surface and accounts for the camera's slight forward-looking tilt.

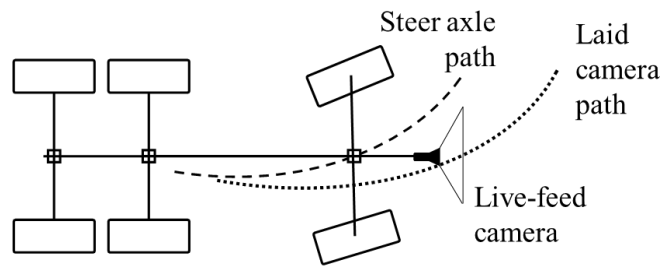


Figure 4-14. Illustration of the difference in the laid camera path and the target steer axle path through a turn.

After accounting for the camera's horizontal FOV, the camera's height from the ground while mounted on the tractor bumper, and the forward-looking pitch angle of the camera, the total width of the camera's FOV center at the ground surface was approximately 1.8 ft. The actual location of the laid path within the camera's FOV is assumed to fit a normal distribution where the FOV edges are at 5 standard deviations from the mean (middle). This distribution replicates variations as the driver navigates a maneuver and tries to maintain the laid path along the camera's centerline. Five standard deviations is appropriate here because the driver always maintained the laid path within the camera's FOV, and from visual inspection, the distribution from 4 standard deviations appeared too wide and 6 standard deviations too narrow. Because the system-wide uncertainty analysis is conducted at the 95% confidence level, the uncertainty is twice the standard deviation. Thus, the uncertainty in the camera's centerline existing along the laid path is estimated as 0.36 ft at the 95% confidence level, as shown by Equation (4-3) below.

$$u_{cam,FOV} = 2 \frac{1}{5} \left(\frac{1}{2} w \right) = \frac{1}{5} (1.8 \text{ ft}) = 0.36 \text{ ft} \quad (4-3)$$

This uncertainty must be translated to uncertainty of the steer axle's position relative to its target path. Figure 4-15 illustrates how an offset in the line's position within the camera's FOV influences the steer axle's position relative to its target path. From the diagram, it is easy to arrive to the sensitivity indices of

the steer axle's path with respect to driver behavior as they attempt to keep the laid path along the camera's centerline. Equations (4-4) and (4-5) provide the sensitivity indices of the steer axle's position perpendicular to and along the target steer axle path, respectively, caused by driver error.

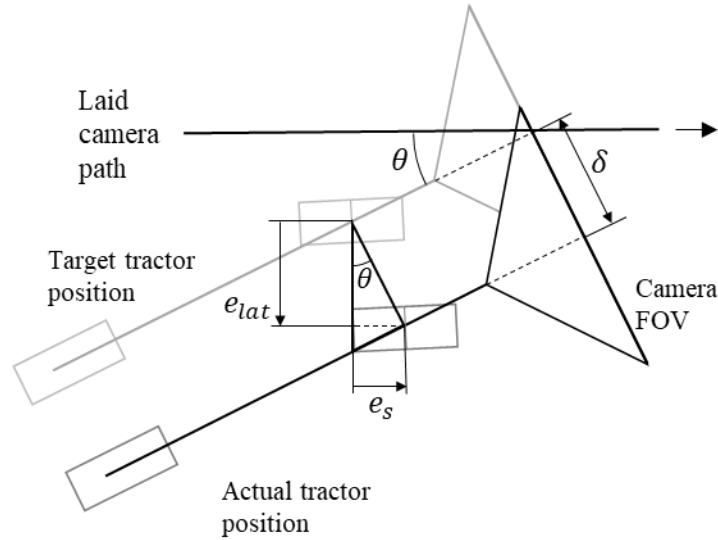


Figure 4-15. Schematic of steer axle position error due to offset of the camera path from the camera's FOV centerline. Tractor is represented as single-track bicycle model and dimensions are exaggerated for clarity.

The sensitivities are functions of tractor body's orientation relative to the steer axle path tangent line. The lateral position error is highest when this angle, θ , is zero, and the forward position error is highest when θ is 90° . At some point during some maneuvers θ is zero resulting the lateral position sensitivity as assumed constant and at its maximum value. This value was provided by Equation (4-5). With regard to the forward position sensitivity, θ never reached 90° in the field tests. The maximum value of θ was instead estimated by the numerical tractrix model through all four tested maneuvers. The maximum value of θ was 26.5° and occurred during the 50-ft J-turn. Conservatively, this value was assumed constant, and the sensitivity is provided by Equation (4-4).

$$\lambda_{s,driver} = \sin(\theta) = \sin(26.5^\circ) = 0.45 \quad (4-4)$$

$$\lambda_{perp,driver} = \cos(\theta) = \cos(0^\circ) = 1 \quad (4-5)$$

From these sensitivity index values and the uncertainty of the line's position within the camera's FOV, the uncertainty of the steer axle path with respect to the drivers' steering behavior is found as Equations (4-6) and (4-7) for the position along and perpendicular to the target path, respectively.

$$u_{s,driver} = \lambda_{s,driver} u_{camFOV} = 0.45(0.36 \text{ ft}) = 0.16 \text{ ft} \quad (4-6)$$

$$u_{perp,driver} = \lambda_{perp,camFOV} u_{camFOV} = 1(0.36 \text{ ft}) = 0.36 \text{ ft} \quad (4-7)$$

Additionally, the tractor's orientation is impacted by the driver's steering behavior while keeping the laid path within the camera's FOV. The tractor orientation sensitivity to driver steering behavior while following the laid path is approximately constant and independent of the angle θ between the tractor and path for angles within the expected range ($|\theta| \leq 26.5^\circ$). This value is specific to the forward position of the camera relative to the test tractor steer axle. The tractor orientation uncertainty due to driver steering behavior while keeping the laid path within the camera's FOV is described by Equation (4-8) and illustrated in Figure 4-16.

$$u_{to,driver} = \lambda_{to,camFOV} u_{camFOV} = (10.2^\circ/\text{ft})(0.36 \text{ ft}) = 3.7^\circ \quad (4-8)$$

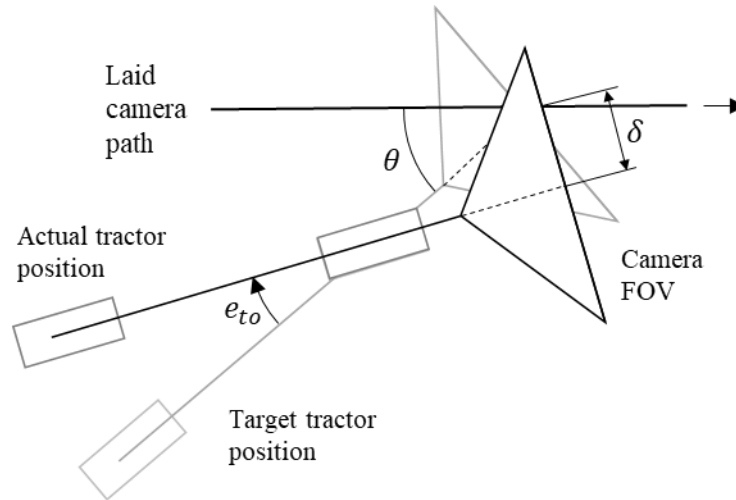


Figure 4-16. Schematic of tractor orientation error due to offset of the camera path from the camera's FOV centerline. Tractor is represented as single-track bicycle model and dimensions are exaggerated for clarity.

For each maneuver, the camera's path cartesian coordinates were converted to polar coordinates with the origin located near or at the centroid of the maneuver's largest turn. These points were realized using a rotary table at the polar coordinate origin to measure the angle coordinates and a Bosch GLM400C laser-distance meter (LDM) to measure the radius from the origin. The LDM was mounted to the rotary table such that the LDM's zero-distance reference was directly above the rotary table's axis of rotation as pictured in Figure 4-17 (a). Each waypoint was positioned and marked by two people: one person (A)

adjusting the rotary table to the desired angle and directing the second person (B) to move forward, backward, left, or right to position the waypoint target at the specified distance from the rotary table as measured by the LDM. This method is illustrated in Figure 4-17 (b). After marking all waypoints for a given maneuver with industry-specific highway reflective tape, the waypoints were splined with a thin white rope. The highway reflective tape and white rope remained in place throughout all planned tests.

Uncertainty that the laid camera path xy-coordinates matched the targeted camera path is a function of the rotary table's accuracy, the LDM's accuracy, the distance from the rotary table to the marked position, and the manual process of securely placing the tape on the asphalt at the intended position. Uncertainty along and perpendicular to the camera path is dominated by the path radius, so the 75-ft J-turn (which has the largest tested radius) is used to provide a conservative uncertainty value for all maneuvers. Although there does exist a small difference in the camera's path with respect to the target steer axle path, this difference is small enough that we may assume that uncertainty in the laid camera path is directly impacting the target steer axle path uncertainty and is thus taken as a component of the steer axle path's uncertainty. Note that the total uncertainty in the steer axle path is a combination of uncertainty due to driver behavior, the process of laying the camera path, the tractrix model, and data synchronization. The latter components are detailed in later sections. At this point, uncertainty of the camera's laid path, and consequently, the target steer axle path is the focus. The laid camera path uncertainty perpendicular to the target camera path is defined as Equation (4-10) where u_{LDM} is uncertainty of the radial measurement determined by the LDM found in Equation (4-9) and u_{man} is the estimated uncertainty for manually positioning the tape marker at the intended location. From the specification sheet, the LDM has a resolution of 0.031 in and typical accuracy of 0.1 in for unfavorable conditions. These values are included in Equation (4-9) to provide the LDM measurement uncertainty and are assumed at the 95% confidence level.

$$u_{LDM} = \sqrt{\left(\frac{1}{2}(0.031 \text{ in})\right)^2 + (0.1 \text{ in})^2} = 0.1012 \text{ in} = 0.0084 \text{ ft} \quad (4-9)$$

$$u_{cam,perp} = \sqrt{u_{LDM}^2 + u_{man}^2} = \sqrt{(0.0084 \text{ ft})^2 + (0.1 \text{ ft})^2} \approx 0.1 \text{ ft} \quad (4-10)$$

Equation (4-11) presents the uncertainty for each marked waypoint's position along the target camera path as a function of the rotary table's angle uncertainty u_{rot} , the radius at the corresponding position from the rotary table r , and the uncertainty of the manual process of positioning the tape marker at the intended location u_{man} . The rotary table's accuracy is 2.91×10^{-4} rad (0.017°) and, as discussed earlier, the radius

is taken from the 75-ft J-turn for a conservative uncertainty estimate. Inspection of Equation (4-10) and Equation (4-11) clearly indicates that the manual step in marking each waypoint is the dominant factor in the laid camera path and target steer axle path uncertainties. Uncertainty along the camera's path is inconsequential to the test procedure and is not included as part of the complete uncertainty analysis. It is provided here as an example and for completion.

$$u_{cam,along} = \sqrt{(ru_{rot})^2 + u_{man}^2} = \sqrt{\left((75 \text{ ft}) \left(\frac{1}{2}(2.91 \times 10^{-4})\right)\right)^2 + (0.1 \text{ ft})^2} \approx 0.1 \text{ ft} \quad (4-11)$$

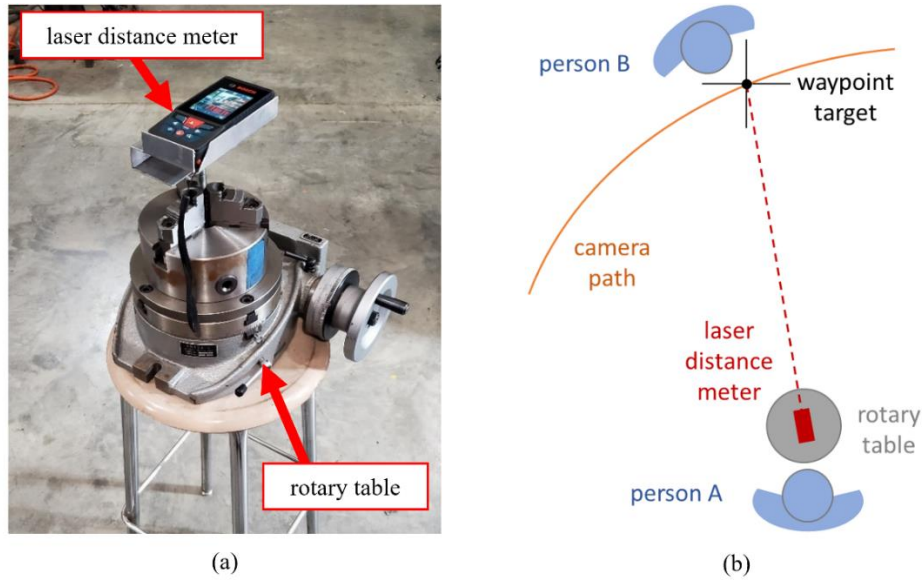


Figure 4-17. (a) Bosch GLM400C laser distance meter mounted to the rotary table used for positioning maneuver waypoints. (b) Schematic of locating maneuver waypoints.

All three trailer configurations were tested five times per maneuver, totaling 15 trials per maneuver and 60 trials in the whole study. To maintain a consistent speed across trials, the tractor's manual transmission was held in 2nd-low gear. In this gear, the engine's idle torque kept the tractor's steer axle moving steady about 2.5-3 mph. This freed the driver from moderating the engine throttle to focus solely on steering the tractor to ensure the laid path stayed within the camera's field-of-view. The beginning of each maneuver was marked in the data acquisition system by a spatial trigger that included a photoelectric sensor on the tractor front bumper and a reflective target at the maneuver starting gate. Details on the spatial trigger are provided later in Chapter 5.

4.3 Non-Contact Method to Measure Articulation Between Vehicle Units

4.3.1 Converting LiDAR Data to Articulation Angle

LeddarTech™ IS16 multi-segment LiDAR sensors measured articulation angles between vehicle units as illustrated in Figure 4-18. One sensor was mounted on the rear of the tractor to measure the articulation angle between the tractor and front trailer, and another sensor was mounted on the rear of the front trailer to measure the articulation between the front and rear trailers for the 28-ft double and 33-ft double configurations.

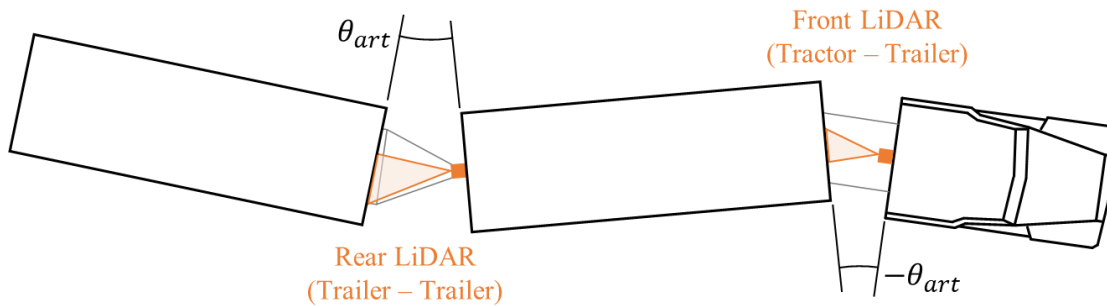


Figure 4-18. Illustration of articulation angles measured by multi-segment LiDAR.

Each LiDAR sensor has 16 individual segments spanning a total horizontal field-of-view (FOV) of 48 degrees as shown in Figure 4-19. Each segment provides its own distance measurement that is assumed to be found along the center of the segment's horizontal FOV. This assumption was provided in the specification sheet and confirmed during preliminary testing. For the LiDAR's local coordinate system where the +y axis is along the sensor's horizontal FOV centerline, each segment centerline and outer edge angles from the +y axis are listed in Table 4-2.

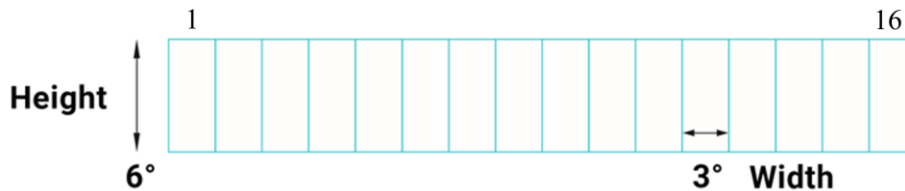


Figure 4-19. LeddarTech™ IS16 segment field-of-views.

Table 4-2. LeddarTech IS16 segment centerline and outer edge angles from the sensor's field of view center.

segment	1	2	3	4	5	6	7	8
center [deg]	-22.5	-19.5	-16.5	-13.5	-10.5	-7.5	-4.5	-1.5
outer edge [deg]	-24	-21	-18	-15	-12	-9	-6	-3
segment	9	10	11	12	13	14	15	16
center [deg]	1.5	4.5	7.5	10.5	13.5	16.5	19.5	22.5
outer edge [deg]	3	6	9	12	15	18	21	24

Each segment's measured distance is equivalent to a radius from the sensor at the segment's angle from the sensor's FOV centerline. These values are converted to a local cartesian coordinate system with the same origin at the sensor's base. Because the LiDAR sensor's +y axis is facing rearward along the lateral centerline of the vehicle unit it is attached, and because each trailer may be assumed to be a near-perfect rectangle, the relative articulation to the following vehicle unit is determined by the slope of the trailer front wall within the LiDAR's cartesian reference frame. This slope was calculated by a single-order linear curve fit of the cartesian coordinates of appropriate segments' measurements.

A 95% confidence interval for the articulation angle from the curve-fit is determined by Equations (4-12) through Equation (4-15). In Equation (4-12), u_{slope} is uncertainty in the linear curve-fit's slope, $t_{v,p}$ is the t-distribution variable, and SE is the standard error of the slope estimate. The number of degrees of freedom v – dependent on N , the number of LiDAR segments used – and corresponding confidence interval determine the value of $t_{v,p}$. This value changes through with articulation angle as more or less segments observe the same trailer wall and are used to estimate the slope.

$$u_{slope} = t_{v,p}SE \quad (4-12)$$

The standard error was calculated from the measured data points y_i and the estimated values y_{c_i} determined by the curve-fit for the appropriate value of x found from the measured distance and angle of each segment used for the articulation angle measurement. The standard error is shown as Equation (4-13) and the slope m is converted to an articulation angle $\phi_{articulation}$ by Equation (4-14).

$$SE = \sqrt{\frac{\sum_{i=1}^N (y_i - y_{c_i})^2}{v(\sum_{i=1}^N (x_i - \bar{x})^2)}} \quad (4-13)$$

$$\phi_{articulation} = \tan^{-1}(m) \quad (4-14)$$

Using Equations (4-12) and (4-13), Equation (4-14) can be rewritten as

$$u_{curve-fit} \approx \frac{1}{2} |\tan^{-1}(m_k + u_{slope_k}) - \tan^{-1}(m_k - u_{slope_k})|, \quad (4-15)$$

which provides the uncertainty of the articulation angle measurement due to uncertainty in the relative slope between the measured trailer surface and LiDAR coordinate frame. Equation (4-15) only describes the raw articulation angle measurement uncertainty. This value is influenced by individual segment performance, trailer wall surface, and trailer box geometry. The trailer wall surface is not perfectly flat, however, deviations in the surface were captured by the uncertainty found as the standard error in the linear curve fit. Uncertainty in the trailer box geometry – i.e., how perpendicular the front wall is to the longitudinal axis – is assumed negligible compared to the uncertainty in the LiDAR’s measurements.

A two-step calibration process was used for the LiDAR sensors. The first step addresses common errors found for individual segments and corrects these offsets to reduce the standard error, and thus curve-fit uncertainty of raw articulation angle measurements. The second calibration step accounts for sensor performance in varying position and orientation relative to the trailer’s walls during a maneuver. The second calibration step converts the raw articulation angle measurement to an estimate of the actual articulation angle. In total, Equation (4-16) defines the articulation angle measurement uncertainty and will be detailed at the end of this section. Notice that in addition to the uncertainty in the curve-fit and calibration, there is uncertainty in the sensor mounting. Sections 4.3.2 and 4.3.3 detail the methods that used to improve the sensor performance and calibration process. Section 4.3.5 details the uncertainty associated with mounting the sensor on each trailer.

$$u_{art} = [u_{curve-fit}^2 + u_{calibration}^2 + u_{mounting}^2]^{\frac{1}{2}} \quad (4-16)$$

4.3.2 Improving LiDAR Performance by Increasing Trailer Surface Diffuse Reflectivity

The LiDAR sensor determines distance by emitting a 940 nm light pulse and measuring the time difference between the emitted source and each segment's detection of that light pulse after the pulse is reflected off an object. This is known as a time-of-flight (TOF) sensor. As with all TOF sensors, the accuracy and precision of measuring distance is influenced by the material surface of the detected object as well as the incident angle of the emitted light pulse to that object. As such, a TOF sensor often struggles to accurately measure the distance of a material with low diffuse reflectivity, such as a mirror, when the emitted light is at a high incident angle. Conversely, the distance to a material with high diffuse reflectivity, such as a cloth, can be more easily measured at high incident angles. This was evident when testing different materials to enhance articulation measurement accuracy with the LeddarTech® IS16. The most consistent and accurate results were found when attaching a black cloth tightly against the trailer walls observed by the LiDAR sensor as shown in Figure 4-20. Figure 4-21 illustrates the difference between specular, spread, and diffuse reflection.

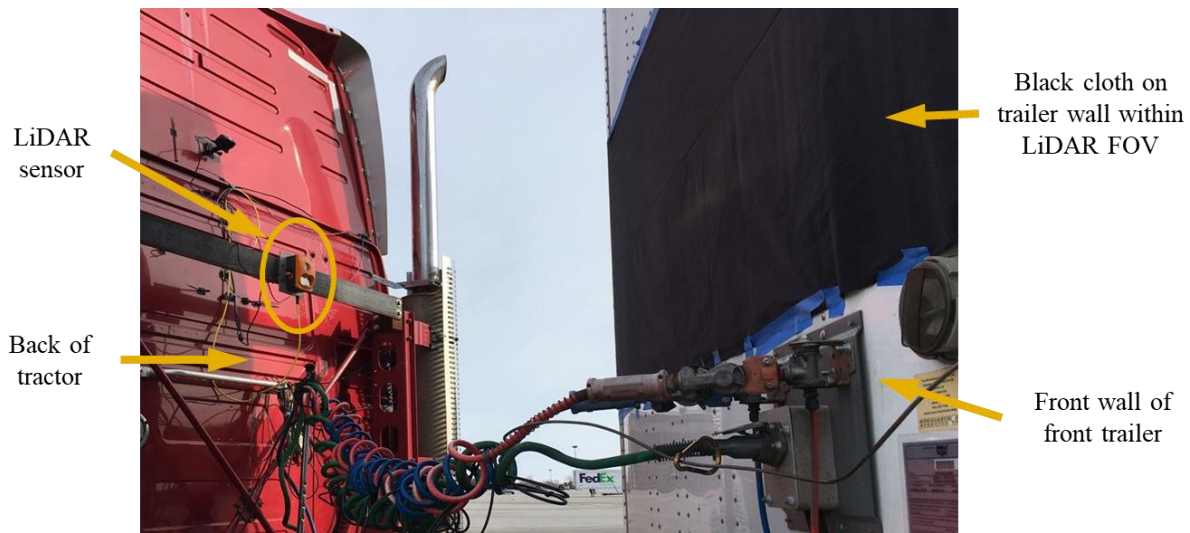


Figure 4-20. Front trailer wall with black cloth used to increase surface diffuse reflectivity to improve articulation angle measurement accuracy.

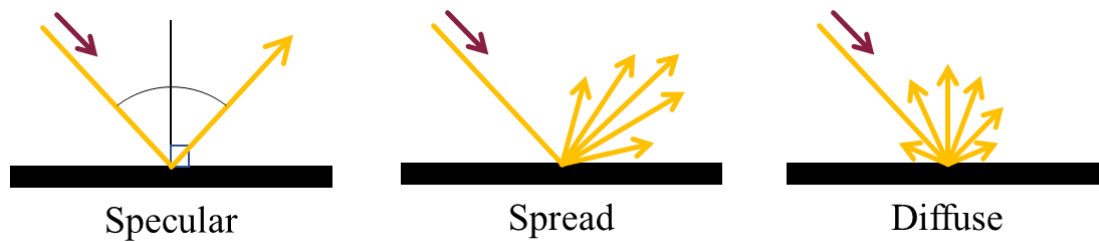


Figure 4-21. Illustration of different surface reflectivity patterns [19].

4.3.3 Calibrating LiDAR Sensors for Articulation Angle Measurements

To convert raw LiDAR distance measurements into accurate articulation angles, a two-step calibration process was created used. The first step was designed to define individual segment performance and reduce standard error – uncertainty of the linear-regression curve-fit – of raw articulation angle measurements. Each segment’s distance measurement accuracy was tested for a wide range of sensor orientations in a controlled environment. The second step was designed to correlate measurement data from the trailer walls to the actual articulation angle. This was done using a jig on a trailer to synthesize articulation between the tractor and front trailer. The jig was made of identical geometry to the tractor with respect to the trailer kingpin that it rotated about. After applying the calibration’s first step to reduce standard error on measured data from the second calibration step, a relationship was determined from the LiDAR’s measured articulation angle to the actual articulation angle. The second calibration step also provided insight on how to handle high articulation angles that result in some segments observing the trailer corner or sidewall or even some segments not observing the trailer at all. The sensor was designed for indoor and outdoor use and did not exhibit performance differences in direct sunlight versus in shade for the range of distances used for articulation angle measurements. Because of this, the first calibration step was conducted inside where the test conditions could be more accurately controlled.

LiDAR calibration step 1: accounting for segment-specific performance characteristics

For the first calibration step, the same cloth attached to the trailers was spread across a flat, polished concrete surface. All wrinkles were smoothed out and weights placed at each corner kept the cloth flush against the seamless concrete surface as shown in Figure 4-22. The LiDAR sensor was mounted to a rectangular aluminum arm fixed to a horizontally positioned rotary table about 1 m above the ground. The rotary table fingers were positioned such that the axis of rotation was directly in line with the LiDAR sensor’s local coordinate origin. The LiDAR sensor was first oriented perpendicular to the ground surface using a digital level as shown in Figure 4-22.

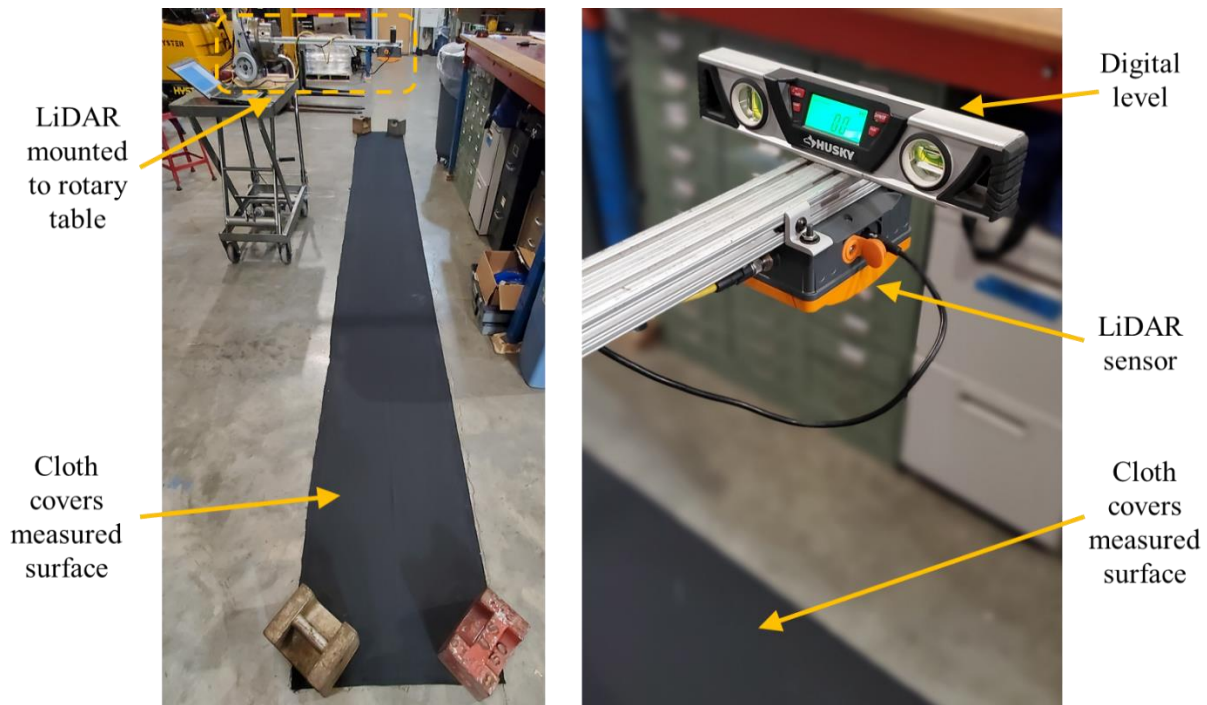


Figure 4-22. LiDAR sensor segment-specific calibration setup.

Using the rotary table to accurately position the LiDAR orientation, the sensor was rotated up to $\pm 50^\circ$ from the original horizontal position. Distance errors for segments 1 and 16 varied significantly throughout the tests. This is likely because the emitter intensity is much lower near the FOV boundaries and because segments 1 and 16 experience the highest incident angles of all segments. Segments 2 through 15 exhibited more consistent performance and were less affected by the LiDAR orientation with respect to the ground surface. After removing outlier data points, each segment's mean distance error from all tested angles was used as the calibration offset. Figure 4-23 plots the range of errors for each segment of all tested angles, sensors, and distances during the first calibration step. The mean distance errors were determined after removing outlier data points and are recorded in Table 4-3. Data from different sensors and tested distances were combined because the results were similar and suspected to be dominated by the tested LiDAR model's physics and algorithms rather than systematic error in the test procedure.

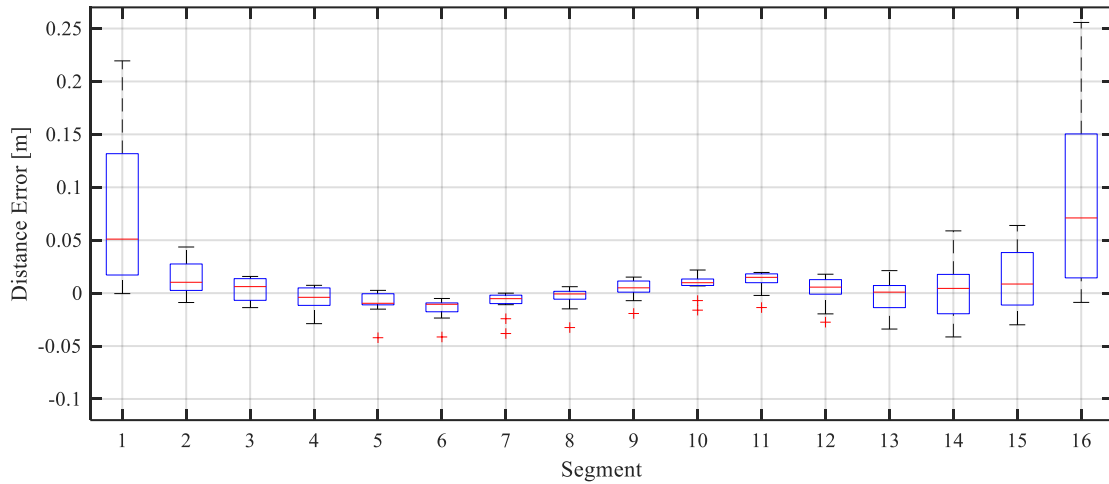


Figure 4-23. Segment-specific distance measurement error.

Table 4-3. Calibrated LiDAR distance offset for each segment.

Segment	1	2	3	4	5	6	7	8
Offset [m]	0.0752	0.0192	0.0083	0.0015	-0.0028	-0.0088	-0.0028	0.0013
Segment	9	10	11	12	13	14	15	16
Offset [m]	0.0089	0.0121	0.0176	0.0069	0.0008	0.0009	0.0078	0.090

Subtracting the mean distance error from raw measurements significantly reduces the articulation angle standard error for all tested angles. Furthermore, excluding the highly variable segments 1 and 16 reduces the curve-fit uncertainty even more. Uncertainty in the rotary table, digital level, and LiDAR mounting to the rotary table are all well below the uncertainty in the LiDAR's measurement characteristics and thus may be neglected. Figure 4-24 plots the first calibration step angle measurement errors at the 95% confidence level of the raw distance measurements using all 16 segments and the calibrated distance measurements using segments 2 through 15. This figure clearly illustrates the benefits of the calibration via segment-specific error correction and eliminating segments 1 and 16 from the orientation angle measurements.

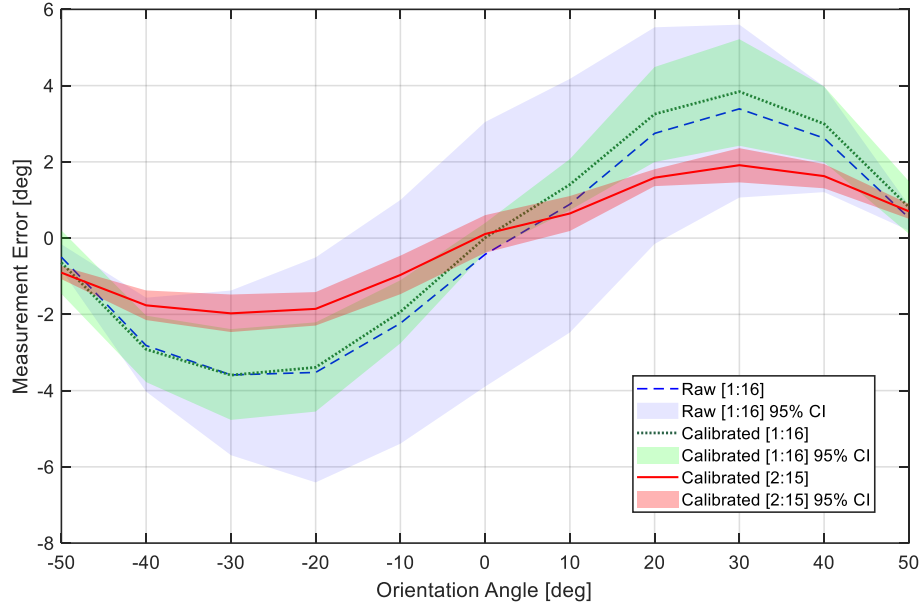


Figure 4-24. Angle measurement error comparison of raw and calibrated LiDAR data. Segments 1 & 16 demonstrated poor performance and were not used directly for articulation angle measurements.

Figure 4-24 clearly indicates sinusoidal bias for nonzero articulation angles, even after applying the offset correction. This bias is acceptable here because it is specific to the test procedure and any similar type of bias in on the tractor-trailer will be accounted for in the second calibration step with the trailer and test jig. It is expected that the bias will be slightly different when measuring off of the trailer surface because: (1) unlike the first calibration step where the sensor is at the same distance for all angles, articulation between the tractor-trailer articulates moves the sensor closer and further away from the trailer, and (2) not all segments will observe the same trailer wall at the same time, so a correction must be made for whichever segments observe the same wall at a given time. In summary, the first calibration step reduces uncertainty in the linear curve-fit applied to distance measurements when estimating the relative articulation between the LiDAR sensor and measured trailer body.

LiDAR calibration step 2: accommodating trailer geometry

The second calibration step replicated articulation between the tractor and front trailer with a jig of identical geometry to the LiDAR's position on the tractor relative to the trailer's kingpin that it rotated about. The jig positioned the LiDAR sensor at the following articulation angles: 0° , $\pm 10^\circ$, $\pm 20^\circ$, $\pm 30^\circ$, $\pm 40^\circ$, $\pm 50^\circ$, $\pm 55^\circ$, $\pm 60^\circ$, $\pm 65^\circ$, $\pm 70^\circ$, and $\pm 75^\circ$. Figure 4-25 (a) illustrates a top-down view of the jig's geometry and how the articulation angle was controlled by the length of vector $\overline{BC}$ while lengths $\overline{AB}$ and $\overline{AC}$ were fixed. Figure 4-25 (b) shows a picture of the jig staged on the tested trailer. Black tape on the trailer marks

the perimeter of where the cloth was attached and held flush against the trailer surface. The test jig also provided insight on how to determine the location of the trailer corner strictly from the LiDAR distance measurements, and more importantly, generate a calibration curve to convert measured articulation angle to actual articulation angle. For the off-tracking tests, all segments were used to identify the trailer corners and sidewall at high articulation angles, but only segments 2 through 15 were used for the articulation angle measurements.

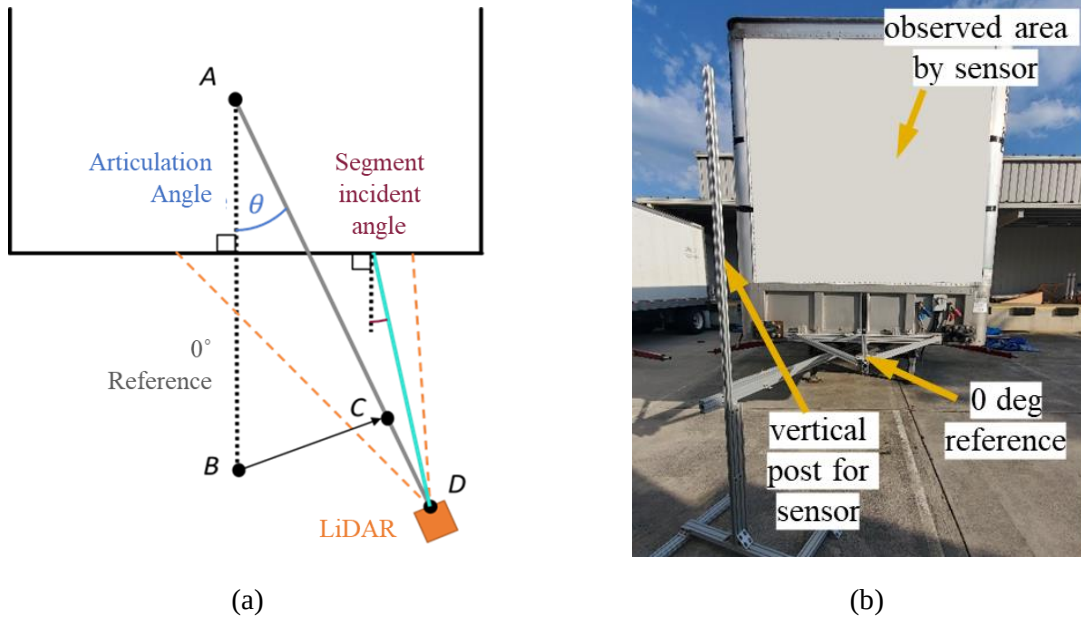


Figure 4-25. (a) Illustration of top-down view for jig geometry. (b) Jig staged at trailer for step 2 of LiDAR calibration process.

Even with the cloth applied to the trailer wall surface within the LiDAR's FOV, there are additional challenges in measuring the articulation angles. Specifically, when the front trailer is at a high articulation angle, the trailer's corners appear within the LiDAR's FOV. Because these corners are rounded, any measurements taken from those corners must be omitted from the articulation angle estimate because they negate the assumption that the trailer is a rectangle and skew the linear curve-fit used to estimate the articulation angle. Furthermore, at very high articulation angles the LiDAR sensor may have some segments observing the trailer's front wall and others observing the trailer's sidewall. These conditions are illustrated in Figure 4-26. After extensive review, only the articulation angle measurements from the front wall were used for determining the articulation angle because too few data points observed the sidewall for the highest articulation angles experienced in the off-tracking test ($\sim 60^\circ$) to achieve accurate articulation estimates.

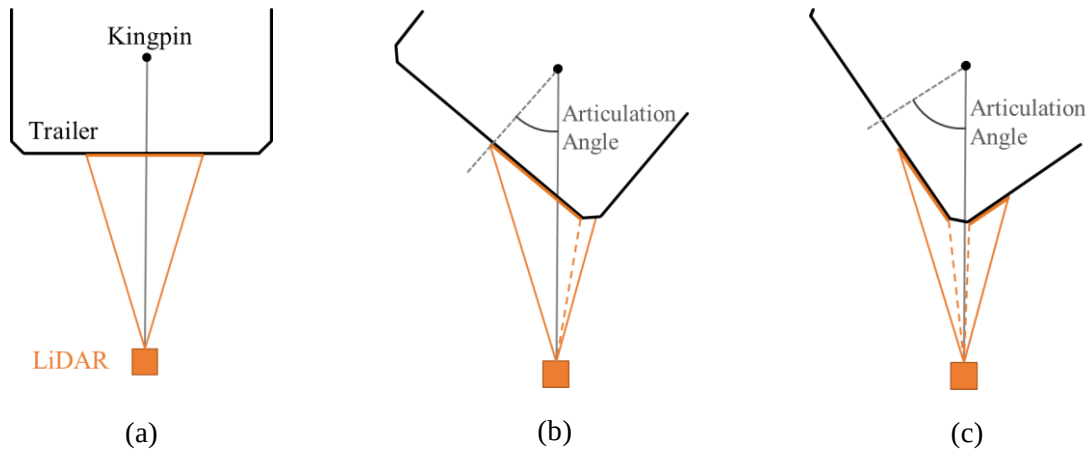


Figure 4-26. Articulation angle LiDAR choosing appropriate segments for the surface walls for determining the articulation angle. (a) All segments observe the front wall. (b) Some segments observe the trailer corner and are omitted. (c) Both front wall and side wall are observed.

Lastly, the LiDAR sensor segments' performance characteristics were used to account for small misalignments in the jig's physical makeup. Although the jig and trailer dimensions were triple-checked before conducting tests, there appeared to be a misalignment in the LiDAR's orientation relative to the trailer. Unfortunately, at the time this was realized, the test jig had already been disassembled for use on a separate project. To create an accurate lookup table that converts the LiDAR's estimated articulation angle and the actual articulation angle, the LiDAR sensor must be assumed to be oriented nearly-perfectly for the calibration procedure. From Figure 4-25 (a), the lengths of segments $\overline{AB}$, $\overline{AC}$, and $\overline{BC}$ were easily constrained with high accuracy, and the trailer's geometry was also easily measured with accuracy. It was found that the source of misalignment occurred at the LiDAR's orientation relative to $\overline{AC}$ and the position of the 0° reference relative to the trailer front wall as illustrated in Figure 4-27. After applying the calibration offset to each segment as described by the first calibration step, each segment's distance error with respect to the LiDAR's orientation angle relative to the measured surface was assumed approximately equal to the mirrored segment's distance error for the negative of that orientation angle. For example, the distance error of segment 3 at an articulation angle of 30° was approximately equal to the distance error of segment 14 at an articulation angle of -30° . By taking the average difference in error for all segments compared to their mirrored counterpart at the tested articulation angles, a metric representative of the symmetry was developed according to Equation (4-17). In an ideal testing condition and sensor, the symmetry metric, ψ , would equal 0. For reference, the orientation symmetry error from the static test jig with the rotary table and singular target surface was 0.046 ft and 0.029 ft before and after including the first calibration step, respectively. It is expected that after correcting for the orientation and swingarm

reference offsets for the articulation angle jig on the trailer, the symmetry metric should be close to but likely larger than 0.029 ft. In Equation (4-17), N_s is the total number of segments (16), N_a is the number of positive angles used in analysis, $\varepsilon_{s,a}$ is the distance error of segment s at angle a . Segments 1 and 16 are omitted from the symmetry metric calculation due to their high variability and because they will not be used for the articulation angle measurement; hence why s starts at 2.

$$\psi = \frac{2}{(N_s - 2)N_a} \sum_{a=1}^{N_a} \sum_{s=2}^{N_s/2} |\varepsilon_{s,a} - \varepsilon_{N_s+1-s,-a}| \quad (4-17)$$

The symmetry metric was calculated for a wide range of corrections to both the LiDAR's orientation relative to the swingarm and the 0° reference. The combination of -2.45° LiDAR orientation and 0.35° reference offset corrections resulted in the minimum symmetry error at 0.037 ft and were implemented to correct for jig misalignments in post-processing and determine the relationship to convert the estimated articulation angle from LiDAR measurements to the actual articulation angle. These angle corrections seemed appropriate for the capabilities in mounting and positing the LiDAR and test jig. Figure 4-28 plots the measured data points before and after the jig corrections on top of the actual trailer walls.

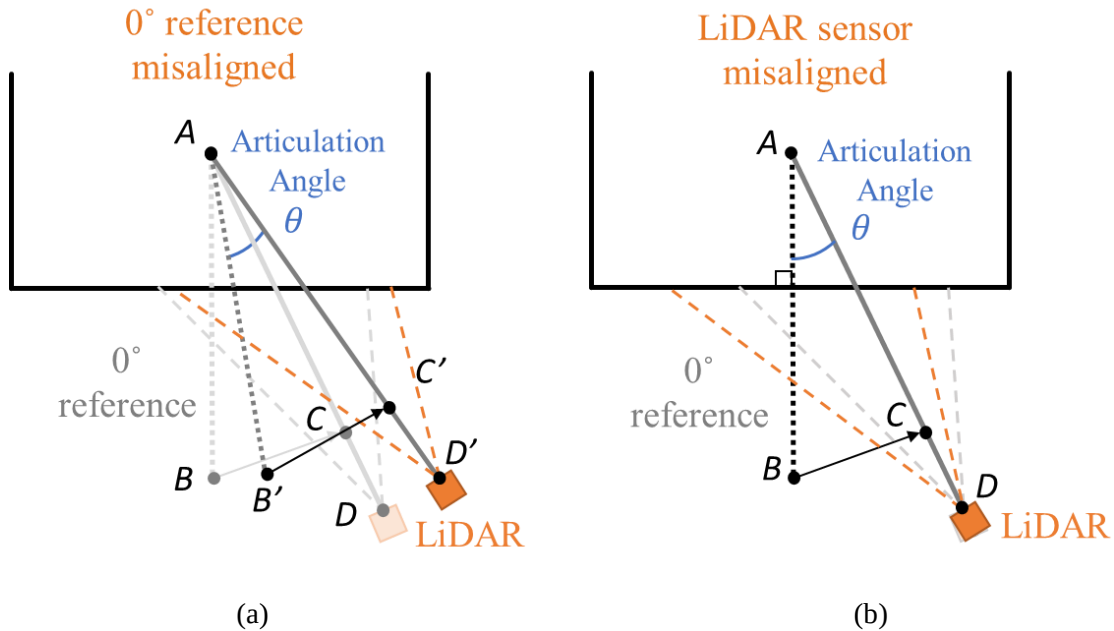


Figure 4-27. Jig misalignment of (a) 0° reference and (b) LiDAR sensor orientation relative to swingarm (AC).

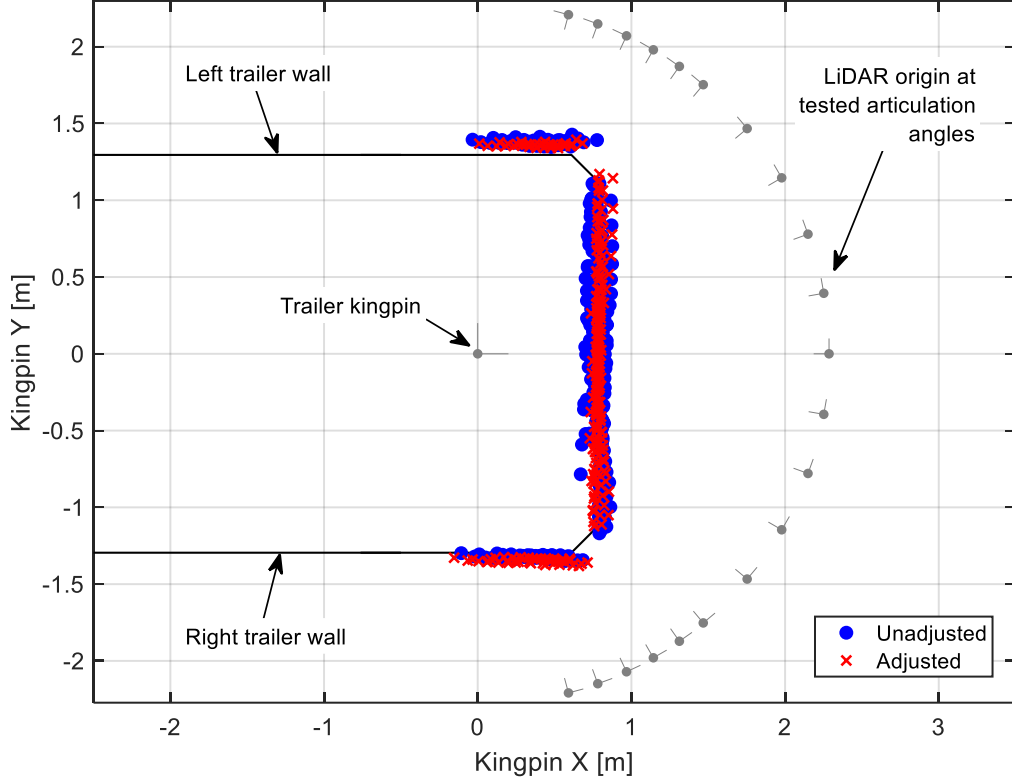


Figure 4-28. Comparison between corrected (adjusted) and unadjusted measurements for 2nd calibration step.

Once the jig alignment was accounted for and the segment-specific distance offset calibration was included, curve-fits were applied to the actual articulation angle with respect to the measurement's estimate. A unique curve-fit was applied for each wall surface. These curve-fits were found from the measured data of three trials at each articulation angle that observed the trailer wall of interest. The slope for each curve fit was near 1 and $R^2 > 0.90$, suggesting good fit. The uncertainty of the curve-fit is a function of both the number of data points and the spread and scatter of data points. Equation (4-18) provides the uncertainty of curve-fits, where the $t_{v,P}$ is the t-distribution value for the specified degrees of freedom v and the confidence level P taken at 95%. The degree of freedom is provided as Equation (4-19) where m is the polynomial curve-fit order which, for a linear regression, is 1. In Equation (4-18), y_{c_i} is the curve-fit's estimated value for a given (x_i, y_i) pair [17,26].

$$u_{calibration} = t_{v,P} \left[\frac{\sum_{i=1}^N (y_i - y_{c_i})^2}{Nv} \right]^{\frac{1}{2}} \quad (4-18)$$

$$v = N - (m + 1) \quad (4-19)$$

The uncertainties for the front, right, and left trailer walls are as follows: 0.286° , 1.467° , and 0.807° , respectively. It is unsurprising that the front wall has the lowest uncertainty because it has significantly more data points than either side wall. Figure 4-29 presents the test data for all three trials at the tested articulation angles after accounting for jig misalignment and incorporating the segment-specific offset corrections to the LiDAR data. Each dataset corresponding to individual trailer walls includes the linear regression curve-fit, its R-squared value, and uncertainty. These uncertainty values were applied to the data translation from measurement to estimated articulation angle throughout testing. Because the highest articulation angles within testing were around 60° , only data from the front wall was used due to its lower uncertainty. In short, Figure 4-29 provides the uncertainty values for each trailer wall surface as measured by LiDAR on the tractor. These values were prepared for processing test data, however, only the front wall curve-fit and uncertainty were necessary. Because the front geometry of the different trailer configurations – 28-ft double, 33-ft double, and 53-ft single – are nearly identical, it was safe to assume the curve-fits apply to all tested trailer configurations.

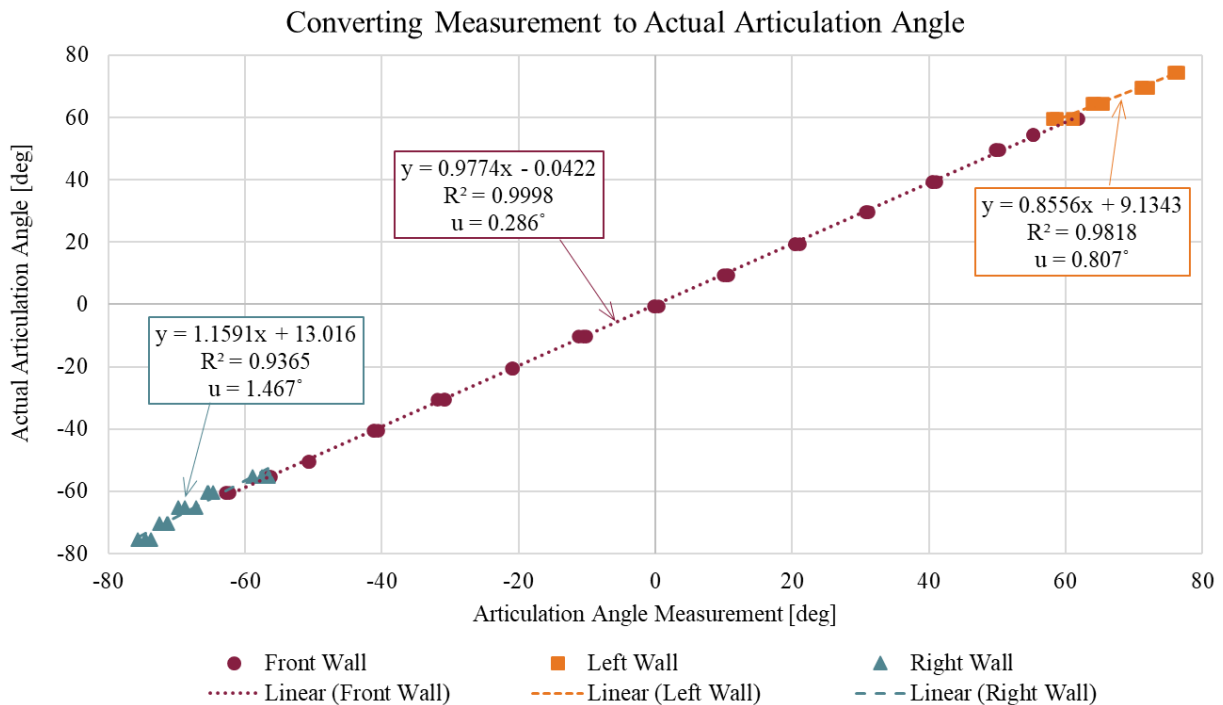


Figure 4-29. Curve-fits for converting LiDAR articulation angle measurement to actual articulation angle.

4.3.4 Determining LiDAR Location for Trailer-Trailer Articulation Angle Measurement

Before mounting the rear LiDAR sensor to measure the articulation angle between the front and rear trailers, a simulation was conducted to determine if the LiDAR sensor should be mounted at the front of the rear trailer facing forward or mounted at the rear of the front trailer facing rearward. The goal was to determine which position provided more segments observing the same trailer wall throughout each maneuver to provide accurate articulation angle measurements. This process was motivated by the fact that the standard error decreases as the number of data points increases. The simulation was based on the SGM-C numerical tractrix algorithm (Chapter 3) of the 33-ft double to estimate the relative articulation between each vehicle unit for the four test maneuvers, then determine which of the 16 segments detected the front (or rear) trailer face and which ones possibly detected the corner or sidewall. The 50-ft J-turn maneuver resulted in the highest articulation between vehicle units and, as illustrated in Figure 2-14, it was clear that mounting the rear LiDAR sensor on the rear of the front trailer facing rearward allowed more segments to observe the rear trailer's front wall without having to selectively choose which segments could be used at a specific time within the maneuver. Figure 4-31 shows the angles of each segment's edges and the angle from the sensor to the rear trailer's left (driver) and right (passenger) corners. When mounting the sensor on the back of the front trailer, nearly all segments may be used at all times, even for the highest predicted articulation angles. This led us to positioning the LiDAR sensor on the rear of the front trailer.

Segments 3 through 14, inclusive, were always used for determining the articulation angle between the front and rear trailers. Segments 1 and 16 were omitted for their variability as discussed previously and segments 2 and 15 were omitted because there could be instances that the trailer corners are observed due to higher-than-expected articulation angles or slight misorientation in sensor mounting. Figure 4-30 illustrates the trailer-trailer geometry and coordinates used to determine trailer corners with respect to the LiDAR positions of interest throughout a simulated maneuver. Coordinate frames A and B are the LiDAR origins at the front and rear trailer, respectively. The articulation angle between the two trailers is θ which is equal to the sum of articulation between each trailer and the dolly, φ_1 and φ_2 . The vector of the rear trailer's front wall corners location with respect to the front trailer's rearward facing LiDAR coordinate frame is described by Equation (4-20) where H_{AO_2} is the homogeneous transformation matrix from frame O_2 to frame A . This equation stands for each trailer corner and simulated LiDAR but with different transformation matrices and vectors r for the corresponding trailer corners and LiDAR frame.

$${}^A r_{P_2} = H_{AO_2} {}^{O_2} r_2 \quad (4-20)$$

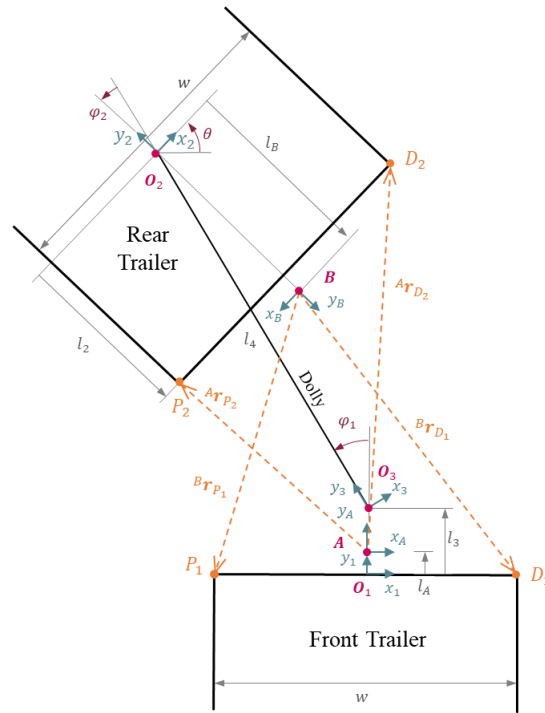


Figure 4-30. Geometry of trailer-trailer articulation used to determine trailer corners with respect to LiDAR frames.

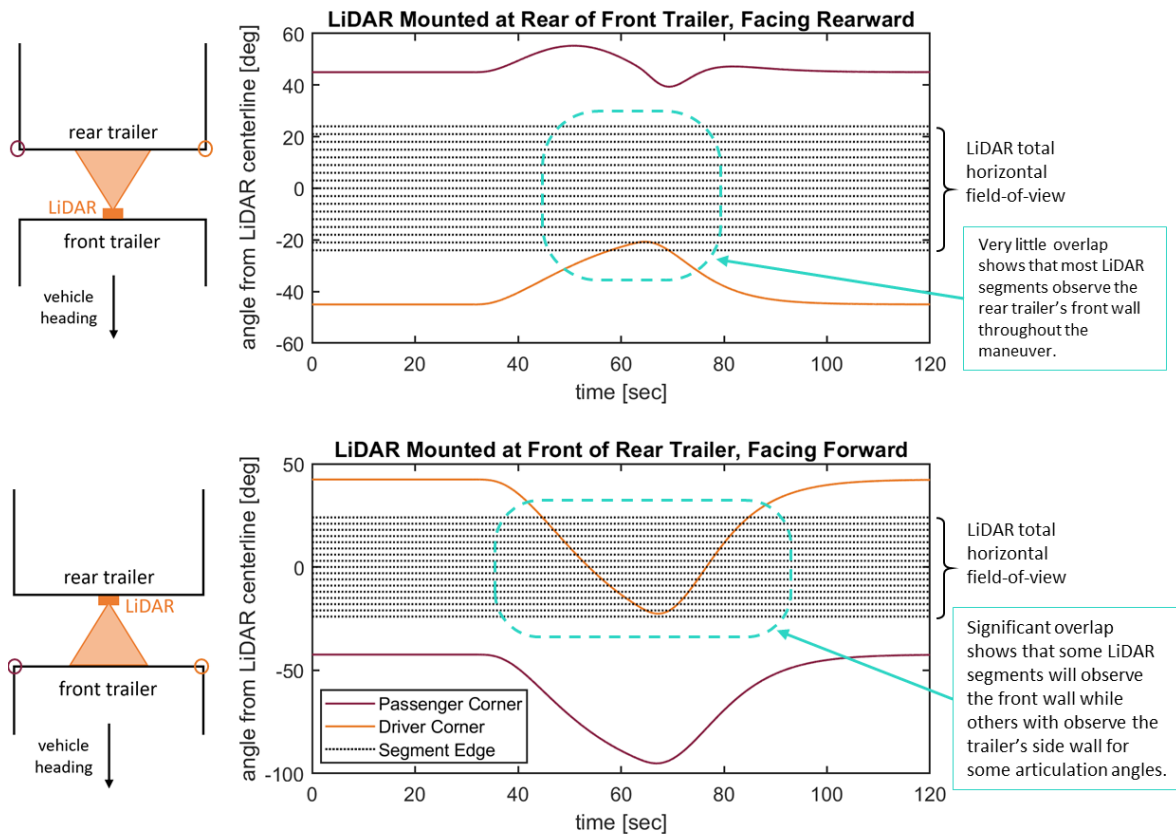


Figure 4-31. Graphs show the LiDAR segment edges and the observed trailer's driver (left) and passenger (right) side corners as the vehicle travels through the tight-turn maneuver.

The rear of each test trailer was retrofitted with a LiDAR mount as shown in Figure 4-32. The mount was rigidly fixed to the trailer and kept the sensor protected from impact when loading material into and out of the trailer. The geometry between the front and rear trailer is similar to the geometry between back of the tractor cab and the front trailer. Because of this, it is safe to assume that the curve-fits and uncertainty derived from the test jig described in the previous section are applicable to the trailer-trailer articulation angle measurements.

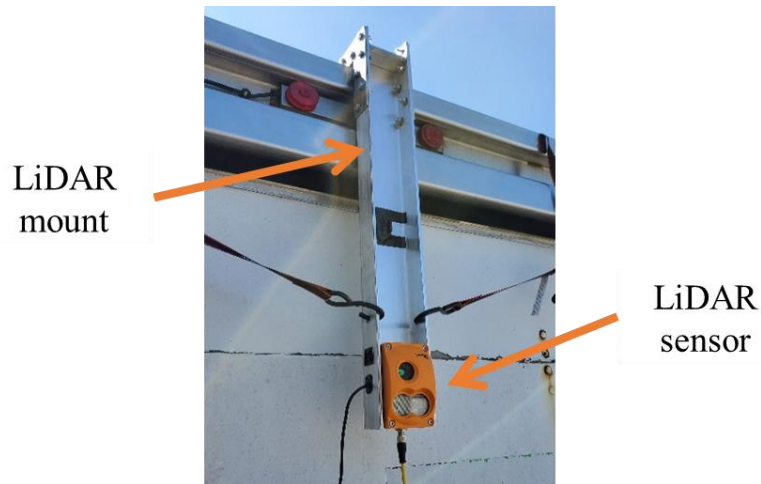


Figure 4-32. LiDAR mount on the back of a front trailer.

Using LiDAR to measure articulation between the two trailers cannot directly determine how much of the measured articulation is contributed by the dolly and how much by the rear trailer. Because the dolly's wheelbase is much shorter than the trailer's, the dolly is quicker to rotate than the trailer resulting in a condition where the dolly is articulated in one direction and the trailer in the opposite direction during complex steering maneuvers like that found in roundabouts. The LiDAR's articulation measurement is simply the summation of these two articulations – front trailer to dolly and dolly to rear trailer – and the percentage of the articulation contributed by the rear trailer was estimated by the SGM-C numerical tractrix algorithm at that given instance. This percentage was then used to estimate the dolly and rear trailer individual articulations from the measured total articulation between the front and rear trailers.

4.3.5 Accounting for Sensor Mounting Alignment on Vehicle

Lastly, before the trailer paths were determined by the measured articulations, small misalignments in mounting the LiDAR relative to the tractor and trailer centerlines must be accounted for. For example, the tractor's LiDAR mount was designed differently than the mounts attached to the rears of the A-double

trailers, and it could not be guaranteed that the mounts were installed perfectly square to the tractor and trailers’ geometries and along their lateral centerline. An appropriate offset correction and its corresponding uncertainty was defined for each mount.

At the beginning of each maneuver and trial, the tractor and trailers were driven in a straight-line for as long as space permitted at the test site so that the articulation between each vehicle unit could approach 0°. Figure 4-33 plots the spread of articulation angles at the beginning of each appropriate trial. The three separate spreads of data are designated for the three sensors mounts: back of the tractor, back of the 28-ft double front trailer, and back of the 33-ft double front trailer. For some trials, it could not be guaranteed that the trailers reached zero articulation due to slight changes in the testing environment near the entrance of the staged maneuvers. These were often caused by cars or pedestrians passing through the test site unannounced. Luckily, this could be easily observed in the data and is not expected to impact the peak off-tracking measurement that occurs much later in the maneuver. These outliers were selectively omitted and are not included in the mounting correction estimate and uncertainty, nor within Figure 4-33. From each spread of articulation angle measurements, the mean value was taken as the correction offset that was applied to all articulation angle data for the appropriate mounted location. Similarly, the uncertainty from these values was found by the 95% confidence interval via standard deviation of the data set and an appropriate Student’s *t* variable. Table 4-4 lists the mounting correction and uncertainty for each LiDAR mount.

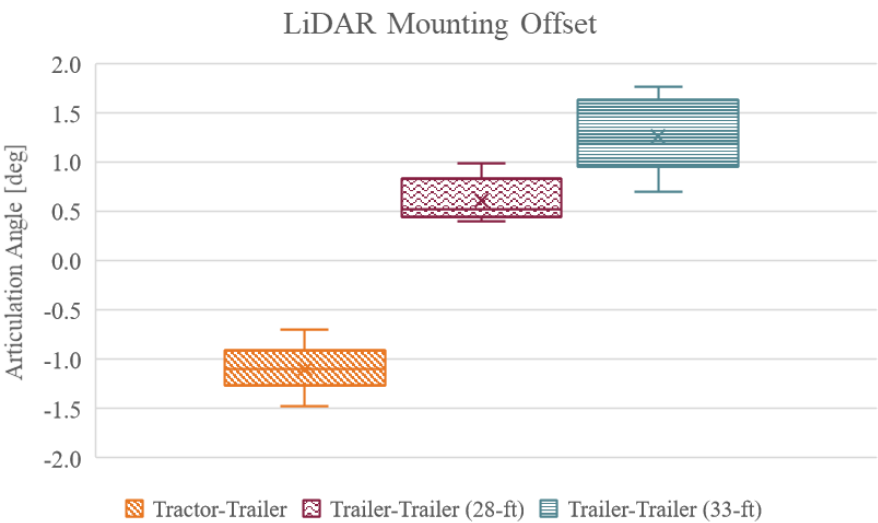


Figure 4-33. Articulation angle LiDAR mounting offset as determined by the estimated zero articulation angle measured near the beginning of each maneuver. X marks the mean.

Table 4-4. Articulation angle offset correction added to the articulation angle measurements to remove error from LiDAR mounting orientation.

Mount Location	Tractor-Trailer	Trailer-Trailer (28-ft)	Trailer-Trailer (33-ft)
Mount Correction	-1.10°	0.61°	1.26°
Uncertainty (95%)	0.45°	0.47°	0.81°

4.3.6 Time Synchronization of LiDAR Data

Each LiDAR sensor requires a separate software application on the data acquisition laptop. GPS data was collected by a data acquisition system operated by the laptop, but both the laptop and data acquisition system had unique internal time frames. Luckily, the LiDAR data filenames included the laptop's time down to milliseconds at the start of data collection. After comparing GPS and other sensor data on the data acquisition to the LiDAR data, approximate values for time differences between the laptop and DAQ were determined for each dataset. Then, in post-processing, each individual trial's LiDAR data was manually adjusted forward or backward in time by a couple samples relative to the GPS data such that the rising slope, peak, and downward slope of the measured and estimated articulation angles overlapped. This time shift can be influenced by error in estimating the tractor's position along the steer axle path for a given timestamp but is believed to be sufficiently small that it may be neglected. It is estimated that the time LiDAR time synchronization uncertainty with respect to the position along the steer axle path was 0.2 sec, or 5 samples at the LiDAR's 25 Hz sample rate. When traveling at 3 mph, this equates to an uncertainty of 0.88 ft along the steer axle path. This value was multiplied by the peak off-tracking sensitivity index with respect to temporal error in articulation angles. Because the sensitivity index for articulation angle measurement synchronization was relatively small compared to other parameters as shown earlier in Table 4-1, this estimate of LiDAR time synchronization uncertainty was acceptable.

4.4 Estimating Tractor Steer Axle Path

The test tractor was instrumented with a GPS receiver mounted on the hood directly above the steer axle center. The Garmin 18x-5Hz receiver uses the Wide Area Augmentation System (WAAS) for position accuracy of 10 ft (95% typical) in good conditions and steady-state RMS speed accuracy of 0.115 mph. GPS accuracy can be significantly influenced by atmospheric conditions, the number of satellites detected by the receiver, physical barriers or reflections of satellite signals caused by environment surroundings such as buildings, and the location of detected satellites with respect to the receiver. The accuracy values listed previously are provided for ideal conditions – clear skies, no obstructions, and detection of many

satellites. Testing was conducted in a large open parking lot and maneuvers were positioned about the center of the lot so that nearby buildings would have little to no impact on GPS accuracy. During tests, skies were partly cloudy with some haze. For the test conditions, it is fair to assume that the GPS position and speed data are expected to be near or within their specified accuracy. Position accuracy is already provided at the 95% confidence level and may be used directly for uncertainty analysis. However, this confidence level assumes normality for an infinity number of data points and thus, for a finite set of data points where drift errors of GPS data persist, this uncertainty must be implemented with caution. Speed accuracy is provided as the steady-state RMS and is equivalent to the error standard deviation when speed is constant. Because tests were conducted such that speed was nearly constant throughout all tests, it is fair to use this value as the standard deviation in measured steer axle speed. The measured speed uncertainty at 95% confidence level is then simply found as Equation (4-21).

$$u_v = 1.96(\sigma) = 1.96(0.115 \text{ mph}) = 0.225 \text{ mph} = 0.331 \text{ fps} \quad (4-21)$$

As defined in Section 4.2, there are estimated levels of uncertainty of the steer axle's path due to uncertainty in physically marking and positioning the camera path and by following the path using the live-feed camera. The tractor's position along the assumed target path is estimated by integrating the GPS speed along that path. For a single point, GPS speed data is inherently more accurate than GPS position data. However, the process of integrating GPS speed with respect to time means that position uncertainty for a given speed measurement uncertainty grows unbounded with time as described by Equation (4-22). Notice that this is the same formulation from Equation (4-1) where T replaces the sensitivity index λ .

$$u_{distance} = \frac{d}{dv} \left(\int_0^T V dt \right) u_v = T u_v \quad (4-22)$$

If the driven path is a perfectly straight line at a constant speed, position uncertainty along that line via integration of speed surpasses position uncertainty from the position data directly after traveling for 30 sec. The longest tested maneuver – the roundabout – requires about 120 sec to complete resulting in nearly 40 ft of uncertainty in the estimated distance traveled along the estimated steer axle path. This uncertainty can be reduced by filtering the data. The filter's objective is to eliminate high-frequency content in the GPS receiver's speed measurement under the assumption that the speed error fits a normal distribution. Most any low-pass filter could be applied here, but for simplicity and to uphold the assumptions applied in the uncertainty analysis, a moving-average was used. From a moving average, the speed uncertainty for a single data point is estimated by Equation (4-23) where s_x is the standard deviation of data points within

the window, m is the number of data points in each window, and $t_{v,p}$ is the t-distribution at the 95% level with degrees of freedom $v = m - 1$ [26].

$$u_{ma} = t_{v,p} \frac{s_x}{\sqrt{m}} \quad (4-23)$$

Of course, the moving average uncertainty will change at each data point because the standard deviation of data points within the window will change. By combining Equations (4-23) and (4-22), the traveled distanced uncertainty after the moving average LPF can be calculated numerically by Equation (4-24) where N is the total number of sampled data points for the maneuver and f_s is the constant sample frequency in hertz.

$$u_{distance} = \sum_{i=1}^N \frac{u_{ma_i}}{f_s} \quad (4-24)$$

Clearly, from Equations (4-23) and (4-24), uncertainty in the moving average decreases as the moving average window size increases. An appropriate window size must be carefully selected such that it is sufficiently small to not be impacted by steer axle speed transients while also sufficiently large to reduce uncertainty. A window size of 51 data points – 5 seconds forward and backward from the current index – was chosen through trial-and-error and visual inspection. Figure 4-34 plots all raw and filtered GPS speed data for each of the four maneuvers. Two trials during the roundabout maneuver were accidentally driven in 1st gear rather than 2nd so their speed is lower than all other trials. This difference is not expected to alter the final test results on axle paths and off-tracking because they still meet the negligible tire slip assumption for the tractrix model.

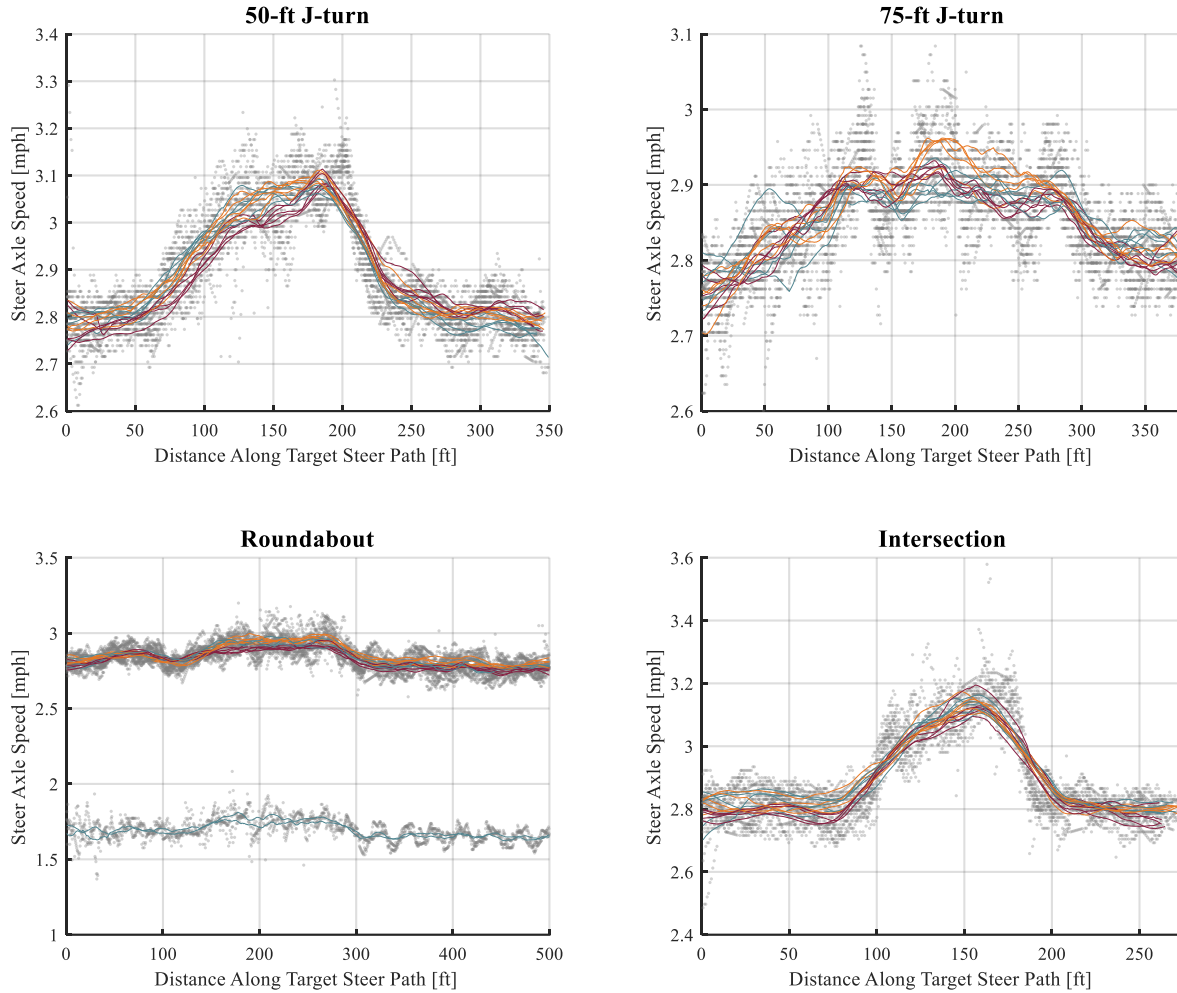


Figure 4-34. Steer axle speed as measured by GPS receiver. Raw data is shown as scatter of grey dots, filtered data with moving average is shown as smooth curves.

From Equation (4-24), the maximum uncertainty in estimating the tractor's steer axle position along the target path was approximately 4.4 ft and occurs at the very end of the roundabout maneuver. Although this value is relatively high, peak off-tracking and the area of most interest occurs well before the end of the maneuver. Table 4-5 lists uncertainty of the steer axle along the target path at the end of each of the four maneuvers. As expected, the shortest maneuvers, the J-turns, have the lowest uncertainty.

Table 4-5. Maximum position uncertainty along the target steer axle path via integrating filtered GPS speed.

Maneuver	50-ft J-turn	75-ft J-turn	Roundabout	Intersection
Max Uncertainty Along Path [ft]	1.90	1.86	4.37	2.08

GPS position data along the steer axle path can be used to qualitatively determine if the uncertainty values in Table 4-5 are appropriate and whether the actual path measured by GPS is indicative of the targeted steer axle path. Figure 4-35 presents the cloud of GPS position data for each of the four maneuvers as well as the target steer axle path and the uncertainty boundaries estimating from the manual process of laying the camera path and driver behavior while following the path (Section 4.2). Notice that the GPS data cloud encapsulates the target path for almost the entirety of all four maneuvers. The 50-ft J-turn and intersection maneuvers appear to have the best match between GPS position data and the target path. The straight-away exits on the 75-ft J-turn and roundabout appear to have a slight offset from the target path. This is likely acceptable because the sensitivity indices of peak off-tracking as shown earlier in Table 4-1 of Section 4.1.3 suggest that the peak off-tracking changes approximately by 4 in for every 1 ft that the steer axle path deviates laterally from the targeted path. At the point of peak off-tracking in these two maneuvers, the tractor steer axle is just slightly beyond the primary turn and traveling along the exit straight-away. At this point, there is good agreement between the GPS data and target steer axle path, and thus, lateral position deviations at the exits are inconsequential to the test results of interest. Likewise, the steer axle path error's impact on off-tracking reduces significantly further along the exit and away from the primary turn.

Figure 4-36 presents GPS position error from the target path after integrating the filtered GPS-measured speed along that path. Because the GPS position uncertainty is radial from the actual position, the error is always positive. All 15 trials for each of the four maneuvers are shown. The GPS position uncertainty according to the receiver's data sheet for best-case conditions is also included as reference. In all four maneuvers, most data points were below the receiver's ideal accuracy, suggesting that the target path was closely followed and that integrating the GPS speed along the target path is a viable method to estimate the tractor's position along that target path to ultimately synchronize the tractrix model and LiDAR data with a corresponding position along the target path.

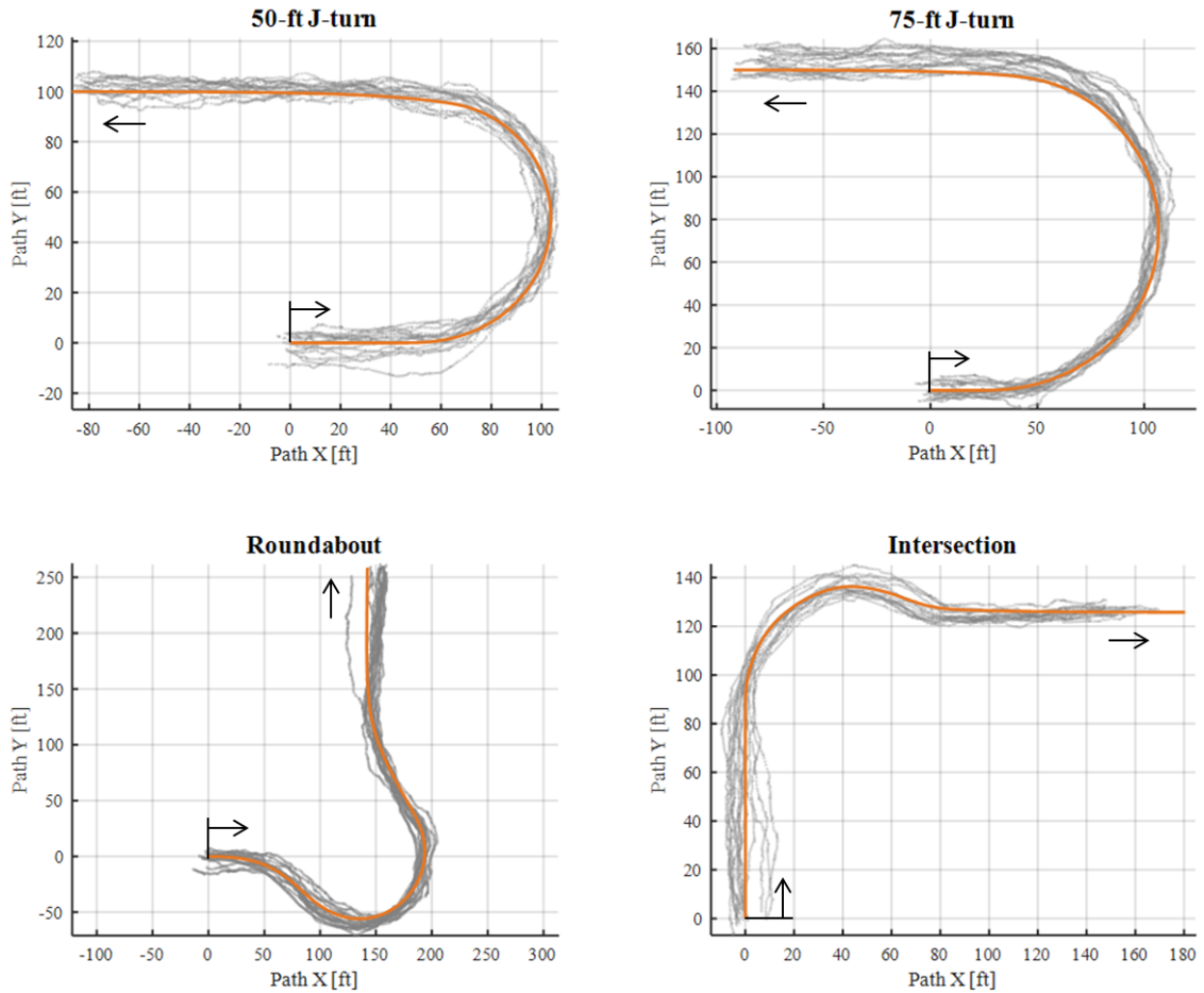


Figure 4-35. GPS position data cloud and target steer axle path for each of the four tested maneuvers.

As expected from the exit path drifts from the target path demonstrated in Figure 4-35, the position error increases toward the end of the 75-ft J-turn and roundabout maneuvers. Because the error appears more random on the 75-ft J-turn exit and fills the range from 0 ft to 15 ft, path deviation on the exit of the 75-ft J-turn is acceptable and cannot be directly concluded with accuracy from the GPS data alone. Position error toward the end of the roundabout maneuver, however, deviates completely above the 5 ft mark for all data points. This observation is indicative that the actual traveled path at the exit is clearly different than the intended path. Part of this error may be attributed to error accumulated from integrating speed to estimate distance, however from Figure 4-35 there is a clear lateral position error from the target path and cloud of GPS data. However, up until the exit, the GPS position error is relatively stable and covers a broad spectrum without a clear delineation from an average below the GPS position's uncertainty. For

these reasons, the increase in error at the end of the roundabout is acceptable because it does not impact the critical test result of interest – peak off-tracking – that occurs in the middle of the maneuver.

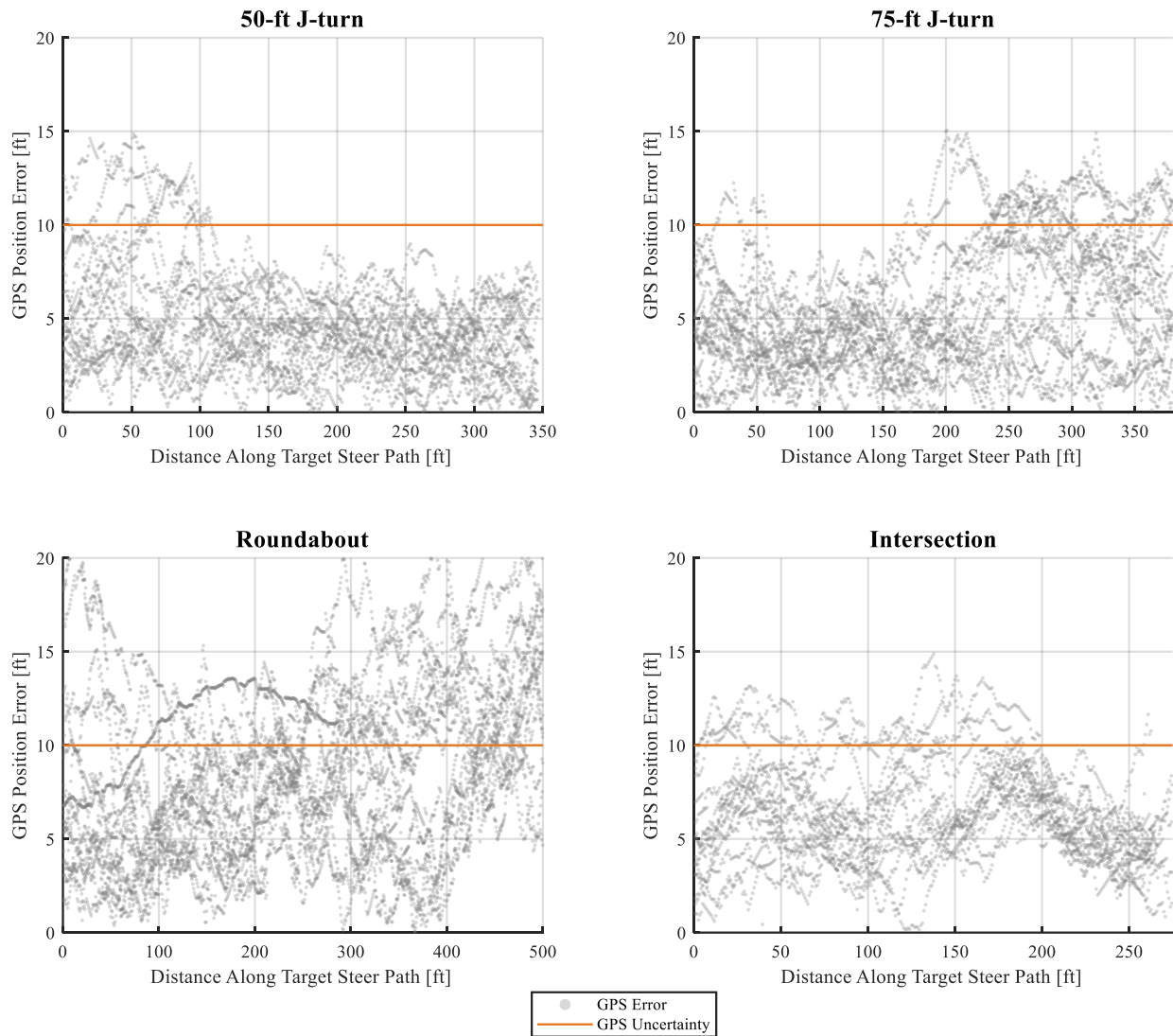


Figure 4-36. GPS position error from target path.

In summary, the tractor's steer axle position along the target path was estimated by integrating the filtered GPS speed (Figure 4-34) along the targeted steer axle path. GPS position data cloud about the target path shown as Figure 4-35 in conjunction with GPS position error shown as Figure 4-36 suggest that, throughout portions of the maneuvers of most interest, the actual steer axle path is close to the targeted steer axle path and that integrating speed to synchronize all data along the target path is an acceptable approach. If the target path was not known or if the GPS position error was consistently well beyond the

expected uncertainty bounds, then other approaches to better estimate the actual steer axle path would've been taken. For example, a principal curve analysis could be applied to GPS position data. 18] is one example of this method applied to sparse GPS data of pedestrian traffic. Principal curve analysis assumes that all data points are independent. For a single passing through the maneuver, this assumption is invalid because drift errors in GPS position data may be similar for all data points within a short window of time. If, however, data is collected more randomly throughout multiple days and different times of the day, then the independence assumption may be acceptable. Because each maneuver has 15 unique trials from different times of the day and for different days, this assumption may be acceptable. Another key step to principal curve analysis is the selection of a non-parametric smoothing function. There are many applicable options, and the selection is dependent on the data characteristics. Each data point may also be weighted individually so that known outlier data points have little or no impact on the resulting curve-fit.

Another viable method is the use of Kalman filters. There are several different approaches that may be implemented, and all are dependent on how the state equations are determined and what measurements are taken. For example, the drive axle speed and steering angle may be measured to predict a steer axle speed and position via a bicycle model. This could then be measured by the GPS speed and position for the correction step. There are countless approaches to this type of problem, however, for this study as demonstrated by the GPS position error and target path, it is fair to accept the target path as the actual steer axle path and synchronize data along that path by filtering and integrating the GPS measured speed. Additionally, the deliberate investigation of uncertainty in laying the camera path and then following it provide analysis necessary to accommodate deviations from the target path within the peak off-tracking value. Visual inspection of GPS position error alongside the GPS data cloud about the target path as presented in this section is a sufficient, but loose verification that the target steer axle path was followed within the limitations described by uncertainty of physically positioning the camera path, following the camera path, and assuming the tractor's orientation is described by the tractrix model.

4.5 Test Results and Uncertainty Propagation

For the test data, each trial's GPS speed was filtered and integrated along the target steer axle path as detailed in the previous section. For each trailer configuration, the GPS and LiDAR data for the 5 trials were resampled spatially so that the articulation angle measurements of all trials could be averaged at each step along the steer axle path. Averaging the five trials helps reduce error from variations in driver behavior and sensor performance, however having only five trials is insufficient to generate a distribution, so the average articulation angle at each step is assumed as a point estimate. Then, at each step along the target steer axle path in sync with the spatially resampled data, the SGM-C algorithm provided an estimate of

the tractor's orientation and contribution of the converter dolly's articulation for the trailer-trailer articulation measurement on the A-doubles. From this model, the tractor was assumed to travel the same trajectory for every trial and trailer configuration of a given maneuver. This is a result of the model's no-tire slip assumption for non-steered axles that is acceptable at the tested speeds on a flat surface as demonstrated by the comparison to the TruckSim dynamic model in Section 3.2 and as found in literature [3,4,5,6,9,12,14,20,21,22]. Then, at each step, each trailing axle path is estimated for the given steer axle position, estimated tractor orientation, average measured tractor-trailer articulation angle, estimated dolly contribution, and measured trailer-trailer articulation angle. Equations (4-25) through (4-30) are the equations in this process. The articulation angles are represented as θ_1 for the tractor-trailer and θ_2 for the trailer-trailer. The dolly's contribution to θ_2 is represented as fraction κ . Each vehicle unit's orientation within the maneuver frame is φ_i , where φ_1 is for the steer axle, φ_2 is for the tractor, and so on. Each vehicle unit's wheelbase and hitch length is represented as L_i and H_i , respectively. The tractor's orientation φ_2 is estimated by SGM-C. The steer axle position $\mathbf{r}_{A_1} = \mathbf{r}_{P_1}$ is given by the target path.

$$\varphi_3 = \varphi_2 + \theta_1 \quad (4-25)$$

$$\varphi_4 = \varphi_3 + \kappa\theta_2 \quad (4-26)$$

$$\varphi_5 = \varphi_3 + \theta_2 \quad (4-27)$$

$$\boldsymbol{\phi}_i = [\cos(\varphi_i), \sin(\varphi_i)]^T \quad (4-28)$$

$$\mathbf{r}_{A_i} = \mathbf{r}_{P_{i-1}} - L_i \boldsymbol{\phi}_i \quad (4-29)$$

$$\mathbf{r}_{P_i} = \mathbf{r}_{P_{i-1}} - H_i \boldsymbol{\phi}_i \quad (4-30)$$

Figure 4-37 and Figure 4-38 plot the rearmost trailing axle path for each of the four maneuvers along with the line perpendicular to the steer axle path that coincides with location of peak off-tracking. For all maneuvers, at the location where peak off-tracking is measured for each trailer configuration, the order from smallest to largest off-tracking was: 28-ft double, 33-ft double, and 53-ft single. Odd kinks and oscillations in the rearmost axle paths relative to the steer axle path are well within the uncertainty bounds defined by the analysis described earlier in this chapter, and the uncertainty in the tractor's orientation is likely to be the primary factor in the deviations from the expected rearmost axle paths.

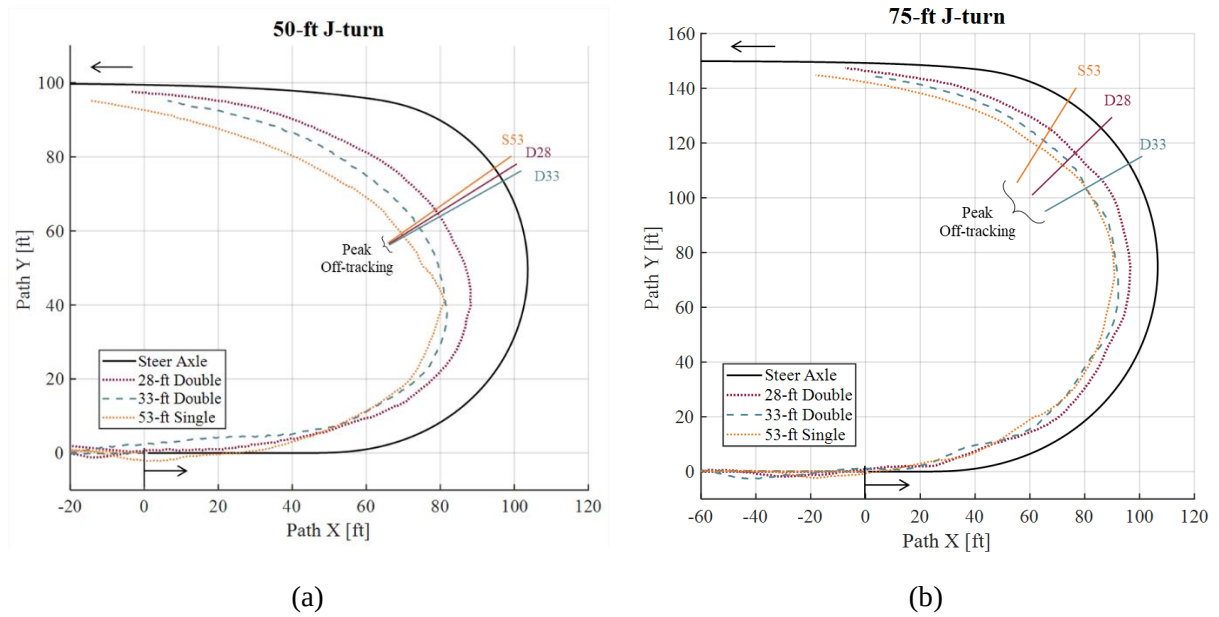


Figure 4-37. Steer axle path and measured rearmost axle paths for the (a) 50-ft J-turn and (b) 75-ft J-turn. Locations of peak off-tracking are marked by solid lines of appropriate color.

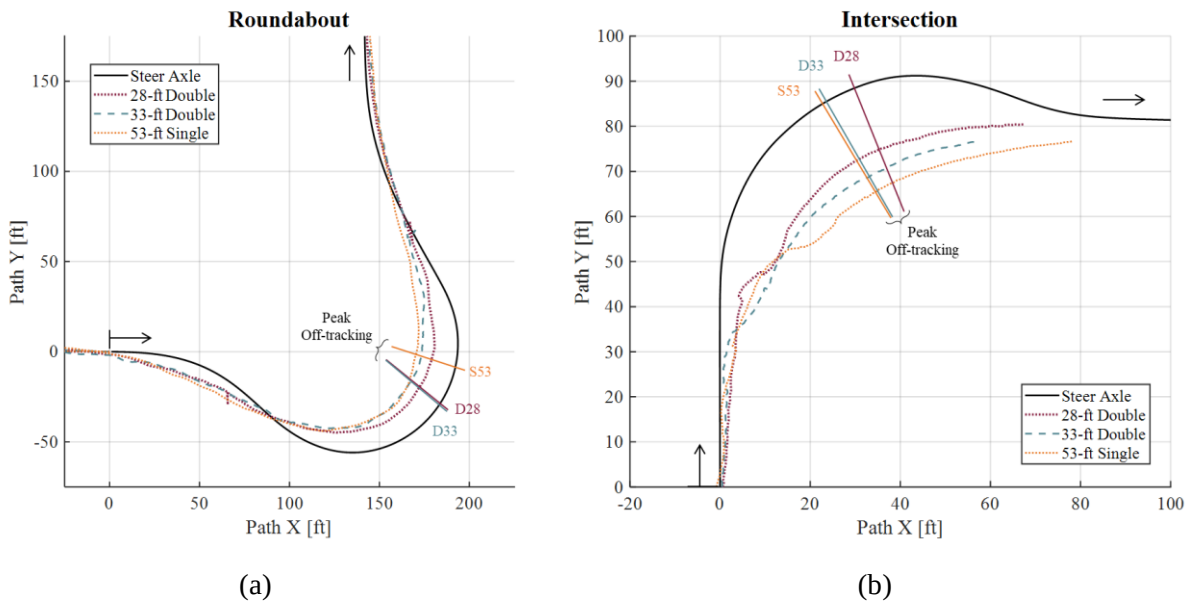


Figure 4-38. Steer axle path and measured rearmost axle paths for the (a) roundabout and (b) intersection maneuvers. Locations of peak off-tracking are marked by solid lines of appropriate color.

Uncertainty values for each of the spatial and temporal parameters at the moment of peak off-tracking are listed in Table 4-6. Each value is the RSS of its elemental uncertainties. As discussed in the previous

sections, some elemental uncertainties were taken as constants throughout a maneuver while others must be defined at the moment of peak off-tracking. The uncertainty values listed in Table 4-6 were then multiplied by the sensitivity indices shown previously in Table 4-1 to generate the peak off-tracking uncertainty for each parameter, maneuver, and trailer configuration in Table 4-7. The combined (RSS) uncertainties in peak off-tracking are also shown. Lastly, Table 4-8 lists and color-codes the average peak off-tracking sensitivity to each parameter, the average uncertainty of each parameter, and the average peak off-tracking uncertainty with respect to each parameter. This highlights the key parameters that strongly influence peak off-tracking uncertainty for all maneuvers and trailer configurations. Although the sensitivity indices and parameter uncertainties have different units, the table is color coded for each row to roughly illustrate the relative magnitude of each parameter. All values within the last row, peak off-tracking uncertainty, have units of feet and may be cross compared. Notice that the tractor's orientation has a large sensitivity index and uncertainty. These values ultimately dominate the uncertainty in the peak off-tracking result.

Table 4-6. Estimated uncertainty for each parameter.

Maneuver	Trailer Configuration	Spatial Parameters				Temporal Parameters		
		Steer Axle Position Perpendicular to Path	Tractor Orientation	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle	Steer Axle Position Along Path	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle
		[ft]	[deg]	[deg]	[deg]	[ft]	[ft]	[ft]
50-ft J-turn	28-ft Double	0.37	3.70	1.07	0.86	0.20	0.88	0.88
	33-ft Double	0.37	3.70	1.14	0.63	0.18	0.88	0.88
	53-ft Single	0.37	3.70	0.64	-	0.19	0.88	-
75-ft J-turn	28-ft Double	0.37	3.70	1.09	1.37	0.14	0.88	0.88
	33-ft Double	0.37	3.70	1.06	0.62	0.13	0.88	0.88
	53-ft Single	0.37	3.70	0.76	-	0.18	0.88	-
Roundabout	28-ft Double	0.37	3.70	1.00	1.18	0.16	0.88	0.88
	33-ft Double	0.37	3.70	0.86	0.64	0.20	0.88	0.88
	53-ft Single	0.37	3.70	0.65	-	0.19	0.88	-
Intersection	28-ft Double	0.37	3.70	1.48	1.96	0.24	0.88	0.88
	33-ft Double	0.37	3.70	1.29	0.67	0.24	0.88	0.88
	53-ft Single	0.37	3.70	1.56	-	0.21	0.88	-

Table 4-7. Peak off-tracking uncertainty due to uncertainty in each parameter and peak off-tracking's sensitivity.

Maneuver	Trailer Configuration	Spatial Parameters				Temporal Parameters			Result
		Steer Axle Position Perpendicular to Path	Tractor Orientation	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle	Steer Axle Position Along Path	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle	Peak Off-Tracking (RSS)
		[ft]	[ft]	[ft]	[ft]	[ft]	[ft]	[ft]	[ft]
50-ft J-turn	28-ft Double	0.17	3.96	0.91	0.36	0.17	0.30	0.20	4.10
	33-ft Double	0.23	4.37	1.12	0.31	0.16	0.33	0.26	4.55
	53-ft Single	0.23	3.81	0.50		0.15			3.85
75-ft J-turn	28-ft Double	0.07	4.37	0.98	0.57	0.11	0.23	0.09	4.52
	33-ft Double	0.09	4.85	1.11	0.32	0.11	0.30	0.12	4.99
	53-ft Single	0.10	4.00	0.59		0.13			4.04
Roundabout	28-ft Double	0.10	4.22	0.88	0.50	0.13	0.33	0.13	4.35
	33-ft Double	0.13	4.74	0.89	0.32	0.16	0.39	0.18	4.85
	53-ft Single	0.13	3.96	0.50		0.13			4.00
Intersection	28-ft Double	0.09	4.66	1.36	0.80	0.10	0.22	0.24	4.93
	33-ft Double	0.10	5.25	1.39	0.33	0.10	0.22	0.25	5.46
	53-ft Single	0.09	4.11	1.20		0.08			4.28

Table 4-8. Average sensitivity index, parameter uncertainty, and peak off-tracking uncertainty with respect to each parameter for the tested maneuvers and trailer configurations. Color coding is specific to each row.

	Steer Axle Position Perpendicular to Path	Tractor Orientation	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle	Steer Axle Position Along Path	Tractor-Trailer Articulation Angle	Trailer-Trailer Articulation Angle
Peak Off-Tracking Sensitivity Index	0.34	1.18	0.90	0.46	0.69	0.36	0.21
Parameter Uncertainty	0.37	3.70	1.05	0.99	0.19	0.88	0.88
Peak Off-Tracking Uncertainty	0.13	4.36	0.95	0.44	0.13	0.29	0.18

Figure 4-39 presents the measured peak off-tracking, error bars representative of the peak off-tracking uncertainty at the 95% level, and the tractrix model's estimated peak off-tracking for each trailer configuration and maneuver. There is good agreement between the field test data and model's estimate. For most trailer-maneuver combinations, the measured peak off-tracking was greater than the model's estimate. This is somewhat surprising because the tractrix model's no-tire slip assumption inherently tends to over-estimate off-tracking. However, the difference between the model's estimate and the measured peak off-tracking was consistently within the test procedure's uncertainty. For all maneuvers in both

measure and estimated peak off-tracking, the order from least to most off-tracking was the 28-ft double, 33-ft double, and 53-ft single. These results coincide with expectations driven by analyses demonstrated in Chapter 3.

Figure 4-40 plots the difference in peak off-tracking from the measured field tests to the tractrix model. The uncertainty values are not included for brevity and because the difference is within the uncertainty bounds for all trailer configurations and maneuvers as demonstrated in Figure 4-39. Peak off-tracking as measured from field test data match the tractrix model within 2 ft for all tested trailer configurations and maneuvers. The difference in peak off-tracking did not exhibit clear trends between trailer configurations and maneuvers and the variations may be explained within the uncertainty bounds from each parameter as detailed in previous sections.

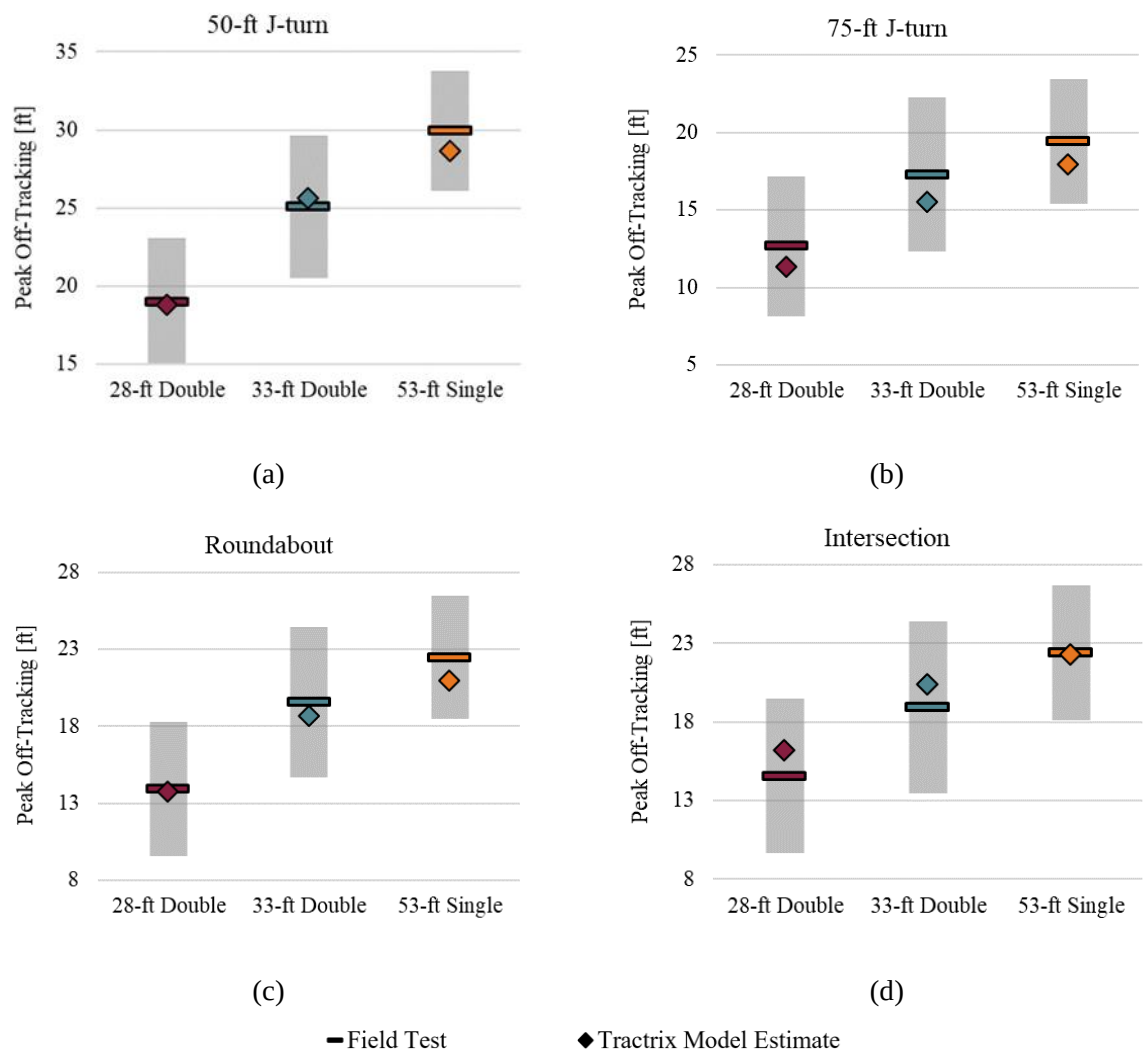


Figure 4-39. Peak off-tracking for each trailer configuration through the (a) 50-ft J-turn, (b) 75-ft J-turn, (c) Roundabout, and (d) Intersection maneuvers.

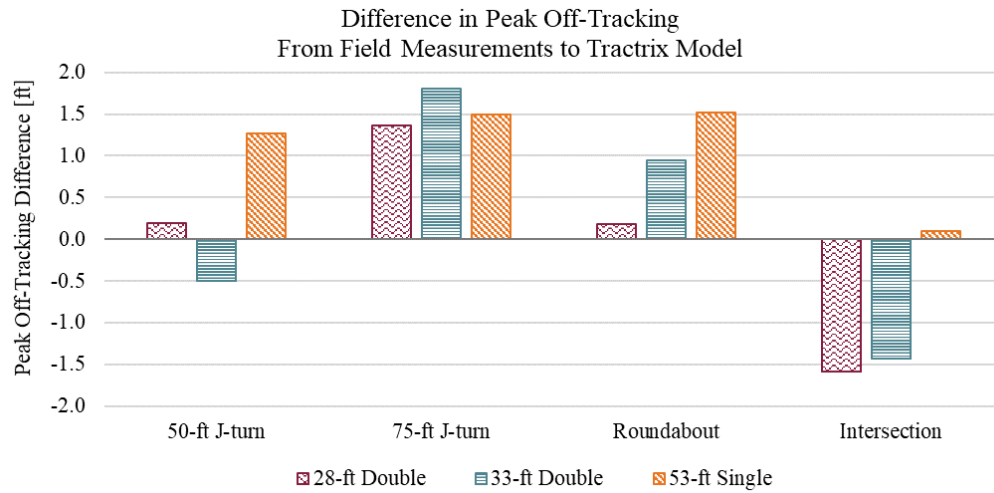


Figure 4-40. Difference in peak off-tracking from field test measurements to tractrix model.

In summary, the field test results coincide with the tractrix model's estimate. The peak off-tracking is highly sensitive to the tractor's orientation relative to the steer axle path and uncertainty in this value amplifies the uncertainty in the peak off-tracking results. On average for all maneuvers and tests, the difference in peak off-tracking was within 24% of the uncertainty in the test procedure and instrumentation. This value is relatively low but is unsurprising considering the rigor involved in estimating parameter uncertainties and that many parameter uncertainties took the value most conservative from available information.

Chapter 5

High-speed Vehicle Dynamics Test Vehicle & Equipment

This chapter details the 33-ft A-double tractor-trailer preparations for high-speed testing. Many mechanical and electrical systems were designed and installed by the project team. This chapter highlights all electrical work, the author's primary responsibilities. The chapter begins with a summary of the entire test vehicle and lists relevant parameters and dimensions. Next, selected sensors to measure dynamics relevant to trailer rollover propensity, stability control intervention, and brake type performance are detailed. Double straight-rail and drop-frame trailers in both heavy- and light-loading conditions were tested throughout the sponsored project; however, this dissertation only covers the studies specific to the heavily-loaded straight-rail trailers. The function and calibration of sensors are detailed, along with a brief overview of the data acquisition system (DAQ) and the instrumentation's measurement uncertainty. The CVeSS designed and built steering robot from previous studies is briefly reviewed and its upgrades are explained. The design, build, operation, and performance of the braking robot is detailed. Lastly, critical structural reinforcements and added mechanical equipment are presented.

5.1 Test Vehicle Parameters

The test vehicle for the high-speed roll-stability and trailer brake type performance studies was a class 8 tractor towing heavy-loaded 33-ft A-double straight-rail trailers. There was ample room inside the 2004 Volvo VNL sleeper-cab tractor to stage the data acquisition system, an electrical junction box, the steering and braking robot controls, and an extra passenger to monitor the live-stream of data from critical data channels. The tractor and both trailers were equipped with a suite of analog and digital sensors as detailed in Section 5.2. The tractor's steering and braking robotics are discussed in Sections 5.3 and 5.4. Each straight-rail trailer was retrofitted with extensive mechanical safety equipment including outriggers to prevent the trailer from rolling over completely and anti-jackknifing mechanisms to prevent excessive and unstable articulations that could harm the test equipment. Reinforcement structures distributed forces from the outrigger and anti-jackknifing systems during rollover or jackknifing events to prevent the trailers from deforming under excessive stress. Load frames supported a collection of sandbag weights to replicate real-world trailer payload and CG position. These mechanical systems are discussed further in Section 5.5. Figure 5-1 presents the test vehicle with outriggers deployed during preliminary shakedown testing at Virginia Tech.



Figure 5-1. Test vehicle.

Table 5-1 below lists relevant properties of each vehicle unit. Mass and inertia values were estimated by validated CAD models of the empty trailers. The trailers' tare eights were measured by a truck scale, and their CG forward and vertical positions were estimated using the truck scale and a tilt-test. In a tilt-test, a trailer axle rests on a scale while the front end is lifted upward. From the load transfer onto the grounded axle, trailer CG longitudinal position, and angle that the trailer is lifted, the CG vertical position was estimated. After validating the empty trailer CAD models, mechanical equipment (outriggers, load frames, reinforcement, sandbags, etc.) was added to the CAD model and the heavily-load trailer's mass and inertia properties were estimated. The targeted trailer payload was determined from data provided by the test program's sponsor. The loaded trailer target CG height was based on a ~15,000 lb payload with an assumed 80% floor-to-ceiling volume fill with uniform density. These tasks were led and conducted by team members other than the author, but the results are included for their relevance to the presented studies. The tractor and dolly wheelbase values are the same as in Table 3-1 because they are the same equipment. The trailer wheelbase is slightly different because the low-speed study used 33-ft drop-frame trailers and the high-speed studies used 33-ft straight-rail trailers. The track width is the lateral distance between the center of each group of tires on the left and right side.

Table 5-1. Test vehicle specifications.

Parameter	Tractor	Trailer	Dolly
Total Weight [lb]	20,820	25,022	3,000
CG height to ground [in]	39.4	77.9	27.2
CG longitudinal distance from kingpin [in]	-	179.6	-
Roll moment of inertia [lb-ft ²]	175,000	403,219	2,849
Yaw moment of inertia [lb-ft ²]	499,000	2,913,493	3,580
Wheelbase [in]	250	330	72
Track Width [in]	82 (steer) 73 (drive)	78	78

5.2 Instrumentation and Data Acquisition

A multitude of sensors were installed to measure key dynamics of the tractor and trailers throughout the roll-stability and brake-type studies. These include pressure transducers, accelerometers, string-potentiometers, load cells, GPS, and multi-segment solid-state LiDAR. Relays were also installed to flag vehicle stability control activations including anti-lock brakes (ABS), tractor electronic stability control (ESC), and trailer roll stability control (RSC). Custom built-in-house junction boxes routed power to all sensors and their signals to the data acquisition unit. LabVIEW™ software controlled the National Instruments CompactRIO™ (NI cRIO) data acquisition unit that provided live readouts of recorded analog sensor voltages and stream of GPS data. The LiDAR sensors required a separate software application to configure and record data and were later synchronized with the cRIO data in post-processing. Additionally, GoPro cameras were installed throughout the tractor and trailers to monitor wheel-lift, outrigger contact, steering, and overall vehicle dynamics. A relay illuminated LEDs within each GoPro's field-of-view during each test trial so that video data could be synchronized with analog and GPS data in post-processing. The following sections detail the selected sensors' measurements and calibration process, junction boxes, power supply, NI cRIO, and GoPros. Measurement uncertainties from sensor calibration, installation, and electrical noise is also discussed.

5.2.1 Sensor Selection and Calibration

Figure 5-2 illustrates the general layout of sensors used in the roll-stability and brake type performance testing on the double-trailer truck. Each trailer had 9 or 10 analog sensors and 3 relays. The trailer relays communicated the activation of RSC step 1, RSC step 2, and ABS by closing a 5 V circuit onto the designated DAQ channels that were normally grounded when the control systems are not intervening. The tractor relays and flags operated similarly but instead were controlled by activation of tractor ESC, tractor ABS, maneuver duration, steering robot operation, or braking robot activation. One LiDAR sensor in front of the tractor was used to measure the braking distance for straight-line stopping tests, and the other LiDAR sensors were used to measure articulation angles between vehicle units. The photoelectric sensor was not part of the DAQ system, but instead was part of the steering and braking robotics. This sensor ensured consistency among test trials by providing spatial triggering to the steering/braking robot and, subsequently, the maneuver flag.

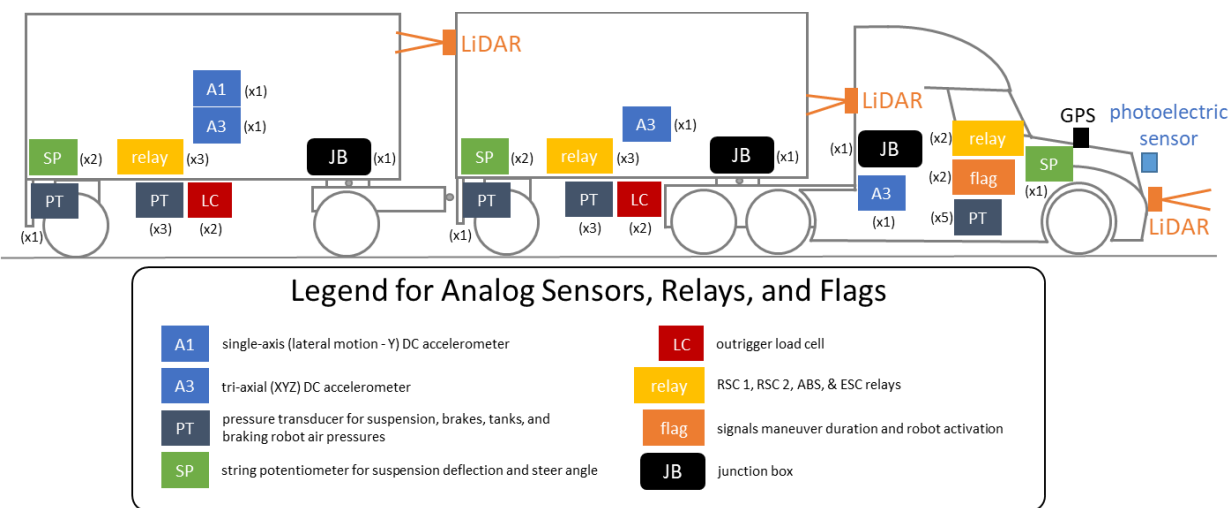


Figure 5-2. Illustration of sensor type, count, and locations on the double-trailer truck. The DAQ is not shown in this illustration, but it was placed in the back of the tractor cab.

Having the photoelectric sensor as a spatial trigger ensured repeatable and reliable positioning of each maneuver start which proved critical to the success of the brake type study. As soon as the sensor passed the reflective target, the maneuver flag would not only close a 5 V circuit onto a normally grounded channel on the DAQ but would also simultaneously close the switched 13 V circuit to all LEDs around the tractor and trailers so that videos could be synchronized with the sensor data. Figure 5-3 illustrates the basic operation of the photoelectric sensor and Table 5-2 provides a comprehensive list of sensors used in the roll-stability and brake type performance studies.

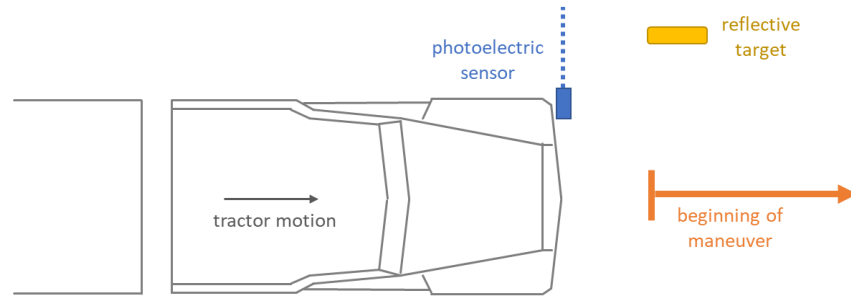


Figure 5-3. Illustration of the tractor with photoelectric sensor and reflective target to flag the beginning of a maneuver for data processing and synchronization.

Table 5-2. A comprehensive list of installed sensors for the roll-stability and brake-type studies.

	Sensor Type	Brand and Model	Measurement Range	Supply Voltage	Application	Output Type
1	GPS	Garmin 18x-5Hz	N/A	5 V	tractor position, speed, and orientation	digital
2	LiDAR	LeddarTech IS16	0-50 m (16 segments)	13 V	articulation angle between tractor & front trailer	digital
3	LiDAR	LeddarTech IS16	0-50 m (16 segments)	13 V	articulation angle between front & rear trailers	digital
4	LiDAR	LeddarTech IS16	0-50 m (16 segments)	13 V	forward distance from braking end target	digital
5	photoelectric	Schneider XUX1ANANM12	0-14 m	5 V	spatial trigger	digital
6	string potentiometer	UniMeasure HX-P510-15-NJC-N6-L3M	0-40 in	13 V	steer angle	analog single-ended
7	tri-axial accelerometer	PCB 3713F1110G	+/- 10g (XYZ directions)	13 V	tractor CG XYZ accelerations	analog single-ended
8	tri-axial accelerometer	PCB 3713F1110G	+/- 10g (XYZ directions)	13 V	front trailer CG XYZ accelerations	analog single-ended
9	tri-axial accelerometer	PCB 3713F1110G	+/- 10g (XYZ directions)	13 V	rear trailer CG XYZ accelerations	analog single-ended
10	single-axis accelerometer	PCB 3741F1210G	+/- 10g	13 V	rear trailer CG lateral (Y) accelerations	analog differential
11	load cell	Forsentek FLP-100t	0-200,000 lbf	10 V	front trailer driver side outrigger contact force	analog differential
12	load cell	Forsentek FLP-100t	0-200,000 lbf	10 V	front trailer passenger side outrigger contact force	analog differential
13	load cell	Forsentek FLP-100t	0-200,000 lbf	10 V	rear trailer driver side outrigger contact force	analog differential
14	load cell	Forsentek FLP-100t	0-200,000 lbf	10 V	rear trailer passenger side outrigger contact force	analog differential
15	pressure transducer	OMEGA PX329-200G5V	0-250 psi	13 V	front trailer suspension pressure	analog single-ended
16	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	front trailer driver side brake pressure	analog single-ended

17	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	front trailer passenger side brake pressure	analog single-ended
18	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	front trailer air tank pressure	analog single-ended
19	pressure transducer	OMEGA PX329-200G5V	0-250 psi	13 V	rear trailer suspension pressure	analog single-ended
20	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	rear trailer driver side brake pressure	analog single-ended
21	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	rear trailer passenger side brake pressure	analog single-ended
22	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	rear trailer air tank pressure	analog single-ended
23	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	braking robot actuator pressure	analog single-ended
24	pressure transducer	Honeywell PX2EN1XX250PSAAX	0-250 psi	5 V	braking robot auxiliary air tank pressure	analog single-ended
25	string potentiometer	UniMeasure JX-P510-10-N13-10S-N1C	0-10 in	13 V	front straight-rail trailer driver side suspension travel	analog single-ended
		UniMeasure VP510-10-N-11-D0S-10C			front drop-frame trailer driver side suspension travel	analog single-ended
26	string potentiometer	UniMeasure JX-P510-10-N13-10S-N1C	0-10 in	13 V	front straight-rail trailer passenger side suspension travel	analog single-ended
		UniMeasure VP510-10-N-11-D0S-10C			front drop-frame trailer passenger side suspension travel	analog single-ended
27	string potentiometer	UniMeasure JX-P510-10-N13-10S-N1C	0-10 in	13 V	rear straight-rail trailer driver side suspension travel	analog single-ended
		UniMeasure VP510-10-N-11-D0S-10C			rear drop-frame trailer driver side suspension travel	analog single-ended
28	string potentiometer	UniMeasure JX-P510-10-N13-10S-N1C	0-10 in	13 V	rear straight-rail trailer passenger side suspension travel	analog single-ended
		UniMeasure VP510-10-N-11-D0S-10C			rear drop-frame trailer passenger side suspension travel	analog single-ended

Although each sensor was provided with performance tolerance and specifications, many of them required us to create calibration methods to more accurately quantify each individual sensor's performance and convert sensor output voltages to engineering units representative of the measured dynamics. The calibration procedure for each sensor type is detailed in the following subsections. The calibration curves including unit conversion factors for each sensor are detailed in Appendix D. Calibration and sensor installation effects on measurement uncertainty are discussed.

(a) String Potentiometers

All string potentiometers used for measuring the suspension travel accept a 13 V power supply and output 0-10 V, where 0 V is fully retracted (0 in) and 10 V is fully extended (~10 in). This output voltage is

independent of the supply voltage if the supply is within the accepted range (4.9-25 V). Due to packaging constraints and availability of sensors already belonging to CVeSS, the drop-frame and straight-rail suspension string potentiometers were different models; however, both provide the same outputs (0-10 V) and have the same range (0-10 in). To verify that each sensor matched the provided datasheets and to derive a calibration curve for data processing and analysis, each sensor was connected to a small jig with hooks at 2.5, 5, 7.5, and 10 inches from the string's end when fully retracted. Once the sensor was secured to the jig, it was connected to a junction box to receive power and provide terminals to measure the sensor's output voltage with a voltmeter. The output voltages were recorded for each of the tested positions in increasing then decreasing order for a total of 9 data points as follows: 0, 2.5, 5, 7.5, 10, 7.5, 5, 2.5, and 0 inches. A linear curve-fit from these 9 data points provided the necessary linear calibration curve, of which all suspension string potentiometers resulted in R-squared values of 0.9999 or higher. Figure 5-4 pictures each type of installed string potentiometer.

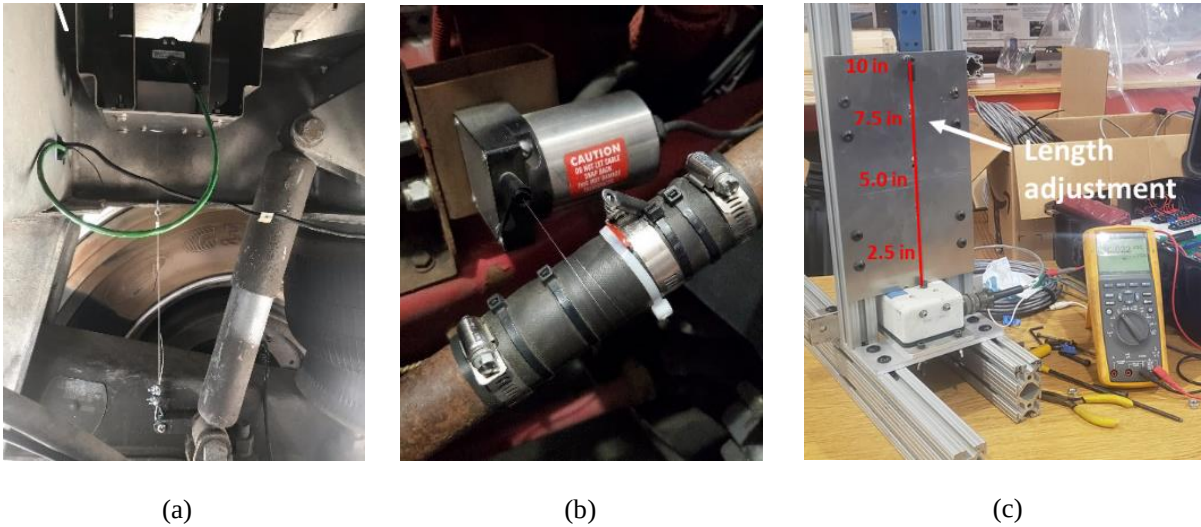


Figure 5-4. Pictures of installed string potentiometers: (a) straight-rail suspension, (b) steering wheel angle, (c) suspension string-potentiometer calibration jig.

The uncertainty of a curve-fit is determined by Equation (4-18) that is duplicated here as Equation (5-1). The maximum curve-fit uncertainty of all tested string potentiometers was approximately 0.023 in – about 3/128 of an inch.

$$u_{curve-fit} = t_{v,p} \left[\frac{\sum_{i=1}^N (y_i - y_{c_i})^2}{v} \right]^{\frac{1}{2}} \quad (5-1)$$

Because the string potentiometer measurements zero-offset was determined after mounting, the calibration uncertainty is determined solely by the curve-fit's slope. Thus, the actual string potentiometer calibration uncertainty is determined by Equation (5-2) that is simply the combination of Equations (4-12) and (4-13) shown earlier for the LiDAR measurements. Notice that, for the same data set, Equation (5-2) is always smaller than Equation (5-1). This makes sense because Equation (5-1) addresses uncertainty in both the zero offset and slope whereas Equation (5-2) only for the slope. For the suspension string potentiometers, the maximum calibration slope uncertainty was found as less than 0.003 in. Considering most suspension travel of interest ranges to +/-1 in at the sensor, the calibration uncertainty is negligible.

$$u_{sp,slope} = t_{v,p} \sqrt{\frac{\sum_{i=1}^N (y_i - y_{ci})^2}{v(\sum_{i=1}^N (x_i - \bar{x})^2)}} \quad (5-2)$$

During suspension jounce and rebound, the trailing arm rotates about a single pivot point and the potentiometer's string connection path traces an arch. Because the string length was large relative to the suspension jounce/rebound, and because the string was approximately tangential to the trailing arm's radius, small angle approximation could be fairly applied to assume that the string potentiometer's measurements equaled the vertical motion of the trailing arm at the string's connection point.

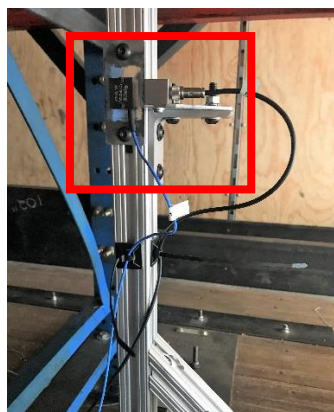
A different method was required to derive the calibration curve of the steering wheel angle string potentiometer due to its installation. Using a junction box to provide power and open terminals for measuring the sensor's output, the steering wheel was rotated to the angles listed in Table 5-3 along with the recorded sensor output voltage. This was conducted before testing and resulted in the linear curve fit equation, $y = 285.6x - 1333$, where y is the steering wheel angle in degrees and x is the measured voltage output. The R-squared value was 1.0000. This was periodically verified throughout the testing program. Using Equation (5-1), the steering wheel angle measurement uncertainty is approximately +/-6.8°. This value is very conservative due to the limited number of data points in Table 5-3.

Table 5-3. Steer angle string potentiometer calibration test results. A negative steering wheel angle turns the vehicle right and a positive steering wheel angle turns the vehicle left.

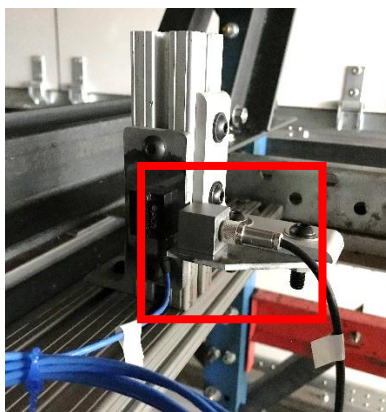
Output voltage (V)	4.662	3.403	2.167	3.401	4.661	5.931	7.200	5.931	4.661
Steering wheel angle (deg)	0	-360	-720	-360	0	360	720	360	0

(b) Accelerometers

A Piezotronics® 3713F1110G 10g triaxial accelerometer rated for frequencies 0-1000 Hz was mounted at each trailer's heavy-loaded center-of-gravity (CG) and on the tractor's frame nearest its center-of-gravity. The rear trailer included an additional PCB Piezotronics® 3741F1210G 10g single-axis accelerometer installed beside the triaxial accelerometer to provide redundancy in measuring the lateral acceleration of the rear trailer. This redundancy safeguards the rear trailer's lateral acceleration measurements which is an important indicator used to analyze the roll stability of the double-trailer truck during dynamic steering maneuvers. Figure 5-5 presents the installed accelerometers on the tractor and trailers. A stiff aluminum structure fixed the straight-rail accelerometers at the CG of the heavily loaded trailer. The drop-frame accelerometers were mounted onto the roller table which was rigidly fixed to the trailer and load frame. The tractor triaxial accelerometer required an aluminum structure to raise and hold its accelerometer nearest the tractor's CG. Because the tractor accelerometer was mounted outside, it was fixed inside a water-proof case.



(a)



(b)



(c)

Figure 5-5. (a) Straight-rail triaxial and single-axis accelerometers on extruded aluminum structure mount, (b) drop-frame triaxial and single-axis accelerometers, (c) tractor triaxial accelerometer mounted inside waterproof case on aluminum extension from tractor frame.

All accelerometers used within the high-speed study were brand new from PCB Piezotronics®. Both the single axis and tri-axial models were rated for 0-1000 Hz, allowing them to measure gravitational acceleration and very low frequency dynamics common to large, heavy vehicles. These sensors were provided with calibration characteristics directly from the manufacturer using special equipment for accurate and precise calibration procedures. Because of this, it was not needed to generate our own calibration process and the calibration uncertainty may be assumed to be negligible. Each sensor's 0 Hz

(DC) acceleration output was verified by resting the sensor on a flat surface with the specified measurement axis experience $\pm 1g$ and comparing the sensor's output with the provided calibration. All accelerometer signals matched their provided calibration curves.

In addition to the sensor's calibration, uncertainty of the CG's acceleration can arise from misplacement of the accelerometer. The loaded trailer's center of gravity was determined by empirically-validated CAD models. Positioning the accelerometer at the estimated CG position accurately in vertical, longitudinal, and lateral directions was easy to accomplish. Assuming the accelerometers fit squarely within each vehicle unit (i.e. $+x$ is parallel to the longitudinal axis), difference between the accelerometer's measurement and the actual CG acceleration is approximated by Equation (5-3) where α is rotational acceleration and r_p is the radius in plane P from accelerometer to the CG's axis of rotation. Each trailer's large mass and size limit its angular accelerations α within the performed tests. Furthermore, radius r_p is assumed to be very small (less than a few inches) for all accelerometers. From this, it is acceptable to assume that uncertainty caused by installation error is negligible.

$$a_{err} = \alpha r_p \quad (5-3)$$

(c) Pressure Transducers

For calibrating the pressure transducers, each transducer was exposed to a slowly changing pressure controlled by a pressure regulator, as shown in Figure 5-6. A reference pressure transducer with a known calibration was exposed to the same calibration test pressures at the same time as each of the pressure transducers being tested. The reference transducer was calibrated from static tests at 0, 20, 40, 60, 80, and 100 psi that were manually set by a pressure regulator and monitored by a digital pressure gauge with ± 0.1 psi accuracy. Uncertainty in the reference pressure transducer's calibration is dominated by the digital pressure gauge's accuracy and thus will take the value of ± 0.1 psi assuming 95% confidence level.

The reference pressure transducer provides a common baseline across all tested pressure transducers. The configuration of transducers on the test jig allows us to assume that the reference and test specimen experience approximately the same air pressure and that dynamic pressure effects are approximately equal for each transducer, as illustrated in Figure 5-6. During the calibration process, high pressure provided by the lab's utility air (red) entered a smaller auxiliary tank through an on/off valve. A pressure regulator was manually adjusted to a maximum pressure of 100 psi. Both the test and reference pressure transducers were exposed to this pressure-controlled air (blue) that was sealed from the environment with an on/off valve. The pressure was slowly ramped up from 0 psi to 100 psi then slowly lowered through the on/off

valve into the atmosphere as shown in Figure 5-7. A digital gauge located at the same point as the pressure transducers helped the test operator control the rate of increasing and decreasing pressure. Signals from the reference and test pressure transducers were collected simultaneously. The test transducer's calibration was then determined as a linear curve-fit from the reference transducer. Due to the large number of data points, curve-fit uncertainty is negligible compared to the reference transducer's calibration uncertainty. Thus, it is assumed that all tested pressure transducers have a calibration uncertainty of ± 0.1 psi.

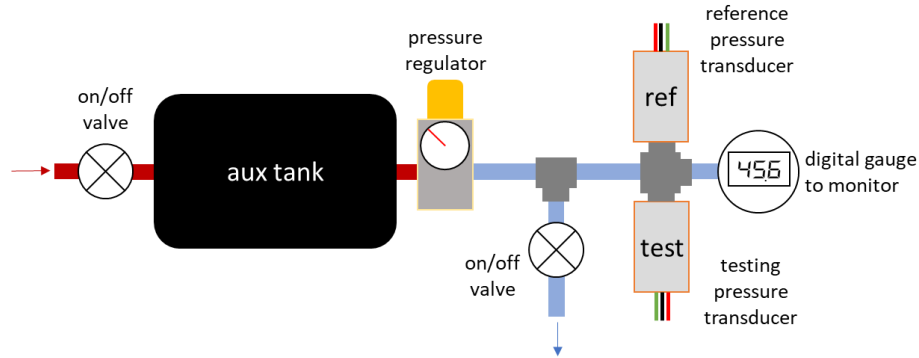


Figure 5-6. Schematic for calibrating the pressure transducers.

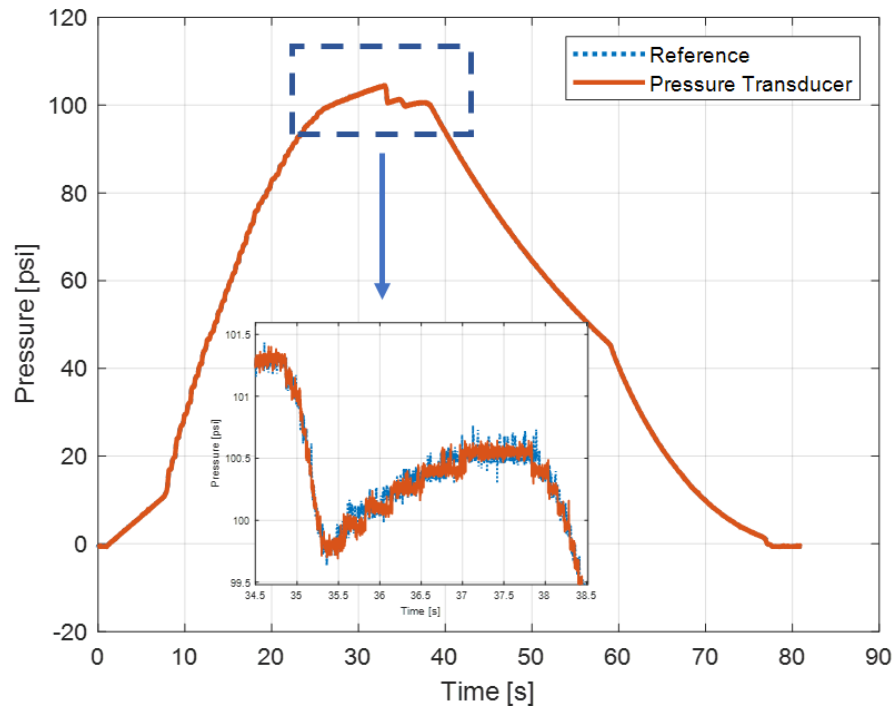


Figure 5-7. Example of pressure transducer calibration.

(d) Load Cells

Each trailer has a pair of outriggers that prevent that trailer from rolling over completely while also providing a means to estimate the impact forces between the ground and outrigger. This contact force was measured by a 200,000 lbf rated load cell mounted inboard of the outrigger supporting the outrigger arm. These load cells were brand new and provided with calibration characteristics determined by the manufacturer. Using the provided calibrations, tests were conducted to determine the ratio from the outrigger wheel contact force to the force measured by the inboard load cell.

A 4,000 lbf hoist with a Futek™ L2900 tensile load cell rated up to 20,000 lbf was positioned over each outrigger wheel bearing surface. A heavy-duty nylon ratchet strap was wrapped around the wheel bearing surface and a hook mounted on the lifting Futek load-cell, as shown in Figure 5-8. Both the Futek and the tested outrigger load cells were connected to a junction box for power and routed their outputs to the same NI cRIO DAQ used throughout the testing program. Once slack was removed, data was recorded continuously as the hoist lifted the outrigger arm with up to 3,500 lbf then back down to just before slack was removed. This process was repeated two additional times for each outrigger. Data from all three trials was used to create curve-fits of the outrigger wheel contact force to the measured reaction force of the inboard outrigger load cell.



Figure 5-8. Test setup for determining the force ratio from the outrigger wheel to the outrigger load cell.

It was found that the vertical force applied on the outrigger wheel bearing (outrigger contact force) was best approximated by a piecewise function of outrigger load cell's measurement. For forces up to about 14,000 lbf measured by the inboard load cell, 3rd-order polynomial curves fitted the outrigger wheel bearing load for the straight-rail trailers. For higher forces, the force ratio was approximately constant, making the outrigger wheel force a linear function of the load cell's measured force. The force ratio of the load cell to the outrigger wheel ranged from approximately 8:1 to 14:1. Figure 5-9 shows examples of the piecewise curve-fits for a straight-rail outrigger. Unique curve-fits were determined for each outrigger arm and load cell. The outrigger arm angle and hinge assembly's kinematics caused the significant variation in load cell to outrigger wheel force ratio.

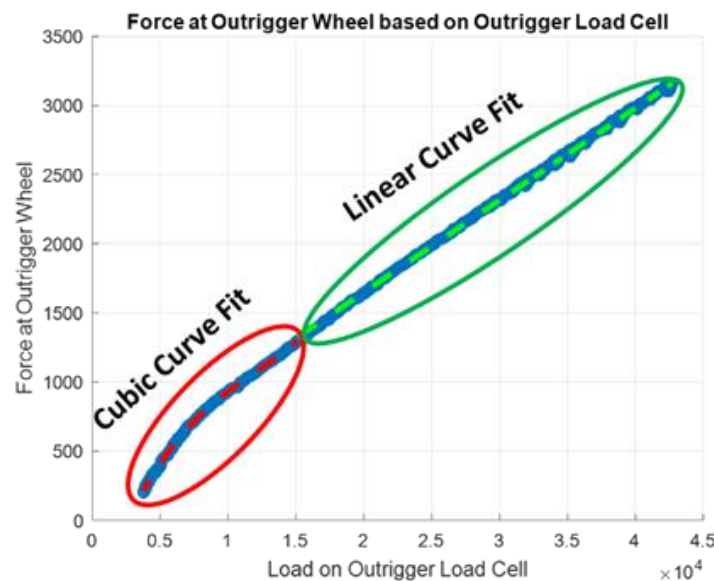


Figure 5-9. Example plots of the measured force at the outrigger wheel versus the force measured by the load-cell inside the outrigger for a straight-rail trailer outrigger arm.

To estimate outrigger wheel force measurement uncertainty, a thorough analysis is needed on the force ratio curve-fits. This process would be similar to those for the string potentiometers and LiDAR sensors. However, as will be discussed in more detail in Chapter 6, significant electrical noise prompted aggressive post-processing digital filtering that drastically impacted measurement accuracy. Rather than using contact forces as point estimates for rollover severity or likelihood, trends in filtered outrigger wheel contact force with respect to entry speed are used. Chapter 6 details this process. For conditions with limited data – one or no contact forces at the estimated rollover speed – videos from the individual test trials are consulted. Because this process is largely subjective, quantifying uncertainty of the outrigger wheel contact force estimate is moot and thus not included within this study.

5.2.2 Post-processing Digital Filtering

Zero-phase digital low-pass filters were applied in post-processing to many data channels to remove high-frequency electrical noise from the measured low-frequency mechanical dynamics. The filters were 5th-order Butterworth with cutoff-frequencies specific to each measurement type. Due to the outrigger load cell's extremely small voltage outputs (typically less than 1 mV) and long cable distance from the load cell, significant electrical noise in the signal made it challenging to decipher the exact measurements. An aggressive 2 Hz cutoff frequency was used. Although the outrigger wheel impact dynamics during a rollover event likely have some frequency content above 2 Hz, the cutoff frequency was chosen from trial and error such that the moment of an impact was easily distinguished from noise. Superposition of electrical noise and the digital filter reduce the peak outrigger contact force accuracy. However, as will be discussed later in Chapter 6, the peak outrigger contact force for each trial is only used as a relative magnitude among trials rather than an exact value to back-calculate roll inertial. Thus, the thresholds for categorizing “light” and “hard” outrigger contacts were subjectively categorized after filtering and extensive review of test video recordings.

The triaxial accelerometers' signals were also very small ($\sim 70\text{mV}$ in y-direction just before rollover) and required aggressive filtering. The filter cutoff frequency was determined by comparison to the single-axis accelerometer measurements. The single-axis accelerometer's signal was much cleaner due to having twice the sensitivity (270mV/g vs. 135mV/g) and a differential output rather than single-ended. From a wide range of test trials, Table 5-4 lists the average difference and absolute average difference of the triaxial accelerometer's lateral acceleration just before rollover from the single-axis accelerometer's measurement for a variety of cutoff frequencies. Notice that for a 2.5 Hz cutoff on the triaxial, the average and average absolute differences between sensors had the least magnitude for nearly all cutoff frequencies on the single-axis accelerometer. Because of this, 2.5 Hz cutoff frequency was used on all acceleration measurements.

Table 5-4. Lateral acceleration just before rollover as determined by triaxial and single-axis accelerometers for a range of low-pass filter cutoff frequencies.

Average Absolute Difference [g]										
fc [Hz]	triaxial:	1	1.5	2	2.5	3	3.5	4	5	10
single-axis	same	0.094	0.076	0.074	0.058	0.085	0.073	0.067	0.089	0.091
	2.5	0.101	0.068	0.071	0.058	0.086	0.075	0.065	0.088	0.092
	3	0.097	0.072	0.076	0.055	0.085	0.073	0.065	0.090	0.094
	5	0.099	0.072	0.074	0.056	0.085	0.073	0.065	0.089	0.091
	10	0.097	0.072	0.076	0.057	0.086	0.074	0.066	0.090	0.091

Average Difference [g]										
fc [Hz]	triaxial:	1	1.5	2	2.5	3	3.5	4	5	10
single-axis	same	-0.073	0.014	0.041	-0.011	-0.044	-0.044	-0.033	-0.050	-0.047
	2.5	-0.076	0.006	0.038	-0.011	-0.048	-0.048	-0.036	-0.053	-0.053
	3	-0.073	0.010	0.042	-0.008	-0.044	-0.045	-0.032	-0.049	-0.050
	5	-0.073	0.009	0.041	-0.008	-0.045	-0.045	-0.033	-0.050	-0.051
	10	-0.070	0.012	0.044	-0.005	-0.042	-0.042	-0.030	-0.047	-0.047

Signals from the string potentiometers and pressure transducers were well above the noise floor and easily captured key measurement dynamics without the need for as aggressive filtering. Furthermore, the suspension and pressure dynamics contain higher frequency content than the body accelerations, so the cutoff frequency needed to be higher than that of the accelerometers. The LiDAR sensor's articulation angle measurements could be filtered in two ways: (1) filter individual segments before estimating the instantaneous articulation angle and (2) filter the estimated articulation angle. Because the instantaneous articulation angle was determined by a curve-fit of the segments' measurements, individual segment measurements were not filtered as the curve-fitting process acts like a filter. Instead, only the estimated angle was filtered. Table 5-5 lists the digital low-pass filter cutoff frequency for each measurement.

Table 5-5. Low-pass filter cutoff frequency by measurement.

Measurement	Zero-phase Digital Low-Pass Filter Cutoff Frequency
Outrigger contact force	2 Hz
Lateral acceleration	2.5 Hz
Air pressure (Suspension, air tanks, brake chambers, braking robot)	25 Hz
Suspension travel	15 Hz
Articulation angle	5 Hz

5.2.3 Junction Boxes

The test tractor and each trailer had a junction box (JB) to power all sensors and route analog sensor and relay signals to the DAQ. Each of the four trailer junction boxes were identical to the next to provide flexibility for testing so that any trailer could be swapped into position as the leading or trailing unit. The tractor's JB differed from the trailers' JBs because it connected fewer analog sensors and combined the signal cabling from the front trailer with the tractor. This juncture was necessary to route signals of the three vehicle units (tractor, front trailer, & rear trailer) to the two available DB-37 analog-to-digital converter modules on the NI cRIO DAQ. All analog sensor outputs were voltages and the DAQ's configuration is discussed in a later section.

The five junction boxes (4 trailers + 1 tractor) contained 4 power circuits: continuous 13 V, 10 V, and 5 V circuits for sensors, relays, LiDAR, and the DAQ, and a switched 13 V circuit for the LEDs used to synchronize video with the sensor data. Power was provided by 13 V Lithium-Iron Phosphate (LiFePO₄) battery packs. A common ground was connected between all junction boxes to ensure that each vehicle unit had the same base electric potential without introducing ground loops. The trailer junction box battery packs were sized at 10 Ah (130 Wh) each and the tractor junction box included a 12 Ah (156 Wh) battery pack for the sensors (analog, GPS, and up to 2 LiDAR) and LEDs, and a 10 Ah (130 Wh) battery pack for the DAQ. All battery packs were configured in a 4s1p layout with Headway™ cylindrical cells and were connected to junction boxes through a fuse. The collection of installed battery packs provided 12+ hours of continuous power for the DAQ and all sensors used in testing. This was more than sufficient for a full day of active testing on the track.

The battery packs' voltages were stepped down to 5 V and 10 V circuits through Texas Instruments uA78xx series fixed-voltage integrated circuit voltage regulators. Only the two outrigger load cells required the 10 V circuits. The 5 V circuits were used for the Honeywell pressure transducers, relays, and GPS. In the trailers, only one 5 V regulator was needed to support the trailer sensors' light current draw. The tractor JB required two 5 V regulators: one for relays and pressure transducers and one dedicated to the GPS receiver. Because the load cells and Honeywell pressure transducers are ratiometric sensors, the voltage regulator outputs were also connected to dedicated channels within the DAQ. Figure 5-10 illustrates the power scheme throughout the system.

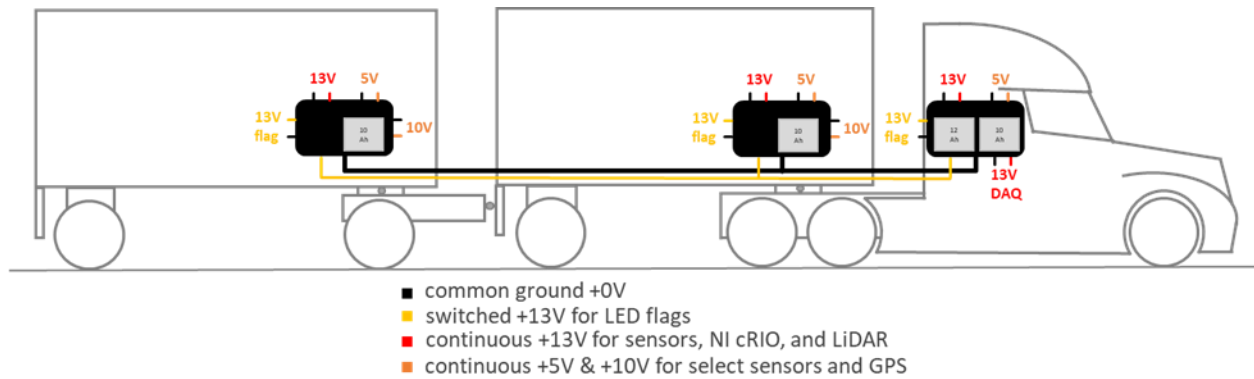


Figure 5-10. Schematic representation of the power provided by the batteries and distributed by the junction boxes.

All JBs were built upon Apache™ protective cases with an added 0.25-in thick high-density polyethylene (HDPE) base for mounting breakout boards and terminal blocks. The sensor connectors were mounted onto one sidewall of the case while the LED flag circuit connectors were mounted on the other side. A large cutout was made in one sidewall for the DB-37 cable that carried the sensor signals to the DAQ, and holes were drilled through the HDPE base for a zip-tie to serve as a strain relief for the DB-37 cable. Figure 5-11 shows the sensor terminals on trailer and tractor JBs. All sensor terminals were the same 5-pin Switchcraft™ connectors to allow modularity, however, each was specifically chosen to receive a 5 V, 10 V, or 13 V supply. The power scheme to each connector was clearly diagrammed inside the junction box. Dean's connectors provided the 5 V and 13 V supplies to GPS, LiDAR, and LEDs.



Figure 5-11. Junction box external connections. (a) Tractor sensor connectors and GPS power. (b) Tractor DB-37 output and DB-37 input from front trailer, LiDAR power port and connectors for flag LEDs on the tractor. (c) Trailer analog sensor connectors and power ports. (d) Trailer flag LED connectors and cutout for DB-37.

A breakout board was placed in the center of each junction box to route all analog sensor and voltage regulator outputs (green, orange, blue, and purple signal wires) to a single DB-37 cable. As the name implies, the DB-37 cable has a total of 37 conductors. Each conductor was well insulated, and the entire DB-37 cable assembly was enclosed in a foil shielding to reduce electrical noise and interference. All wires in the junction boxes were color coordinated as described in Table 5-6. The pin assignments are detailed in Appendix E. Pull-down resistors on the breakout board grounded relay channels when relays were not active.

Table 5-6. Junction box wire color coding.

Wire color	Corresponding use
Red	Power (+5 V, +10 V, or +13 V)
Black	Common ground
Green	Sensor pin 1
Orange	Sensor pin 2
Blue	Sensor pin 3
Purple	Voltage regulator output (+5 V or +10 V)

The junction box on the tractor includes a dual DB-37 breakout board for connecting the DB-37 cable from the first trailer, as shown in Figure 5-12. These signals were combined with the signals from the tractor sensors and sent to the DAQ through the other DB-37 cable. Great efforts were made to ensure that packaging within each JB cleanly matched with the available channel inputs on the breakout board and DAQ. In Figure 5-12, the 12-Ah battery was at a slight angle to fit within the JB's outer casing when the lid was closed. This obstructs the view of the switched 13V terminal blocks for the LEDs around the tractor. The 10-Ah battery is not shown as it was mounted on the inside of the JB lid.

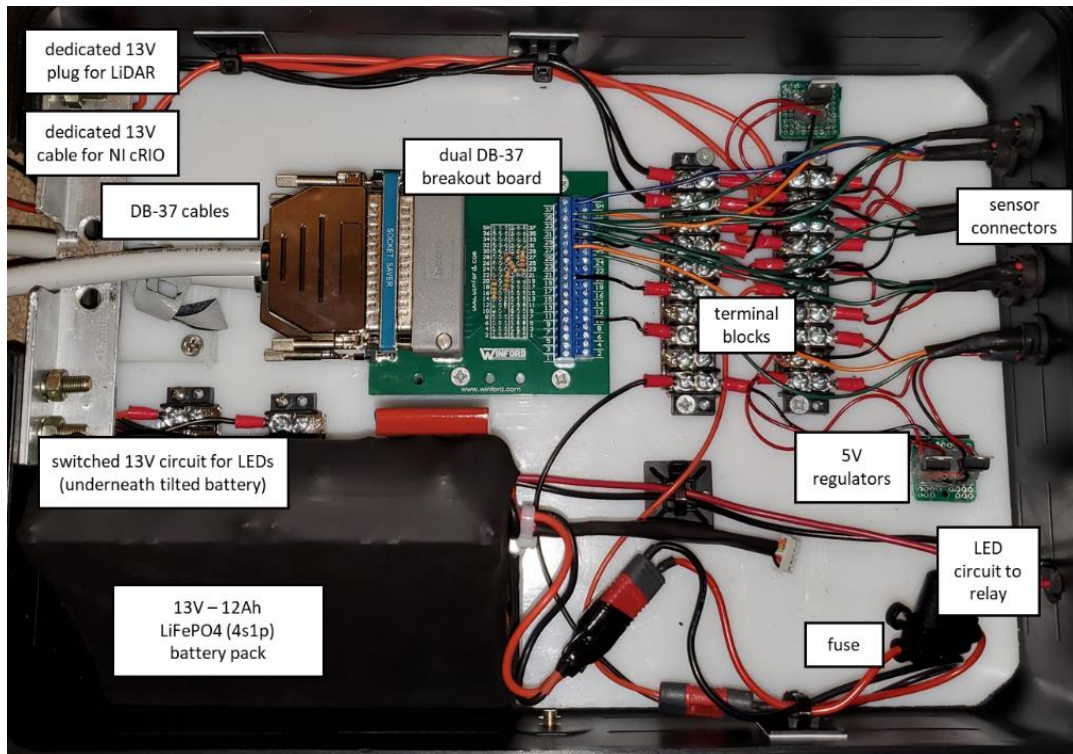


Figure 5-12. Tractor junction box internals.

Figure 5-13 shows the internals of a trailer junction box. All four trailer junction boxes used in the double straight-rail and drop-frame trailers were identical. Trailer JBs were slightly larger than the tractor junction box to provide space for the additional analog sensor connectors. In total, there were 13 installed sensor and relay connections on each trailer junction box. The signal wires from each sensor terminal connector are routed below the HDPE base to the breakout board so that the working area was not cluttered by cables. Because all wires were color-coded, the sensor connectors were placed nearest their corresponding channels on the breakout board and grouped by their supply voltage for easy diagnosis of any unforeseen issues and quick disconnections. Figure 5-14 illustrates the pin assignments for each sensor connection as viewed from the outside of the JB. This schematic was printed inside of the lid of the trailer JBs for easy

diagnostics in event of sensor or soldering failure and to act as a guide for future users. The lettering for each connector was a short-hand notation for the sensor type and location. For example, LC-D was the outrigger load cell on the driver (left) side and PT-B-D was the brake chamber pressure transducer on the driver (left) side.

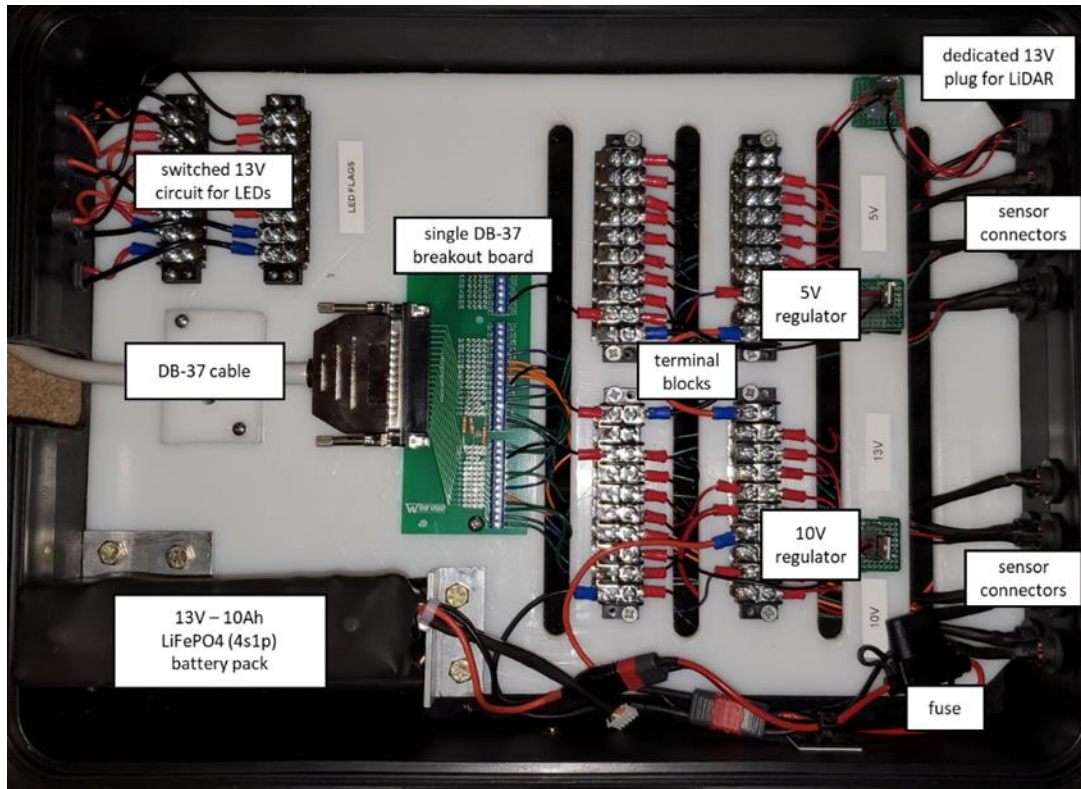


Figure 5-13. Trailer junction box internals.

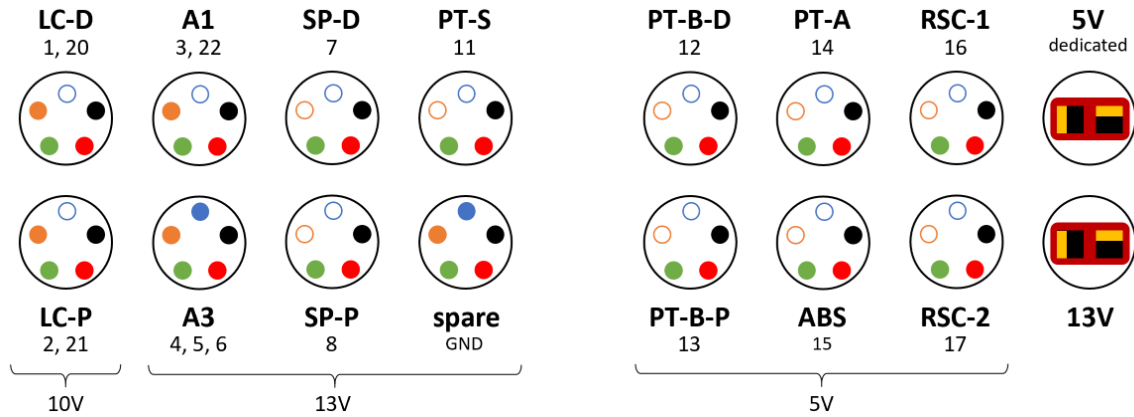


Figure 5-14. Sensor connector wired signals and supply voltage on the trailer JB as viewed from the outside. The code names for each sensor/relay are listed next to the respective connections (i.e.: LC-D = driver side load cell).

5.2.4 Data Acquisition and Synchronization

The suite of data acquisition equipment included the National Instruments CompactRIO (NI cRIO) controller, its attached analog-to-digital (A-DC) converters, a host PC laptop to control the NI cRIO and run the LiDAR software, and several Go Pro cameras. Each of these items are detailed in the following sections. A brief description of how the data was synchronized between different items is also provided. The table below lists the sampling rate for each system.

Table 5-7. Measurement sampling rates.

Sensor	Sampling Rate
Analog sensors & relays (NI cRIO DAQ)	250 Hz
GPS	5 Hz
LiDAR	50 Hz
GoPro Cameras	60 fps

(a) National Instruments CompactRIO

The National Instruments CompactRIO (cRIO) is a real-time embedded industrial controller for control systems and data acquisition. In our application, the cRIO was used solely to collect and record information from all analog sensors in the tractor and trailers as well as the GPS receiver on the tractor. The cRIO system is configurable such that the NI-9024 controller supports a field programmable gate array (FPGA) chassis that connects different modules to the controller. For our tests, two NI-9205 analog-to-digital

converter (ADC) modules collected data from the analog sensors. These modules have 16 differential or 32 single-ended analog voltage inputs up to ± 10 V and may be configured to accept a combination of differential and single-ended inputs as long as the total channel count does not exceed 32. The channel assignments used throughout testing are listed in Appendix E. Analog sensor output voltages connected to the ADC modules via DB-37 cables. Both the tractor and front trailer analog sensors were connected to Module 1 via a single DB-37 cable, and the rear trailer analog sensors were connected to Module 2. The NI cRIO system communicates with the host PC via an ethernet cable. The cRIO and ADC modules are shown in Figure 5-15.

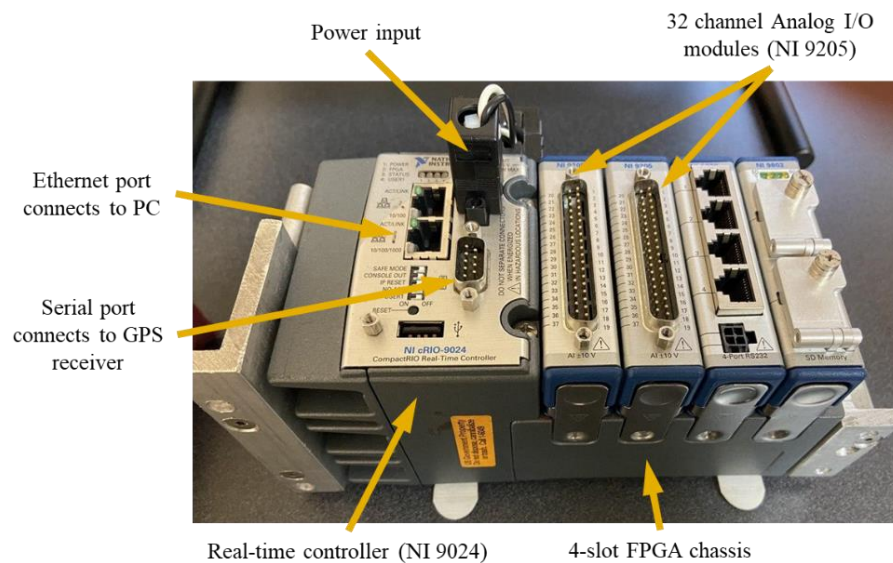


Figure 5-15. National Instruments CompactRIO system used for data acquisition.

The CompactRIO controller, FPGA, and I/O modules were programmed with LabVIEW, National Instruments' graphical programming language illustrated in Figure 5-16. Programming is done on the host PC running Microsoft Windows® and is deployed to the cRIO via ethernet. There are two main windows within a LabVIEW code: the front panel (gray background), which serves as the user interface, and the block diagram (white background) where code is developed. The code developed for the study acquires data in a high-priority timed loop, stores it in a buffer, and deterministically transfers it to a lower priority loop to display to the user interface. The front panel for this code includes a live feed of data for monitoring the recorded data during field tests. This was crucial for ensuring sensor signals were collected and the values appeared appropriate for the tests conducted before leaving the track.

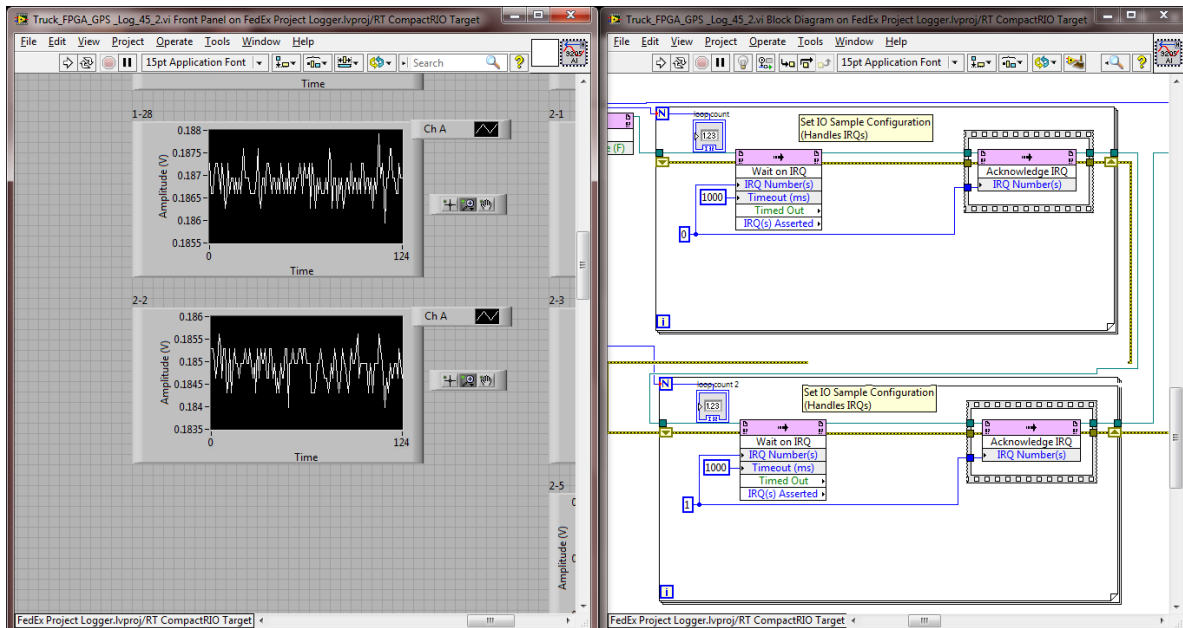


Figure 5-16. LabVIEW interface. The window on the left is the front panel of the block diagram code on the right. Only a very small portion of these programs is displayed within this screenshot.

(b) LeddarTech® Configurator

The LeddarTech® IS16 LiDAR sensors that measure articulation angles did not directly communicate with the NI cRIO but instead utilized LeddarTech®'s proprietary software application for recording data on the host computer. This application provided an easy way to change sensor parameters such as sample rate, measurement range, and emitter intensity as well as display a connected sensor's measurements in real-time as demonstrated in Figure 5-17. The software application can only connect to and record data from one LiDAR sensor. During testing, three or four applications on the host PC were running simultaneously when collecting data: one application of LabVIEW running the NI cRIO and two or three Leddar Configurator™ applications recording data from LiDAR sensors, depending on how many were used for a given test. By nature of running multiple applications, the NI cRIO and LiDAR applications begin and finish recording at different times. To correlate LiDAR data with the analog sensors recorded by the NI cRIO, files were synchronized in post-processing.

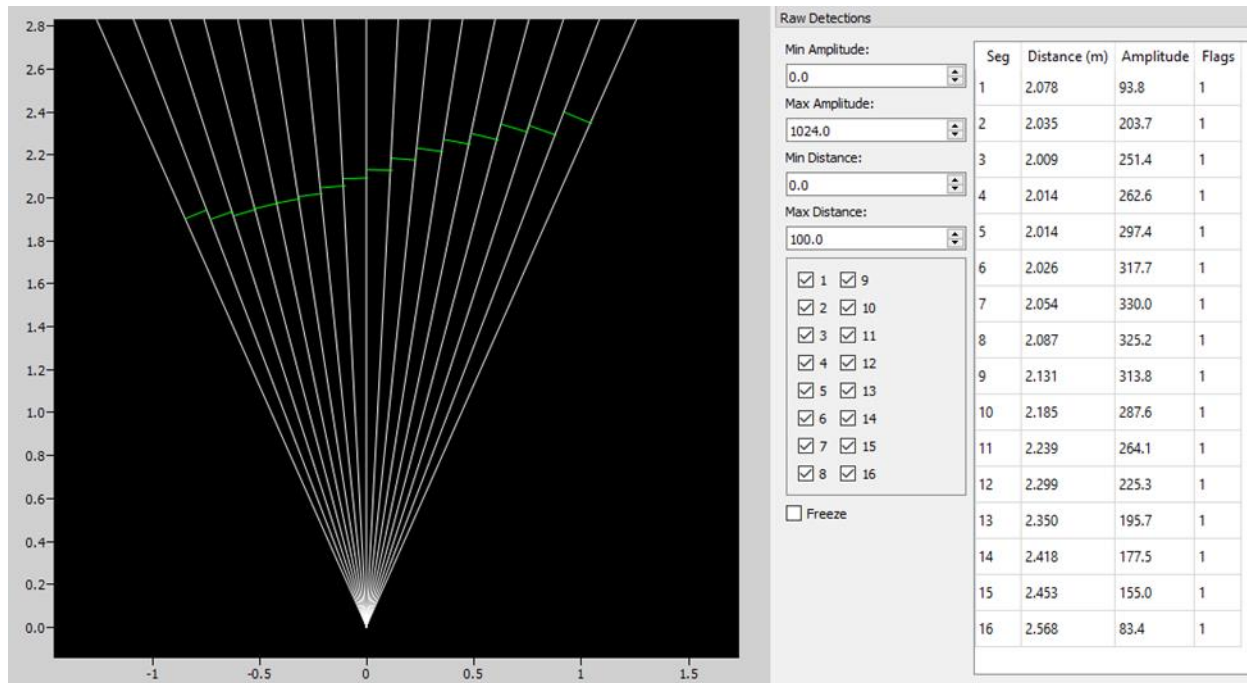


Figure 5-17. Screenshot of Leddar Configurator™ software for real-time display of sensor data.

(c) Data screening and Synchronization

Before analyzing all data, the LiDAR, GPS, and analog sensor data were synchronized in time. Because the GPS receiver's data was recorded by the NI cRIO with the analog channels, the start and end time of the GPS and analog channels were already synchronized. However, the GPS receiver was sampled at 5 Hz whereas the analog sensors were sampled at 250 Hz, so in addition to having different start and stop times, the data sets had different time steps. Synchronizing the data was done in post-processing before analysis. After the LiDAR data was synchronized to the same reference time as the cRIO, it was truncated to the individual maneuvers just like the data from the cRIO as determined by the maneuver flag and spatial trigger. Data was screened for abnormalities and corrected or omitted accordingly through the synchronization process. For example, the GPS receiver may momentarily miss a data point due to interrupted communication with satellites. For this type of scenario, missing data points were determined by linearly interpolating the neighboring data points. This ensured that all data arrays maintained constant time periods between successive data points.

(d) Video Recording and Synchronization

Eight GoPro Hero® 8 cameras were mounted to the tractor and double trailers to capture different vehicle dynamic events such as outrigger contact, wheel lift, wheel lockup, etc. Videos were useful to clarify uncertainty within the recorded data from sensors and to provide a wholistic perspective on the vehicle's

dynamics. Individual power stations within the trailers or on the outriggers supported the GoPros for continuous data collection throughout the day. Memory cards were swapped periodically to prevent data loss from lack of available storage. Figure 5-18 shows the mounting locations of the cameras and power stations. In some later testing, additional GoPros were included on the front trailer's outrigger arms as well as trackside on tripods or carried by observers.

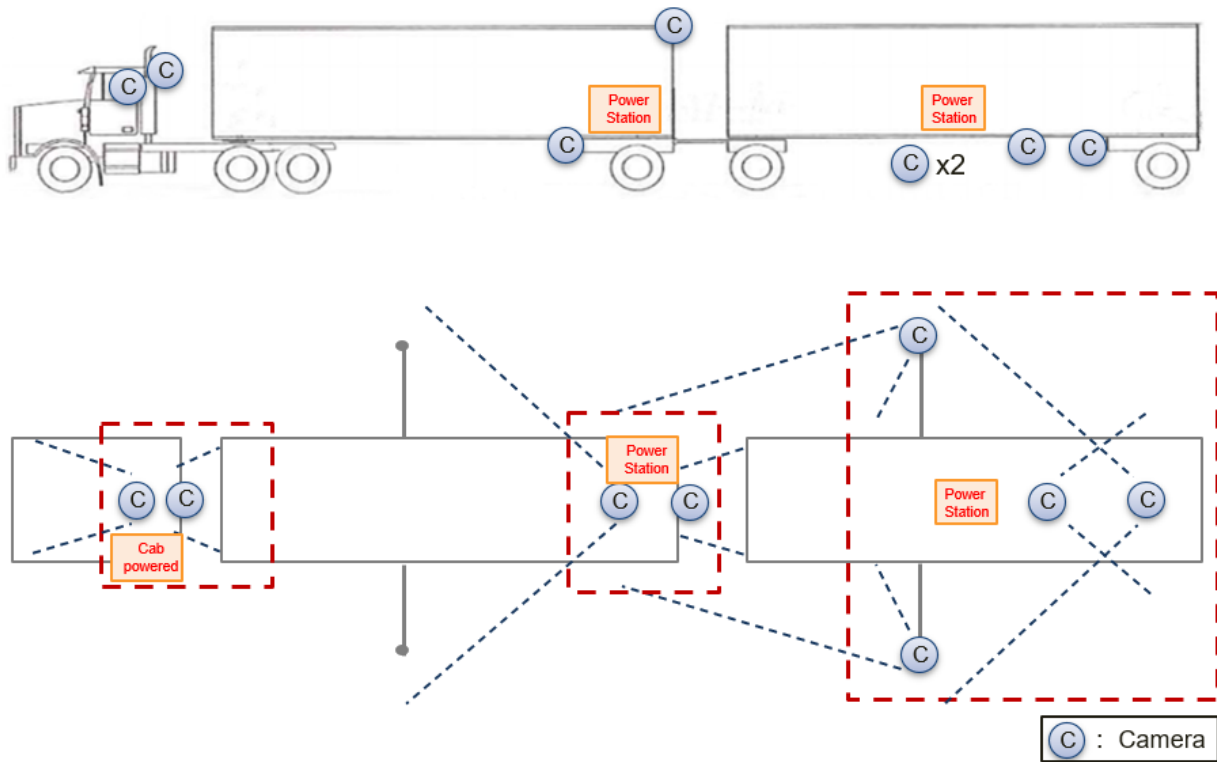


Figure 5-18. GoPro mounting locations.

LEDs within each camera's field-of-view were powered by the tractor junction box and illuminated for the entire duration of each maneuver. These LEDs were switched on by the spatial trigger and switched off whenever the test operator released the standby switch for the steering robot. The LEDs provided an easy means to check individual maneuvers by referring to the time stamp of a maneuver within the recorded test session.

5.2.5 Cumulative Measurement Uncertainty

Chapter 4 detailed individual measurement and procedural uncertainties to accurately quantify the confidence in the low-speed study's tests results: peak off-tracking. In Chapter 4, each measurement's uncertainty was related to the peak off-tracking result through the sensitivity analysis. Throughout the roll-

stability and brake type studies, the measurements are the results themselves and thus do not require an additional sensitivity analysis. Here, only each measurement's uncertainty will be discussed for the sensors related to the high-speed studies. Each measurement has uncertainty dominated by the following factors:

- Calibration
- Installation
- Noise

Section 5.2.1 covered calibration and installation uncertainties for each measurement type. Section 5.2.2 introduced the type of digital filters applied to data in post-processing. The filters' cutoff frequencies were carefully chosen to reduce measurement noise without hindering the signal within the bandwidth of interest. Of course, some of the actual measured dynamics may be skewed by the filter, but we will assume that this is negligible and does not impact the measurement uncertainty within each measurements' bandwidth of interest. The low pass filters effectively reduce measurement uncertainty caused by high-frequency electrical noise characteristic of the test environment, sensor output type (differential vs. single-ended), wire length, junction box wiring, and data acquisition system ADC. It is assumed that the measurement uncertainty at the 95% level caused by electrical noise is characterized by twice the standard deviation of the signal during a static measurement. Because the electrical noise's impact on the measurement is largely dependent on the sensor's type and output rather than the sensor's measurement itself, the values shown in Table 5-8 correspond to the average uncertainty from noise of all installed sensors of the same type. For example, some pressure transducers measured brake chamber pressure while others air tank or suspension air spring pressure. The average noise of all pressure transducers was taken as the uncertainty parameter. Figure 5-19 provides an example of the difference between raw and filtered data of a triaxial accelerometer's measurement.

Table 5-8. Measurement uncertainty (95%) in engineering units caused by electrical noise for each sensor type.

Sensor	Units	Raw	Filtered
Triaxial Accelerometer	[g]	0.078	0.024
Single-axis Accelerometer	[g]	0.007	0.001
Pressure Transducers	[psi]	0.48	0.21
Suspension String Potentiometers	[in]	0.005	0.001
Steering String Potentiometer	[deg]	0.32	0.21
Outrigger Contact Force	[lbf]	768	100

Tractor-Trailer Articulation Angle	[deg]	0.41	0.24
Trailer-Trailer Articulation Angle	[deg]	0.61	0.39

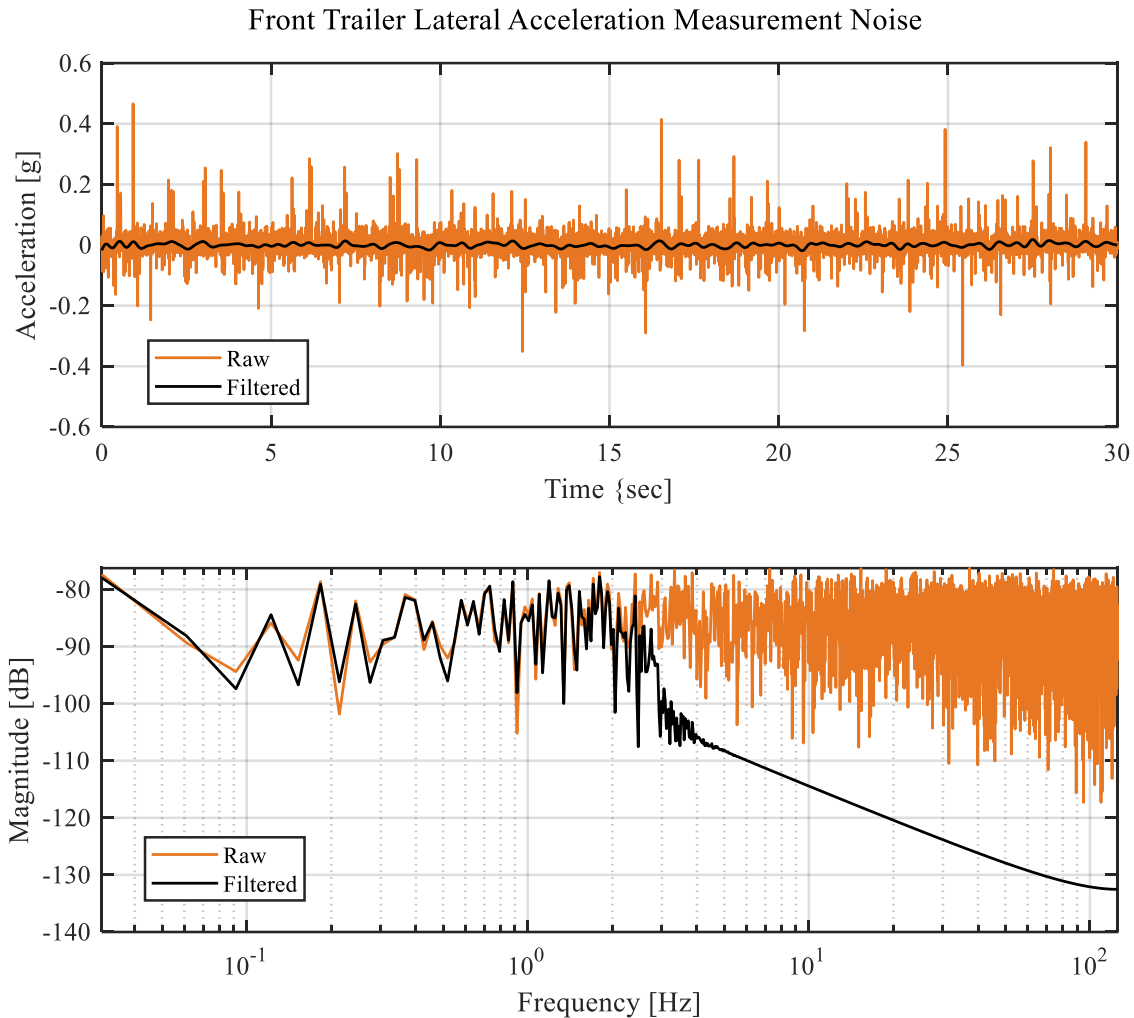


Figure 5-19. Example of signal noise from front trailer triaxial accelerometer signal for lateral acceleration.

Resistance within the analog-to-digital converter (ADC) can also impact measurement accuracy. If the ADC's resistance is too low, current can flow through the ADC and reduce the detected voltage compared to the sensor's output. This is called a loading error. Because the NI 9205 ADC modules have an input impedance $>10\text{ G}\Omega$, this is not an issue and can be neglected. Table 5-9 provides the estimated cumulative measurement uncertainty in engineering units for each sensor type. As discussed in Chapter 4, the total uncertainty is the root-sum-squares (RSS) of each uncertainty component for the measurement. For all analog data, the least-significant bit from conversion to digital data is several orders of magnitude less

than the output signal voltage. Similarly, GPS and articulation angle data points are not limited to specific resolution during recording. Thus, the zeroth order uncertainty is negligible and not listed in the table. Uncertainty in the LiDAR's articulation angle measurements were taken from Chapter 4. As detailed in Section 5.2.1, the accelerometer's calibration and installation uncertainties may be assumed negligible. Noise in the GPS position and speed data measurements is captured within the manufacturer's specifications. These were not filtered, and the raw data points are used throughout analysis. Note that the GPS speed's uncertainty is listed for steady-state speed. Speed uncertainty is expected to increase slightly with acceleration. However, the speed data's primary purpose is to determine the entry speed to a maneuver. Just before the maneuver, the driver typically has the vehicle at a relatively constant speed, permitting the use of the steady-state speed uncertainty value.

Table 5-9. Total measurement uncertainty by sensor type in engineering units.

Measurement	Units	Calibration	Installation	Electrical Noise (after filtering)	RSS
Suspension String Potentiometer	[in]	0.003		0.001	0.003
Steering String Potentiometer	[deg]	6.7		0.21	6.70
Triaxial Accelerometer	[g]	-		0.024	0.024
Single-axis Accelerometer	[g]	-		0.001	0.001
Pressure Transducers	[psi]	0.1		0.21	0.23
Tractor-Trailer Articulation Angle	[deg]	0.286	0.45	0.24	0.59
Trailer-Trailer Articulation Angle	[deg]	0.286	0.81	0.39	0.94
WAAS GPS Position	[ft]	10			10
WAAS GPS Speed	[mph]	0.23			0.23

5.2.6 Preliminary System-wide Shakedown Testing

Preliminary shakedown testing was conducted at the Virginia Tech Duck Pond parking lot shown in Figure 5-20. More than 130 test trials, including 150-ft J-turn, 75-ft constant-radius turn, and straight-line braking, were conducted for the double straight-rail and drop-frame trailers. Space constraints limited maneuver test speeds to 25 mph or less. The preliminary testing's main purpose was to validate the sensors' and data acquisition's functionality. Prior to testing, an instrumentation failure modes and effects analysis (FMEA) was conducted, and a checklist of high-priority failures was developed. An issue log tracked potential problems and failures that occurred prior to and during preliminary testing. In addition to identifying any existing or potential problems with sensors or wiring, the parking lot testing also provided valuable

experience to decide on the type and quantity of spare sensors that would be taken to the test facility, the Transportation Research Center (TRC) in East Liberty, OH. During the shakedown testing at Virginia Tech, four instrumentation issues occurred: two soldering failures (out of >400 solders), one pressure transducer failure, and a fray on a string potentiometer extension cable. The failed pressure transducer had been reused from past studies, so spare pressure transducers were prepared in the event other reused sensors were to fail while at the test track. Additionally, several spare sensor cables were soldered in preparation for a similar soldering or cabling failure. All encountered issues were resolved before concluding preliminary testing to ensure all data could be collected reliably before traveling to TRC.



Figure 5-20. GPS paths of preliminary testing maneuvers (Commonwealth of Virginia, Maxar Technologies, 2022).

5.3 Steering Robot

A steering robot design and built by CVeSS was used for track testing. This section provides an overview of the robot and the author's upgrades for the 33-ft double trailer roll-stability and brake type testing.

5.3.1 Overview

The steering robot's purpose is to provide repeatability and accuracy beyond what an experienced driver can achieve. This was critical to fairly assess the vehicle's dynamics, RSC intervention, ESC intervention, etc. for various tested loading conditions and trailer configurations. The basic design of the steering robot

includes a servomotor attached to the steering column, a RoboteQ® motor controller, and a box which houses the controller, batteries, and accessory circuits such as an E-stop and relays. The control box sits inside the cab between the driver and passenger seats, and the servo motor attached to the steering column was mounted inside the engine bay so that it did not interfere with normal steering when the steering robot is not used. Figure 5-21 shows the servomotor mounted inside the engine bay and attached to the steering column through a gear reduction and chain.

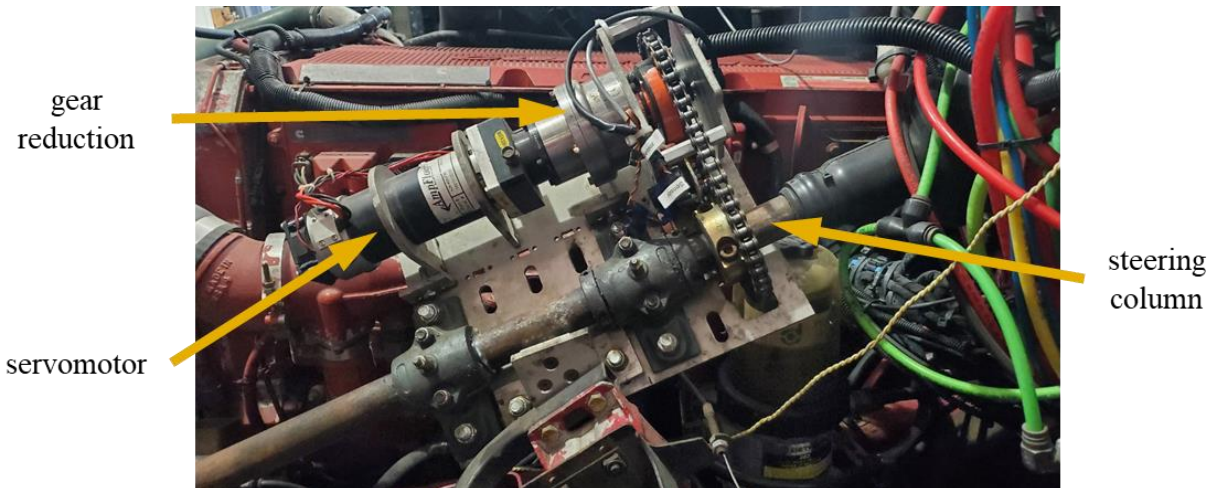


Figure 5-21. Steering robot servomotor mounted on the steering column within the engine bay.

The RoboteQ® controller runs a MicroBasic™ script that commands the servo motor to rotate to programmed positions at specific time steps. During a maneuver while the controller runs the main script controlling the servo motor, the controller activates a relay connected to the DAQ. This relay was the steering robot flag mentioned earlier in Section 5.2. The robot will only operate the servomotor while a “deadman” switch is held down by the robot operator (passenger) and the E-stop is released. As a safety measure, if the deadman switch is released, the controller immediately exits the section of code that provides commands to the servomotor and the driver can retake control of the steering wheel. If the E-stop is hit, the entire system is de-energized and the driver can immediately retake control of the steering wheel. The E-stop is accessible to both the driver and passenger while the deadman switch is only operated by the passenger.

5.3.2 Upfitting the CVeSS Steering Robot

There were three main changes to the steering robot from the 2016 to 2020 studies. These include: (1) addition of spatial trigger, (2) new battery packs, and (3) addition of a braking robot relay. The spatial trigger is simply a photoelectric sensor that was mounted in front of the tractor facing laterally. A reflective

target (a free-standing board with reflective tape) was staged at the maneuver start point and when the photoelectric sensor observed this target the robot controller began the steering script. This was illustrated earlier in Figure 5-3. Rather than the robot operator manually triggering the steering robot using the deadman switch, the deadman switch was used to put the robot on standby and search for the reflective target at the maneuver's starting gate. Once triggered, the deadman switch was held down for the duration of the maneuver for the robot to continue steering. The control box wiring and controller software were configured so that a locking switch permitted switching between modes of requiring the spatial trigger or only needing the deadman switch. For all testing, however, the spatial trigger was 100% successful in identifying the starting gate, initiate the steering robot, and flag recorded data for both driver and robot steered maneuvers. The process flow for completing a robot-steered maneuver is presented as Figure 5-22. The same procedure was followed for driver-steered maneuvers, but the robot servo was replaced by the driver.

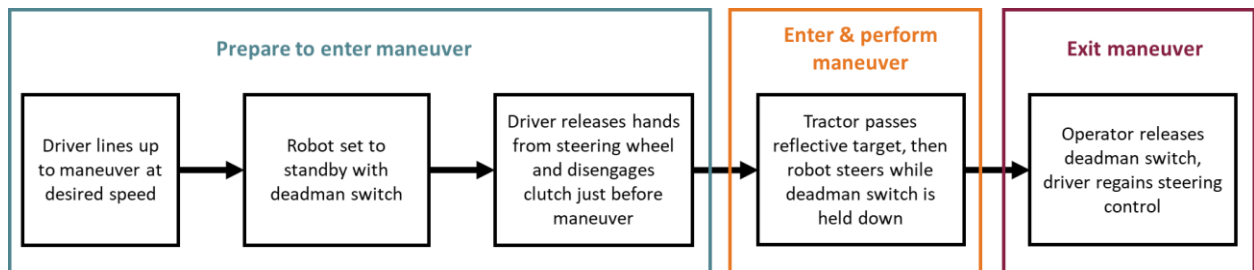


Figure 5-22. Operation flow for robot-steered maneuver.

The second change to the steering robot was new battery packs. The original Max Amps® 6.2Ah, 23.1V 7s1p LiFePO4 battery packs began to swell from age and use, and the lack of available direct replacements necessitated a search for new battery packs. After extensive research on battery chemistries and available battery packs that could supply sufficient power, current, and energy, Zippy® Compact 4.2Ah, 13.2V (4s2p) LiFePO4 battery packs were chosen. Lithium-iron phosphate (LiFePO4) battery chemistry provides immense power density and decent energy density while being significantly more stable than more powerful but often unstable lithium-polymer (LiPo) batteries. Table 5-10 lists the changes in battery specifications including total voltage for the system and the expected maximum current draw by the steering robot for the given voltages. As shown below, the new battery packs were spec'd to surpass the steering robot's power requirements while adhering to the controller's maximum input voltage of 50 V. Because the previous batteries provided sufficient energy to power the steering robot for more than one full day of testing, it was expected and confirmed that the new, Zippy®, batteries provided more than

enough energy for a full day of testing. The expected maximum current draw was a conservative estimate based on past test data and the servomotor and RoboteQ® controller specifications.

Table 5-10. Old and new battery pack specifications for the steering robot.

	MaxAmps® Green Series (old)	Zippy® Compact (new)
Cell Configuration (individual pack/total)	7s1p / 14s1p	4s2p / 12s4p
Voltage (V)	23.1 V (x2) = 46.2 V	13.2 V (x3) = 39.6 V
Capacity (Ah)	6.2 Ah	4.2 Ah (x2) = 8.4 Ah
Total Capacity (Wh)	286 Wh	332 Wh
Max discharge rating (1C = 1A/Ah)	40C	30C
Expected max current draw (A)	60 A (9.7C < 40C)	70 A (8.3C < 30C)

The third major update to the steering robot was the addition of a relay circuit for the braking actuator. This addition provided a means to use the steering robot controller and spatial trigger to activate the pneumatic actuator used for robotic braking. Because this circuit was an addition to the existing steering robot, simultaneous robotically steered and braked maneuvers could be performed.

The controller, battery packs, power and accessory electronics were housed inside the control box which sat between the driver and passenger seats. Two contactors were at the center of the box, one for powering the controller and accessory electronics (relays, feedback sensor, deadman switch, etc.) and one for connecting the servomotor to the controller's output. The RoboteQ® HD460 controller sat on the right, underneath the flag and braking robot relays, and the battery packs sat to the left. The control box is shown in Figure 5-23 along with the deadman switch, photoelectric sensor, and E-stop. As mentioned before, these components sat outside of the control box; the deadman switch was held by the robot operator in the passenger seat, the E-stop was mounted on the dashboard readily accessible to both the driver and passenger, and the photoelectric (photoelectric) sensor was mounted outside on the front bumper of the tractor. At the time this picture was taken, the battery packs were set aside charging, but their placement within the box is indicated.

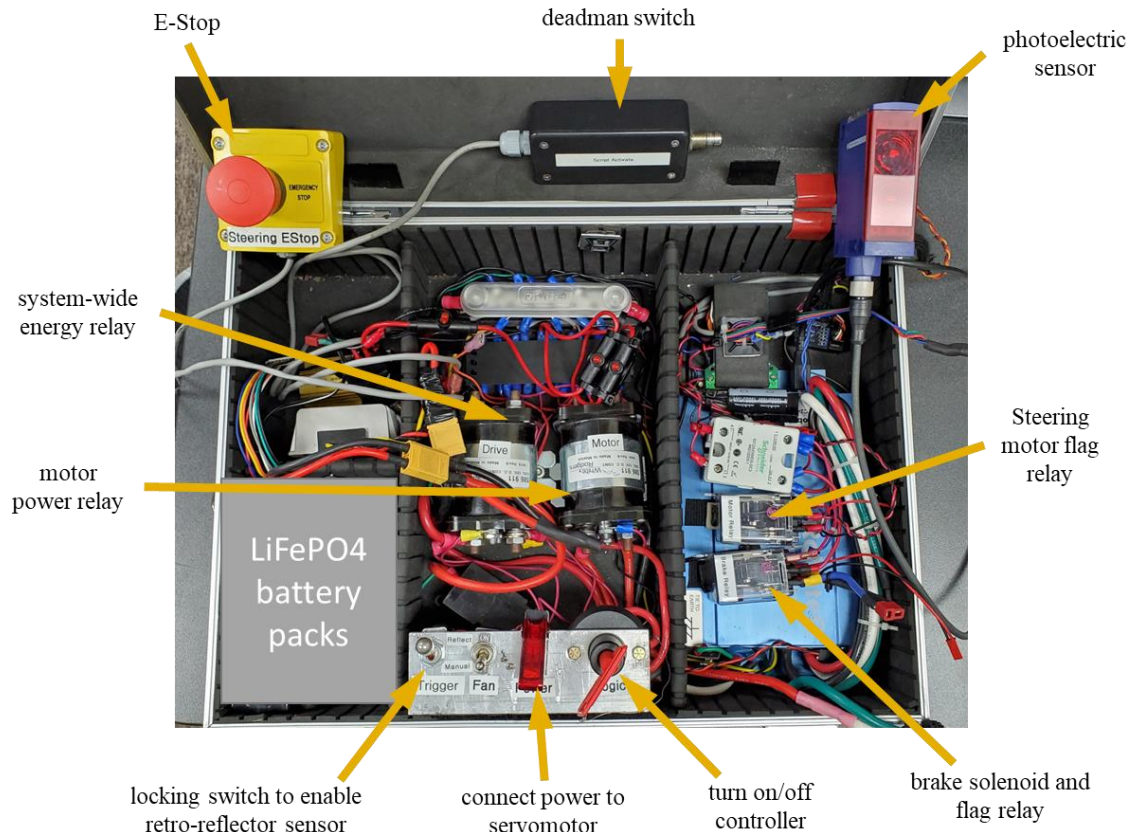


Figure 5-23. Steering and braking robot control box. The E-stop, deadman switch, and photoelectric sensor are also shown.

5.3.3 Steering Robot Performance

The steering robot was first tested in Blacksburg, VA during the instrumentation shakedown tests. Preliminary PID gains for the servomotor were identified. In this process and throughout several days of tests, the robot proved to be reliable and repeatable. The spatial trigger worked 100% for the time and the batteries were unphased by high power draws from the servomotor and long days of testing. Once at the Transportation Research Center (TRC) in East Liberty, OH, further tuning was conducted on the steering robot where the vehicle could be safely operated at the desired speeds for roll-stability testing. Table 5-11 presents the PID gains used for the 3 robot steered maneuvers. These maneuvers and their steering profiles are discussed in detail in Chapter 6. Figure 5-24 compares the steering wheel angle between the robot-steered and driver-steered J-turn maneuvers. These plots clearly demonstrate that the steering robot provided repeatability and accuracy beyond what an experienced driver could achieve.

Table 5-11. PID gains for the steering robot servomotor for each maneuver.

	Proportional (P)	Integral (I)	Derivative (D)
Robot J-turn	2.5	0	0
0.25 Hz SWD	5	0	0.7
0.5 Hz SWD	6	0	0.8

Driver steering **varies** significantly between test trials



Robot steering is **accurate and consistent** for all test trials

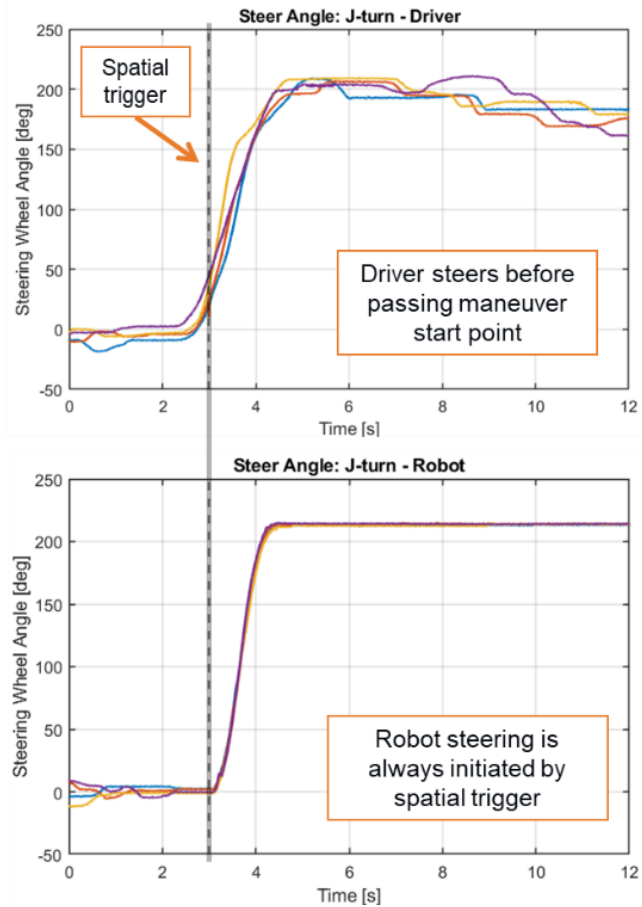


Figure 5-24. Steering wheel angle comparison of driver-steered (top) and robot-steered (bottom) J-turn maneuver during roll-stability testing at TRC.

5.4 Braking Robot

To conduct brake type testing, robotic braking was developed to ensure fast and consistent braking action that could be activated by the spatial trigger and recorded by the DAQ. Robotic braking action is provided by a pneumatic actuator mounted between the driver's feet and pressed the brake pedal. The fixture supporting the pneumatic actuator mounted to the cab floor using the same bolts that secured the driver's

seat. Longitudinal supports against the driver seat base secured the actuator and mount from lifting upward in reaction to the force against the brake pedal. The fixture raised the actuator approximately 4 inches above the floor so that the actuator could maintain clean contact with the brake pedal's extension throughout the entire brake pedal stroke. The fixture also held and guarded the 5-position, 2-way pneumatic solenoid valve that directed pressurized air to either extend or retract the pneumatic actuator. A pressure transducer sat within the fixture underneath the actuator to measure the air pressure supplied to the actuator during braking. The entire actuator, fixture, and brake-pedal extension were unintrusive to the driver, leaving clear access to the clutch, brake pedal, and accelerator pedal. When the actuator was not in use, a clevis pin inserted on the fixture prevented the actuator from extending. Figure 5-25 presents the actuator and its fixture installed in the tractor cab.

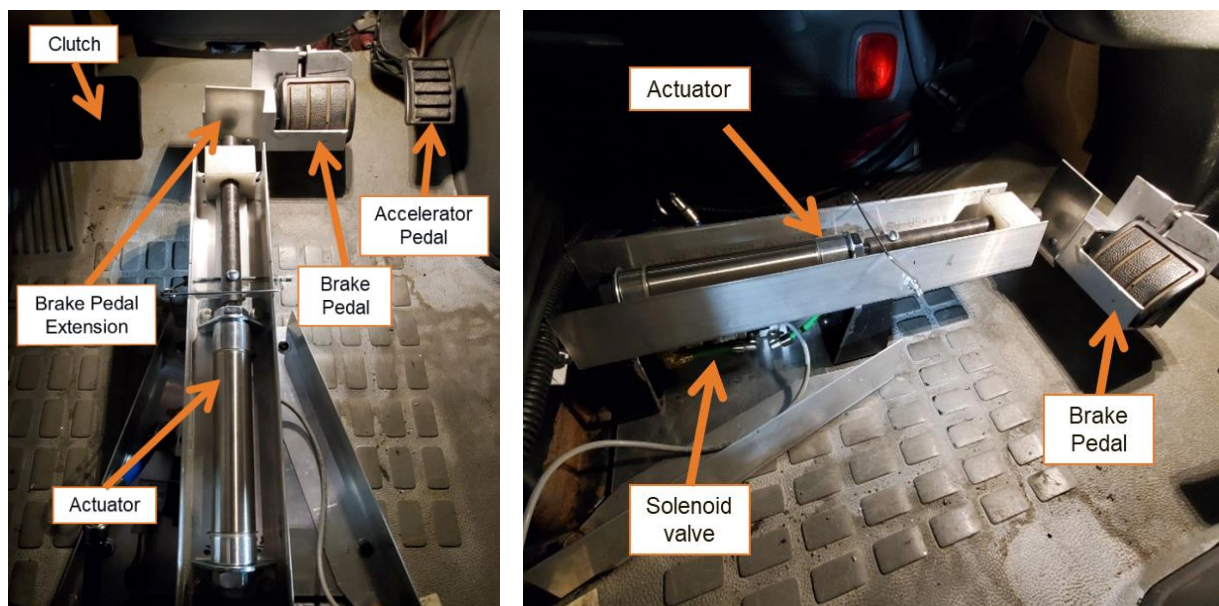


Figure 5-25. Braking robot pneumatic actuator installed in tractor cab.

5.4.1 Brake Pedal Extension

As a safety measure, it is imperative that the brake pedal could always be accessed by the driver, even when the braking robot was installed and operated during testing. A brake pedal extension is designed to ensure this accessibility while centering the actuator between the driver's feet. Due to the odd shape of the brake pedal and an inability to drill holes into it, the brake pedal extension had to be carefully designed so that it could clamp onto the brake pedal without inhibiting the driver's access and without limiting the pedal's stroke. This was especially challenging considering the tight space behind, below, and beside the pedal when it reached the end of its travel. The final design for the brake pedal extension uses two clamping

bolts to secure the extension around the brake pedal. One bolt squeezes the brake pedal arm between two pieces of aluminum angle, one hinged and one fixed. The other clamping bolt compresses the brake pedal pad against a lip of the extension which sat along the front bottom edge of the brake pedal pad. When activated, the actuator collides with the contact plate on the brake pedal extension. The contact plate is smooth so that the actuator's end can easily glide along the surface while the pedal rotated through its stroke. Force on the brake pedal extension's contact plate naturally induces a torque on the brake pedal arm. This torque is small and does not affect the brake pedal's motion nor reliability. A close-up view of the installed pedal extension is shown in Figure 5-26.

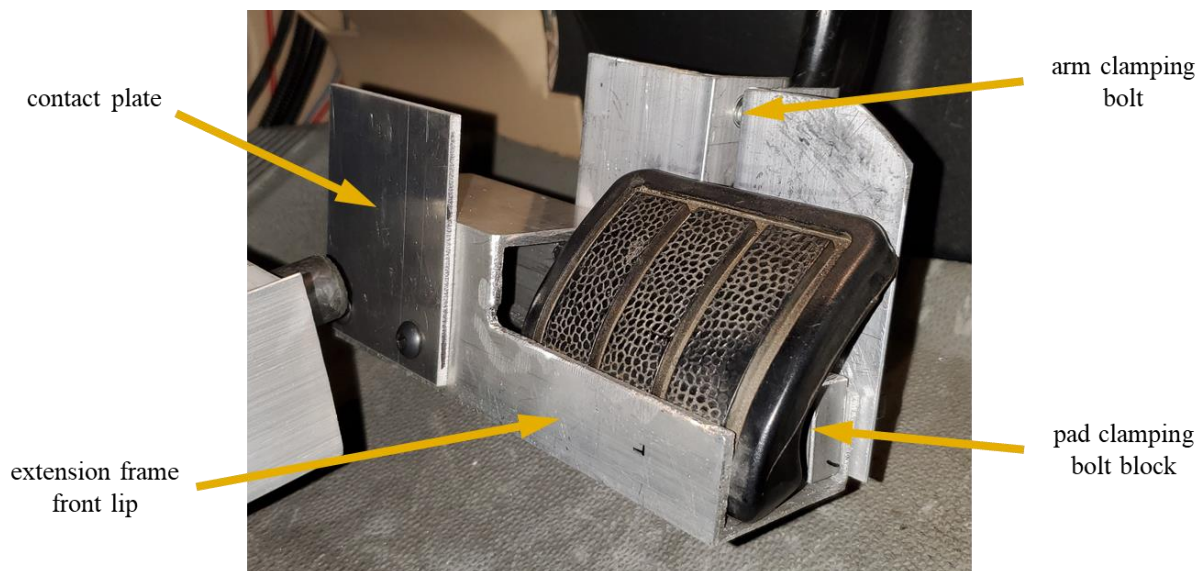


Figure 5-26. Close-up view of the installed brake pedal extension.

5.4.2 Braking Robot Pneumatics

Because the actuator is pneumatically driven, a consistently pressurized air supply had to be readily available for braking action. To achieve this, a small air tank is installed inside the tractor cab. This tank is constantly fed air supplied by the tractor's air tank through a pressure regulator. A relief valve is also installed to ensure that this air tank never exceeded an unsafe amount. Analog gauges are installed so that the robot operator (passenger) can monitor the tractor tank and robot air tank pressures throughout testing. In the short amount of time to reposition the tractor for the next braking maneuver, the small air tank for the actuator refills to the desired pressure, ready to provide the same actuating force on the brake pedal as a preceding test trial. Figure 5-27 presents the installed auxiliary tank with the analog gauges, valves, and pressure transducer.

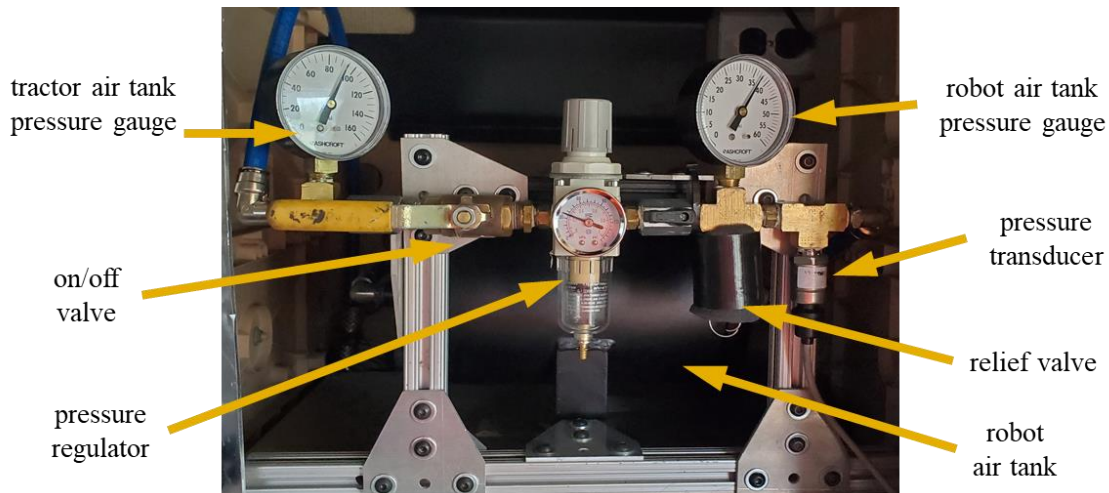


Figure 5-27. Pneumatic supply to the braking robot.

Figure 5-28 shows the plumbing arrangement for the braking robot system. Air is supplied by the tractor and regulated to a specific pressure for a designed brake pedal force. Throughout the brake testing at the Transportation Research Center (TRC), this pressure was set to 40 psi, resulting about 50 lbf against the brake pedal. The solenoid valve was controlled by the steering robot controller. When inactive (no voltage supplied), the valve routed pressurized air to the front end of the bi-directional actuator, forcing the actuator to retract and hold its retracted position. When 12-V power is supplied to the solenoid by the robot controller, the solenoid valve routes the pressurized air to the back end of the actuator to make it extend and push in the brake pedal. When switching between on and off states, air on the unpressurized side of the actuator is exhausted into the atmosphere. High-speed video recording (240 fps) shows that the pneumatic delay from passing the spatial trigger to the actuator striking the brake pedal extension was 40 milliseconds and 105 milliseconds to when the brake pedal reaches the end of its stroke. This coincides with the first and second peaks in the braking robot actuator's air pressure shown in Figure 5-29. The pneumatic delay is accounted for in all tests where it was relevant. The start of braking for tests detailed in Chapter 7 is the moment the braking robot actuator first contacts the brake pedal. The outside on/off valve was typically left on throughout a testing day that included the braking robot and switched off for all other times. The on/off valve inside the cab was only on during test sets that required the braking robot.

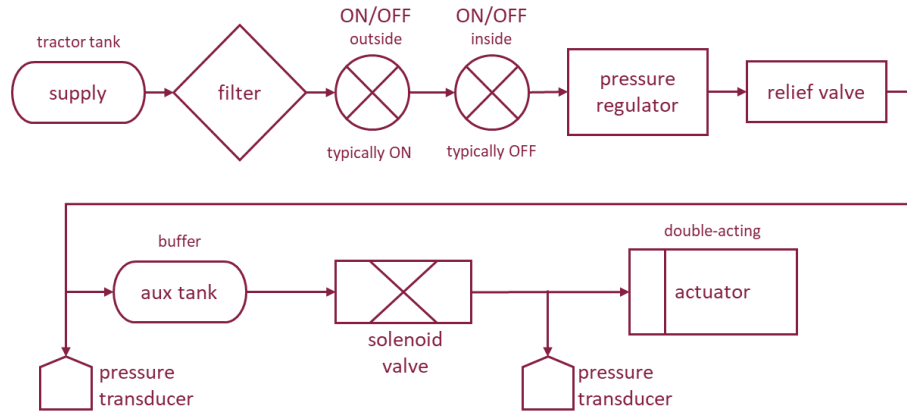


Figure 5-28. Pneumatic plumbing arrangement of braking robot system.

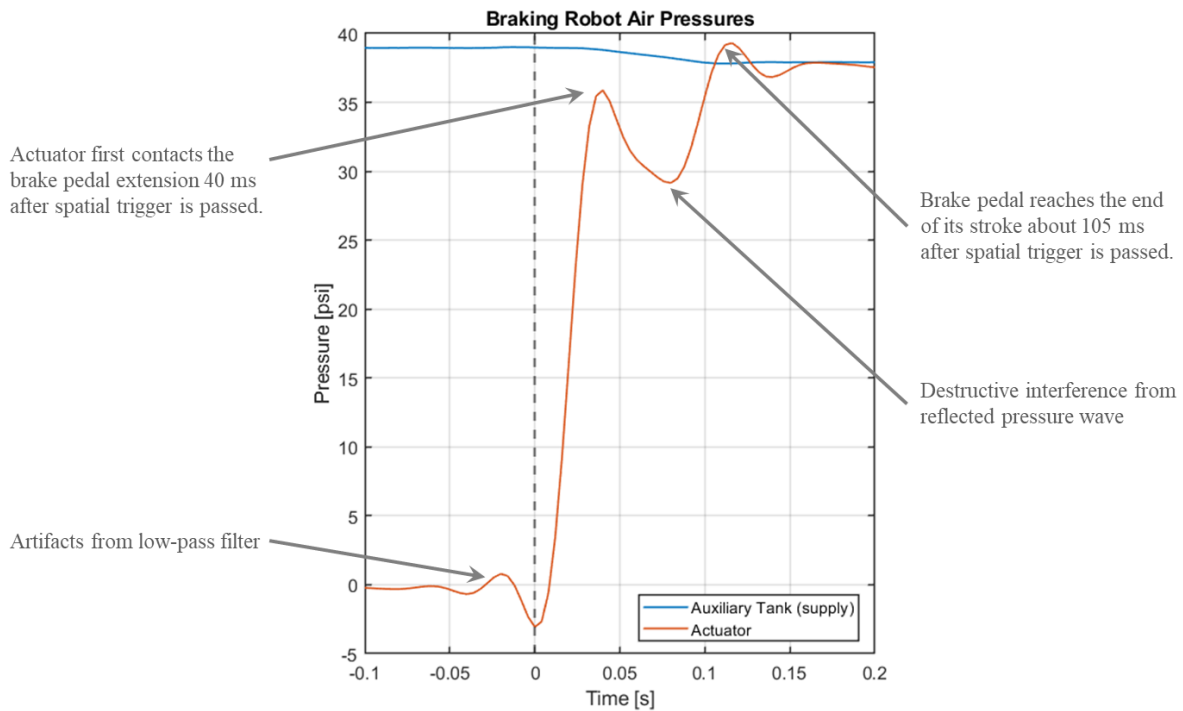


Figure 5-29. Braking robot pneumatic time delay.

5.4.3 Partial Braking

The braking robot is also designed to accommodate partial braking. Using the actuator and pneumatics already installed, partial braking could be achieved one of two ways: either reduce the auxiliary tank pressure or limit the actuator stroke. Because the brake pedal's force against the actuator may vary under different braking conditions and ABS intervention, reducing the auxiliary tank pressure would inherently have inconsistencies between test trials. Thus, it was decided that limiting the brake pedal stroke would

provide the most robust means to create repeatable partial braking force. The brake pedal stroke is limited by repositioning the safety-stop clevis pin to a point on the fixture that permitted some, but not all actuator motion. It was decided that the actuator stroke should be limited such that for partial braking, the brake chamber pressure should be between 30-50% of the brake chamber pressure under full braking.

A series of tests were conducted using spacers within the actuator fixture to determine the clevis pin location needed for partial braking. Figure 5-30 plots the average tractor drive axle brake pressure for each tested stroke reducer. As you can see, there is a very sharp transition from full braking to very little braking. For a stroke reducer width of about 1.5 in, a change in reducer width by 0.1 in would result approximately 16.5 psi (about 18% of full braking) change in tractor brake pressure, as illustrated by the linear fit between the two highlighted data points. This sensitivity made it challenging to accurately constrain the braking force. Figure 5-31 plots the brake chamber pressure on the tractor drive axle right side during a braking J-turn maneuver. Notice that the partial braking resulted in ~40% of the brake chamber pressure as full-braking for each vehicle unit. This was well within the target pressure and was consistent throughout testing. Some fluctuations in the brake pressure are caused by anti-lock-brake system (ABS) intervention to prevent wheel lockup.

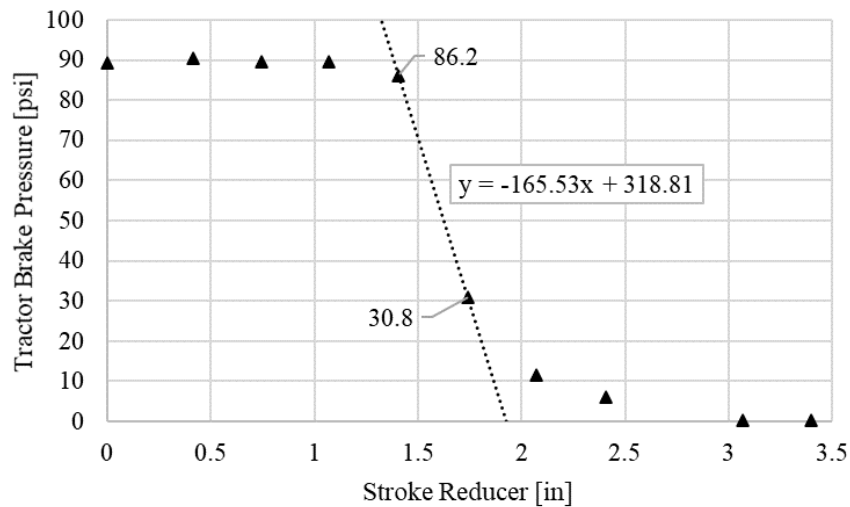


Figure 5-30. Tractor brake pressure as measured from different brake pedal actuator stroke reducers.

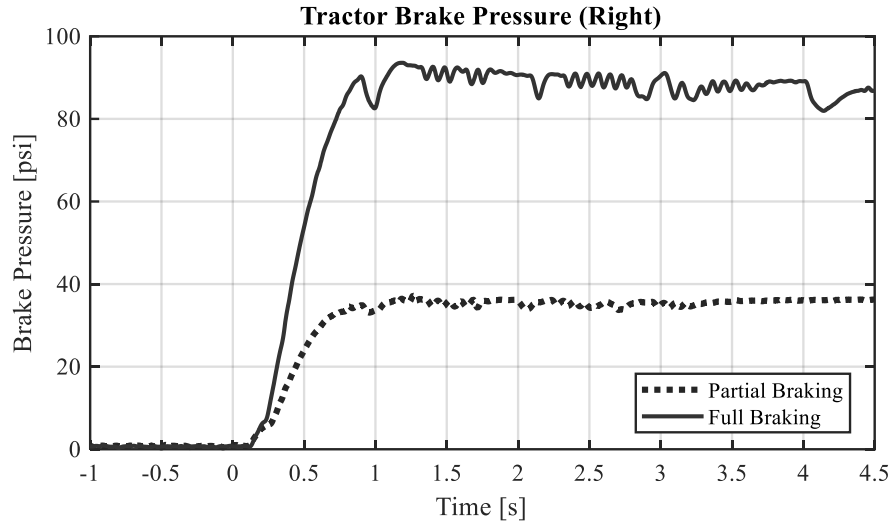


Figure 5-31. Tractor drive axle brake pressure on right side for full and partial braking J-turn maneuver.

5.5 Trailer Mechanical Safety Equipment

Significant structural reinforcements and mechanisms were installed on the test trailers to ensure they could withstand the rigors of repeated high-speed roll-stability testing. The mechanical design and installation of this equipment was conducted by other team members. Extensive finite element analysis (FEA) was performed on all mechanical systems to ensure the equipment would last through the test program. This section provides an overview of the systems and their function related to the roll-stability testing. Detailed CAD models and FEA are omitted for brevity as they were not the author's original work.

5.5.1 Outriggers

The outriggers prevent the trailer from completely rolling over and, as discussed in Section 5.2.1, utilize load cells to estimate the outrigger wheel impact force during a rollover event. The outriggers were positioned slightly forward of half the trailer length so that the front trailer outrigger could support roll from the tractor transmitted through the fifth-wheel hitch and to the trailer. The tractor did not have outriggers. The outriggers were made of three different components: arms, hinges, and inboard assembly. The arms extended outward holding a plastic spherical wheel that contacts and rolls along the ground during a rollover event. The outrigger wheels were plastic to limit the increase in the test trailer's roll inertia compared to a fully-loaded, road-going trailer. The hinge assembly connected the outrigger arms to the inboard assembly and allowed the arms' angles to be adjusted so that the outrigger wheel could be accurately positioned above the ground to ensure that the trailer rolls enough to experience wheel lift without rolling so much that the outrigger arm could not counter the trailer's roll momentum. The hinges also enabled the outrigger arms to be folded inward along the trailer length for transport. The inboard

assembly was mounted underneath the trailer and contained the load cells and hinge assembly. Figure 5-32 illustrates the outrigger. Each trailer had one outrigger assembly.

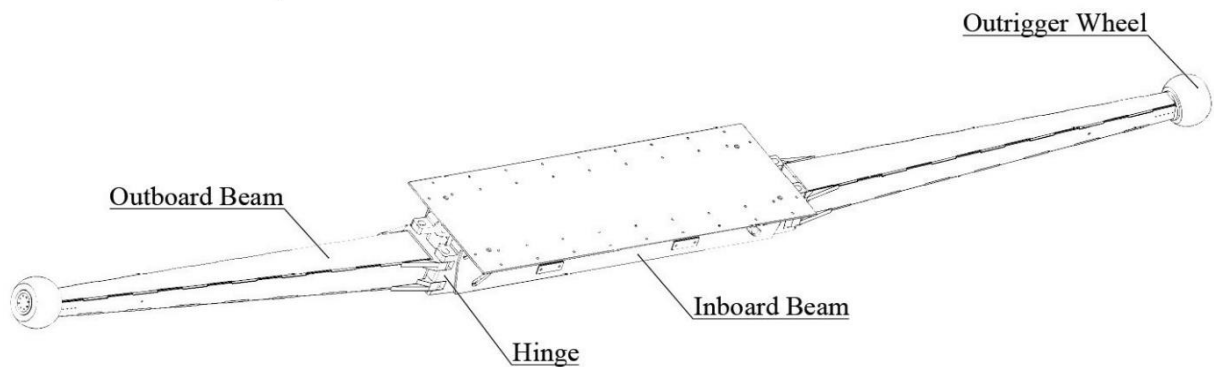


Figure 5-32. CVeSS outrigger assembly.

5.5.2 Anti-Jackknifing Mechanisms

As discussed earlier in Section 2.5, LCVs are susceptible to jackknifing. A jackknifing event during high-speed testing could irreparably damage the test equipment and pose a threat to driver and passenger safety. Anti-jackknifing mechanisms were installed on the test trailers to ensure that they would not collide during large articulation angles that occur during jackknifing. The jackknifing constraints include a series of kinetic ropes and grade 100 metal chains between each vehicle unit. As mentioned in Section 2.5, an A-double LCV has three locations of jackknifing: (1) tractor trailer, (2) front trailer to dolly, and (3) dolly to rear trailer. The ropes and chains were strategically arranged to prevent all three types of jackknifing. Pintle hooks attached to the trailer corners and the tractor frame rails provide attachment points for the ropes and chains. These were carefully positioned so that trailers could be switched in order of front to rear and for mixed trailer type testing (drop-frame and straight-rail) conducted for the project sponsor. The dollies were already equipped with hoops that could be used to attach the ropes and chains. The ropes were dimensioned to absorb some of the vehicle's energy before the metal chains create a hard stop in the articulation between vehicle units without interfering within the dynamics of interest for the roll-stability testing. The ropes did not engage until articulation angles well above the those that occurred during the roll-stability testing. Additionally, the added mass from the ropes and chains is negligible compared to the total vehicle mass. Thus, the anti-jackknifing mechanisms did not interfere with the vehicle's dynamics and the roll-stability test data. Figure 5-33 illustrates the rope and chain arrangement.

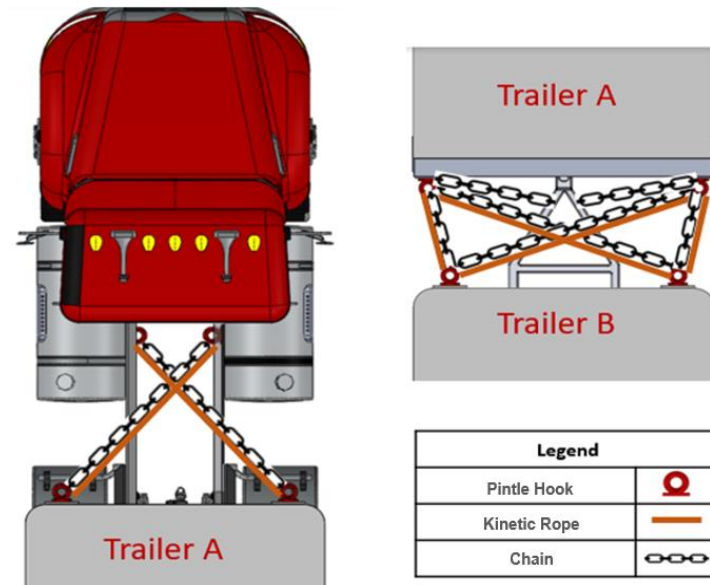
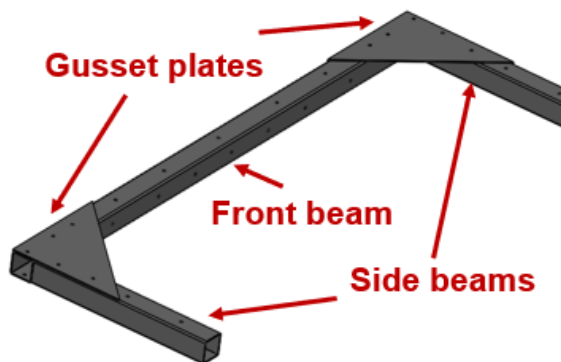


Figure 5-33. Anti-jackknifing system configuration.

5.5.3 Structural Reinforcements

If the jackknifing mechanism were to engage during high-speed tests, tremendous forces would be exerted onto the trailer body where the pintle hooks are attached. Yaw reinforcement structures were installed on each trailer bulkhead to distribute the load from the anti-jackknifing mechanism and prevent the trailer bulkhead from bending and twisting under high forces. The yaw-reinforcement structure was made of 0.25 in thick steel tubing and plates and fastened together by grade 8 hardware. Figure 5-34 presents the CAD model and picture of the installed yaw reinforcement structure. Figure 5-35 presents an exploded view of the CAD model of the pintle hook assembly fixed to the trailer and the yaw reinforcement structure.



(a)



(b)

Figure 5-34. Yaw reinforcement structure: (a) CAD model, and (b) installed in trailer.

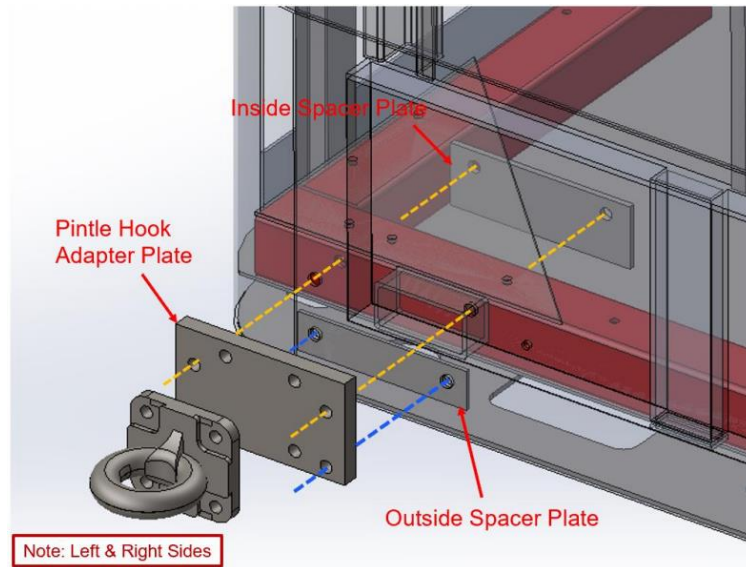


Figure 5-35. Exploded view of the pintle hook assembly installed on trailer front wall and yaw reinforcements.

During a rollover event, an outrigger wheel hits the ground preventing the trailer from completely rolling over. Significant forces from the outrigger are transmitted into the trailer body. Roll-reinforcement structures were installed to distribute these loads across the trailer body to prevent the trailer from bending and twisting. On the straight-rail trailers, the roll reinforcement structure was a set of 8x4 in steel tubing with 0.25 in wall thickness positioned in the shape of an “H”. Two lateral members spanning the trailer width provided attachment points for the outrigger inboard assembly. Tension rods were bolted through top plates, the trailer floor, the roll reinforcement structure, and the outrigger to hold everything in place. Two longitudinal members spanned numerous trailer lateral I-beams. These distributed the impact forces across a wide area of the trailer body. Figure 5-36 presents the roll-reinforcement CAD model and installation on a straight-rail trailer.

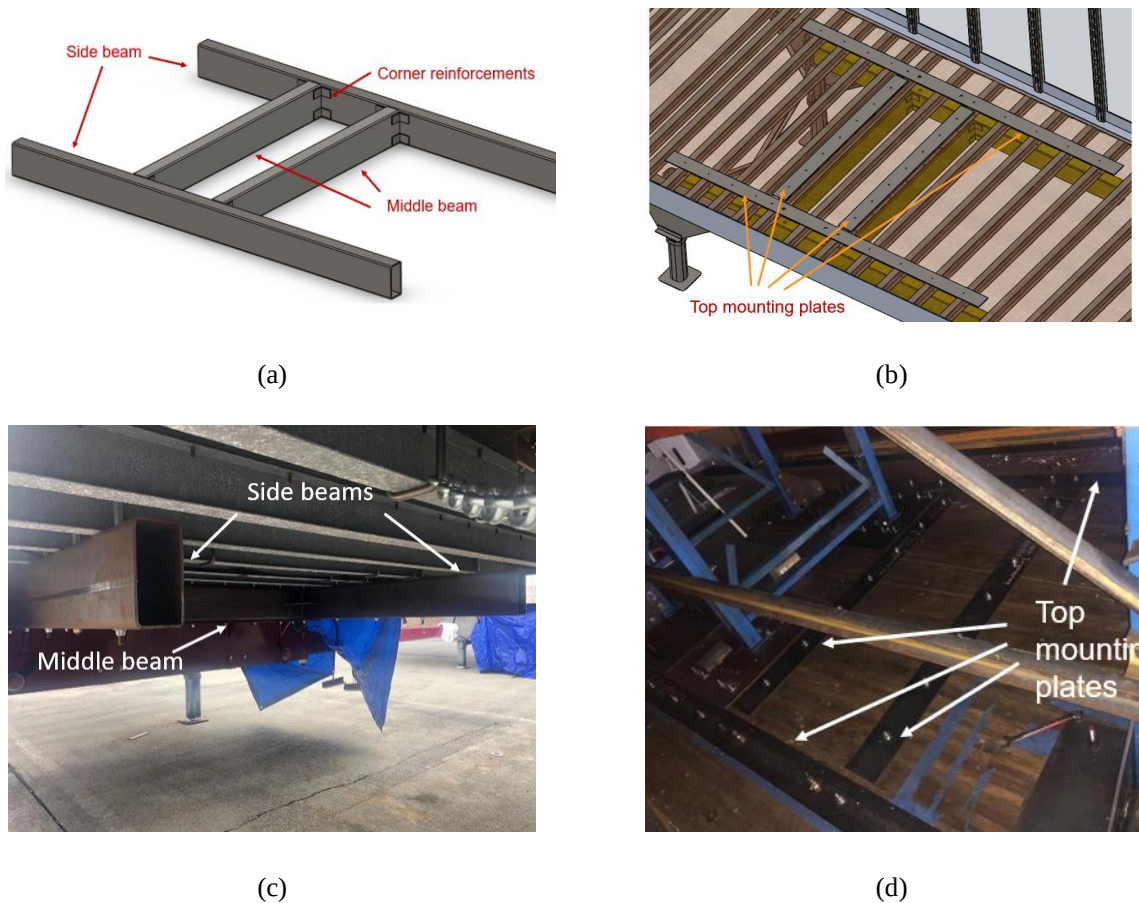


Figure 5-36. 33-ft straight-rail roll reinforcement structure: (a) CAD model of “H” shaped reinforcement structure, (b) CAD model of the top mounting plates, (c) “H” shaped reinforcement underneath the trailer, (d) top mounting plates on the trailer floor to hold the reinforcement structure.

5.5.4 Load Frames

To replicate actual loading condition and CG height in the test trailers, to the text possible, a multi-level load frame were installed on each trailer. The load frames were based on pallet racks and were extensively strengthened with bracing from steel angles and plates. The vertical posts were fixed to the trailer floor through steel plates and tension rods clamping down on several lateral I-beams. A truss structure on the left and right side fixed the top of the load frame to the top corners along the trailer wall. Each load frame included four wooden boxes that contained a total of about 150, 50-lb sandbags per trailer. Some boxes contained more sandbags than others to position the CG height and forward position appropriately. The boxes were strategically positioned high up to counter the lowered CG height from the attached roll-reinforcement and outrigger. The combined weight of the load frame, structural reinforcements, outriggers, and sandbags resulted about 15,000 lb payload on each trailer. Figure 5-37 presents the load frames with

sandbags in a straight-rail test trailer. The sandbags were held in place within the load frame boxes by wood planks and ratchet straps.



(a)



(b)



(c)

Figure 5-37. Installed load frame on 33-ft straight-rail trailer. (a) Back of load frame, (b) sandbags, (c) truss.

Chapter 6

Stability Control Systems' Effects on Trailer Rollover Propensity

This chapter details the test procedure, results, and analysis of high-speed roll-stability testing on 33-ft double straight-rail trailers. The study's focus is to identify rollover propensity of the heavily-loaded trailers through an array of maneuvers for different stability control system implementations. Trailer roll-stability is evaluated through rollover events determined by outrigger contact forces. Yaw motions are also discussed in terms of relative articulation between successive vehicle units. Stability control system intervention timing and duration is also discussed. Four different stability control system configurations were tested: stock (none), trailer roll-stability control (RSC) only, tractor electronic-stability control (ESC) only, and RSC+ESC. For RSC and RSC+ESC configurations, RSC was active on both trailers. Testing both trailers significantly reduced the number of test permutations, especially considering the additional tests conditions conducted for the project sponsor not discussed in this dissertation. This was necessary to conduct all essential tests in a timely manner for the project sponsor.

The tractor's ESC was the original Bendix® system on the Volvo® tractor. The trailer RSC systems were also provided by Bendix® and were installed prior to testing. Without access to the tractor ESC and trailer RSC control algorithms, details on how the stability systems operate were estimated by observed trends in test data, documentation in literature, and discussions with the product supplier.

6.1 Test Maneuvers

Four maneuvers were used to evaluate trailer roll-stability: Robot J-turn, Driver J-turn, Double-Lane-Change (DLC), and 0.25-Hz Sine-with-Dwell (SWD). The DLC and Driver J-turn were driver-steered whereas the Robot J-turn and SWD were robot-steered. The entry speed for the DLC and both J-turns was incrementally increased until a rollover event occurred. The entry speed was fixed for the SWD maneuver, but the steering profile amplitude increased until trailer rollover. For all maneuvers, the driver put the tractor's transmission into neutral just before the starting gate. The J-turn maneuvers exhibit dynamics similar to an exit-ramp whereas the DLC and SWD maneuvers mimic obstacle avoidance events. The following sections illustrate the four maneuvers' steering profiles and vehicle's traveled path.

6.1.1 Driver J-turn

The Driver J-turn maneuver closely resembled an exit-ramp maneuver. The driver entered at a prescribed speed and steered the vehicle to follow outside a 150-ft constant radius curve as illustrated in Figure 6-1. The driver continuously adjusted the steering profile through the maneuver to maintain course while

accounting for understeer or oversteer resulting from different entry speeds and stability control interventions. Figure 5-24 shown previously illustrates the variability in the driver's steering profile for different test trials.

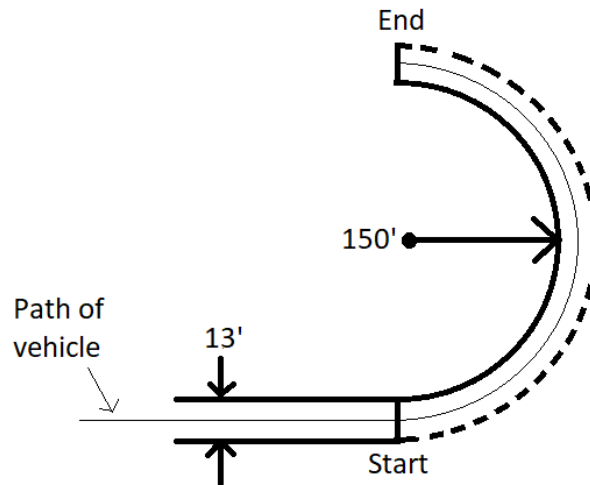


Figure 6-1. J-turn maneuver path layout.

6.1.2 Robot J-turn

The Robot J-turn is similar to the Driver J-turn but the steering profile is fixed for each trial. This maneuver is also sometimes called the “ramp-steer maneuver” because the steering profile begins at zero degrees, ramps-up to a set value at an approximately constant angular rate, then holds that end value through the maneuver. The ramp-up duration was 1.5 sec and the held steering wheel angle was about 215° resulting a little more than 10° steering angle at the axle. These values were chosen based on past studies [34,68]. Figure 6-2 presents the steering wheel profile. The Robot and Driver J-turn maneuvers differ in that the robot provides repeatability in the steering profile whereas the driver provides repeatability in the vehicle's trajectory. For higher entry speeds and unchanged steering angle, the vehicle tends to understeer, tracing a path of larger radius. At 32 mph, the robot J-turn's path is approximately the same as the driver J-turn.

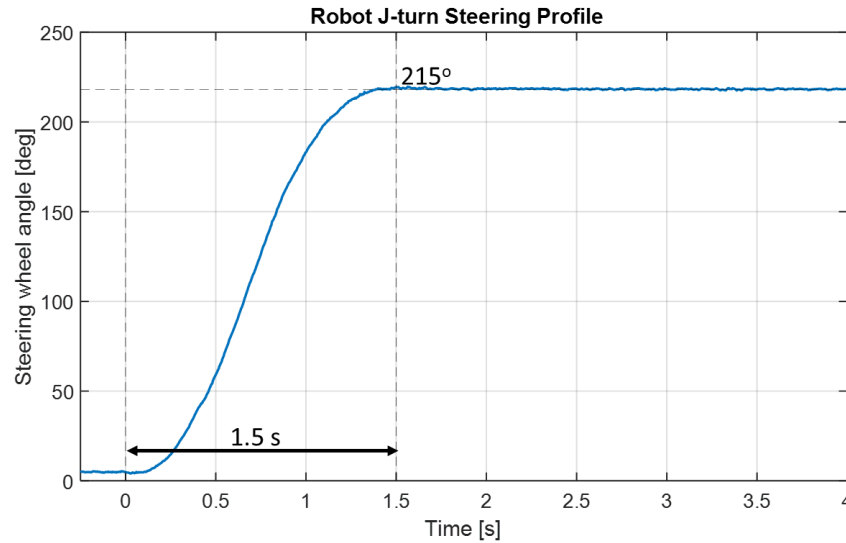


Figure 6-2. Robot J-turn steering profile.

6.1.3 Double Lane Change (DLC)

The Double Lane Change (DLC) maneuver mimics a driver changing lanes from the right to the left then immediately back to the right as if to avoid an obstacle. The lane changes were enforced by having the driver travel through a series of cone pairs mapped as shown in Figure 6-3. This path was adopted from NHTSA [68]. Figure 6-4 provides an example of the steering profile through the DLC for a range of speeds. Notice that the steering profile increases in both frequency and amplitude as the entry speed increases. This is because the driver must correct for understeer and ensure the vehicle travels through each set of cones. At low speeds, there is often a stutter in steering through the second turn (the trough). This stutter smooths out for higher entry speeds like that shown in Figure 6-4. There is significant variability in the start of steering by the driver relative to the starting gate and steering amplitudes during the first, second, and third turns. These small variations occur throughout the tests not only on different test days but also in back-to-back trials of the same entry speed.

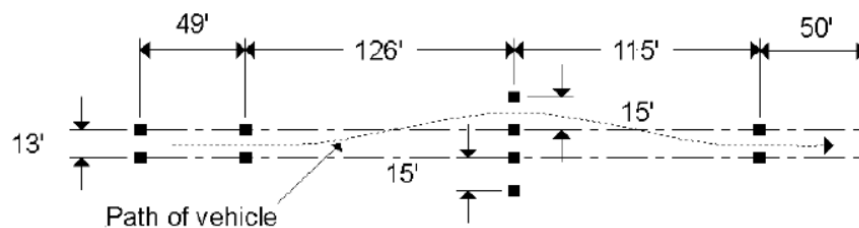


Figure 6-3. DLC path layout.

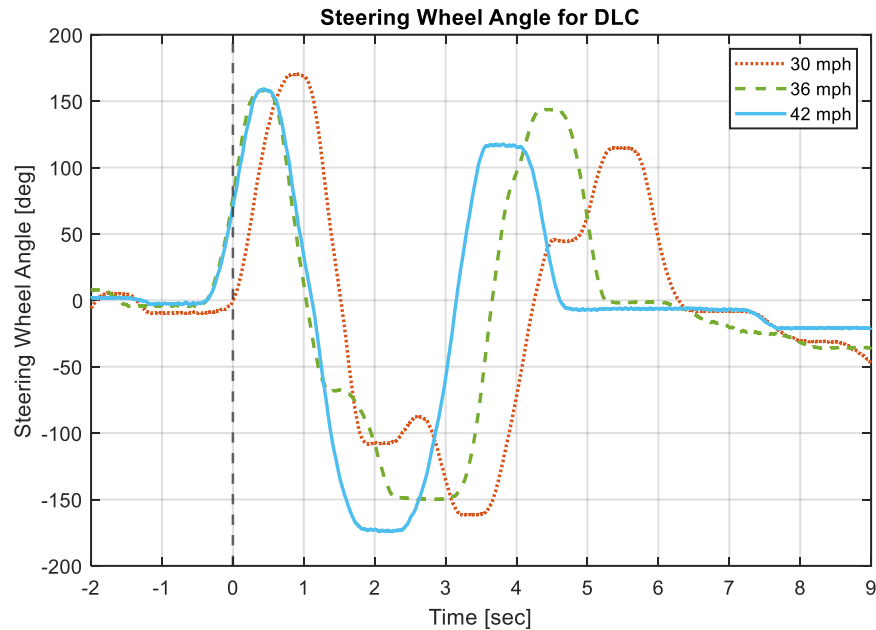


Figure 6-4. Steering wheel angle changes with speed for DLC maneuver. ESC and RSC off.

6.1.4 Sine-with-Dwell (SWD)

The 0.25-Hz Sine-with-Dwell (SWD) maneuver differs from the other three in how the rollover event is determined. Rather than increasing entry speed until rollover occurs, the entry speed was fixed at 45 mph and the steering amplitude was increased until rollover. The steering was robotically controlled for precise and accurate steering profiles. The steering profile follows a 0.25 Hz sine wave where the second peak is held (dwell) for two seconds before returning to center as shown in Figure 6-5. The steering amplitude is defined as percentages where 100% steering is considered as the sine peak reaching 250° at the steering wheel. This steering angle was determined in a previous study that conducted the slowly-increasing-steering (SIS) maneuver with the same test tractor and steering robot. The SIS maneuver was performed at a constant 30 mph and increased the steering wheel angle at a constant rate until the tractor's lateral acceleration reached 0.50g. The average steering angle from tests conducted in both left and right directions was found as the 250° [34,35]. The SWD maneuvers were tested first at 40% steering amplitude, then increased in 5% or 10% increments until rollover. Figure 6-6 plots the tractor's path at 40%, 50%, and 60% steering in stock condition.

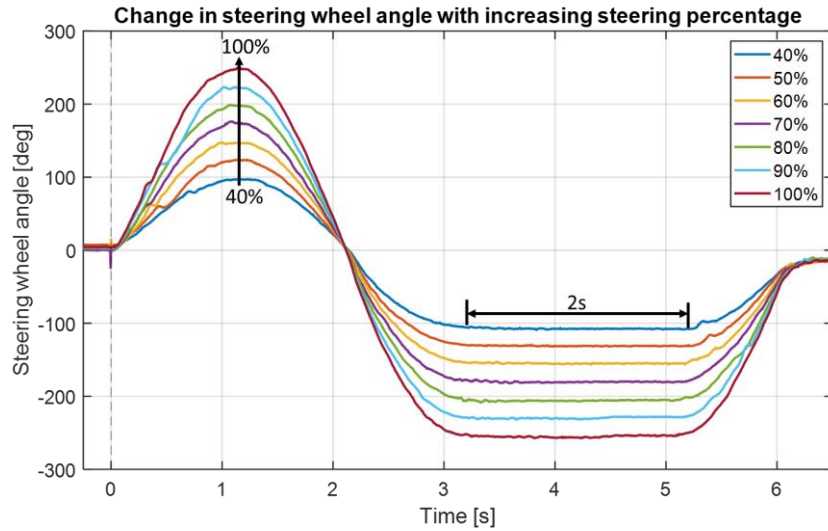


Figure 6-5. 0.25-Hz SWD maneuver steering wheel angle profile.

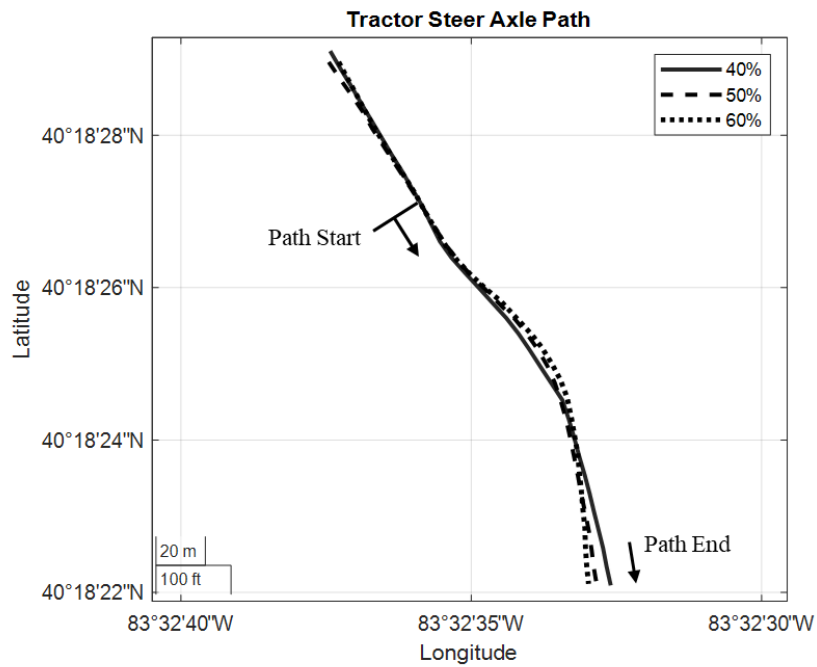


Figure 6-6. 0.25-Hz SWD maneuver path in stock condition (RSC and ESC off).

6.2 Determining Rollover Threshold from Test Data

Load cells installed within outrigger assemblies estimated contact forces of the outrigger wheel and ground during a rollover event. The load cells measure these forces indirectly through the hinge linkage described in Chapter 5. Mechanical compliance of the outrigger's long outboard arm and inboard hinge assembly connecting the load cell and outrigger wheel effectively acts as a large spring and friction damper. In

theory, energy absorbed and redirected by the outrigger assembly during a rollover event could be estimated from the load cell measurements and outrigger mechanical stiffness and damping. The calculated energy could then estimate the trailer's roll angular momentum to quantify likelihood of total rollover. Unfortunately, this formulation requires a highly accurate model of the outrigger assembly mechanical stiffness & damping and, more importantly, highly accurate force measurements from the load cells. The load cell force rating was based on our experience with maximum expected impact loads, derived from both simulations and prior testing. During most tests, the rollover impact loads were substantially smaller than the maximum rating, hence resulting in relatively small voltage output. These small voltage signals were subject to distortion from electrical noise, reducing measurement accuracy. Ultimately, the combination of noisy signals and uncertain mechanical properties of the outrigger assemblies prevented accurately quantifying rollover likelihood through a physics-based model. Instead, the contact force estimates provided a comparative metric for rollover severity. This required a more subjective approach based upon test video data and trackside observations corresponding to the measured data.

After reviewing numerous test videos, the team decided that an outrigger wheel contact force of 400 lbf or more was considered as a hard outrigger contact and indicated a high likelihood of complete trailer rollover. If outrigger contact force was greater than 1000 lbf, trailer rollover was likely to be guaranteed for those test conditions. For outrigger contacts less than 400 lbf, the trailer was not likely to rollover, but wheel lift occurred. These contact force thresholds for likelihood of rollover are inherently subjective; however, the significant number of data points for each maneuver and consistent behavior for similarly parameterized trials provides confidence in this analysis. For each maneuver and stability control implementation, the first trial's entry speed or steering percentage was predetermined to not induce rollover. Each successive trial then had a target entry speed or steering percentage that was greater than the previous by 1-2 mph or 5%-10% steering until outrigger contact occurred. If the outrigger contact appeared to be hard from trackside observations and a live-reading of contact force measurement, the entry speed or steering percentage was then reduced by 1 mph or 5% and the maneuver repeated. Without high-resolution and accurate real-time observations of outrigger contact forces and exact entry speeds during the tests, there was some variation in the actual entry speed with respect to the target entry speed. To account for these variations, a scheme was developed to best estimate the rollover entry speed and steering percentage from test data in post-processing and appropriate filtering of signals. This scheme, presented as Table 6-1, becomes especially important where there is limited data for a critical entry speed. Some maneuvers can induce multiple outrigger contacts within a single trial. The rollover threshold is determined by the highest contact force unless the first is lowest but still above the rollover threshold contact force (≥ 400 lbf).

Table 6-1. Decision matrix for determining rollover threshold.

Number of trials at specified entry speed	Trials without outrigger contacts	Trials with at least one outrigger contact			Rollover threshold?
		< 400 lbf	≥ 400 lbf & < 1000 lbf	> 1000 lbf	
0*	-	-	-	-	Maybe**
1	0	1	0	0	No
1	0	0	1	0	Maybe**
1	0	0	0	1	Yes
2	1	1	0	0	No
2	1	0	1	0	Maybe**
2	1	0	0	1	Yes
2	0	2	0	0	No
2	0	1	1	0	Maybe**
2	0	0	2	0	Yes
2	0	0	1	1	Yes
2	0	0	0	2	Yes
3	2	1	0	0	No
3	2	0	1	0	No
3	2	0	0	1	No
3	1	2	0	0	No
3	1	1	1	0	No
3	1	1	0	1	No
3	1	0	2	0	Yes
3	1	0	1	1	Yes
3	1	0	0	2	Yes
3	0	3	0	0	No
3	0	2	1	0	No
3	0	2	0	1	No
3	0	1	2	0	Yes

3	0	1	1	1	Yes
3	0	1	0	2	Yes
3	0	0	3	0	Yes
3	0	0	2	1	Yes
3	0	0	1	2	Yes
3	0	0	0	3	Yes

* In cases where there was no test performed at a particular speed, the outrigger contacts at the speeds 1 mph below and 1 mph above were used to determine the rollover speed.

** In cases where the rollover threshold was determined as “Maybe”, the outrigger contacts at the speeds 1 mph below and 1 mph above were used to determine if the speed is or is not the rollover speed.

6.3 Test Environment & Test Sessions

Tests were conducted on the Vehicle Dynamics Area (VDA) at the Transportation Research Center (TRC) in East Liberty, Ohio shown in Figure 6-7. The North Loop was used throughout testing to prevent significant brake fade (when ESC and/or RSC were active) and provide additional space for the vehicle to get up-to-speed for each test trial. The asphalt was dry in all test conditions and had a coefficient of friction of about 0.8 [98]. Between test days, the test trailers were parked at a holding area within TRC near the VDA. To protect the test tractor and its external sensors from rain, it was kept at a Link Engineering™ facility a couple miles from the VDA and outside TRC. This reduced the number of sensors that needed to be removed between test days and sessions. As with all field tests, time spent on the test track collecting data was contingent upon weather, track availability, and unforeseen delays such as flat tires caused by debris on the test track. Most test days were split into two parts with a brief lunch break between the two. During each lunch break, data collected in the morning was reviewed and verified before proceeding with the afternoon’s scheduled tests. All mechanical and electrical equipment was visually inspected after each session.

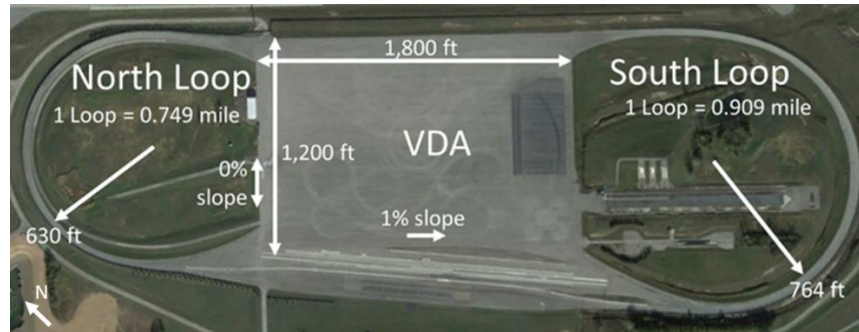


Figure 6-7. Aerial view of TRC Vehicle Dynamics Area (Maxar Technologies, State of Ohio, 2022).

The entire testing program for the project sponsor was divided into five sessions over three months. Three of five test sessions are related to this dissertation and the other two sessions included additional trailer types, configurations, and loading conditions not discussed here. The first test session included one day of preparing mechanical equipment and another day of testing to validate instrumentation after the equipment had moved from Virginia Tech to East Liberty, OH. The maneuver paths were laid out on the test track using weather-resistant, reflective highway tape. This ensured that the same maneuver performed on different test days would remain unchanged. Because the shakedown tests conducted at Virginia Tech were limited in speed and space, the steering robot PID gains could not be safely tuned until tests were conducted at speed on the VDA at TRC. A failed solder on the steering robot prevented complete tuning for the SWD maneuver at TRC during the first test session. Successful baseline tests – stock (ESC and RSC off) – were completed for the Driver J-turn and DLC maneuvers. Table 6-2 lists the test date, maneuver, configuration, and number of trials for the three relevant test sessions of roll stability testing with the heavily-loaded double straight-rail trailers.

The second test session included robot tuning for the SWD maneuver. By the end of the first and second sessions, all four maneuvers had been tested in stock, RSC, and RSC+ESC configurations. The final session concluded testing at TRC with all four maneuvers in ESC only configuration. Trailer brake tests described later in Chapter 7 were conducted before the ESC only tests. These are not shown in the table because they are unrelated to the analysis in this chapter. A total of 165 tests were conducted to tune the steering robot, stage the maneuvers, and test all maneuvers to compare the different stability control systems for the heavily-loaded double 33-ft straight rail trailers in the four unique maneuvers.

Table 6-2. Summary of test sessions for evaluating stability control systems.

Session	Maneuver	RSC	ESC	# of Tests
1	Driver J-turn	Off	Off	8
	Robot J-turn (tuning)*	Off	Off	11
	45 mph – 0.25 Hz SWD*	Off	Off	5
	DLC	Off	Off	13
	45 mph – 0.25 Hz SWD*	Off	Off	6
	Robot J-turn*	Off	Off	5
2	Driver J-turn	On	Off	7
	Robot J-turn	On	Off	9
	Robot J-turn	Off	Off	4
	SWD (robot tuning)*	Off	Off	17
	45-mph 0.25-Hz SWD	Off	Off	5
	Driver J-turn	On	Off	6
	Driver J-turn	On	On	8
	Robot J-turn	On	On	6
	Robot J-turn	On	Off	5
	45-mph 0.25-Hz SWD	On	Off	6
	45-mph 0.25-Hz SWD	On	On	6
	DLC	On	On	7
	DLC	On	Off	6
	Robot J-turn	Off	On	6
3	DLC	Off	On	7
	45-mph 0.25-Hz SWD	Off	On	6
	Driver J-turn	Off	On	6

* Data not used for analysis; SWD = sine with dwell; DLC = double lane change

Testing was only conducted when the asphalt was dry. Because the first and last test day were about 7 weeks apart, ambient temperatures covered a wide range. The hottest test day for the dates shown in Table 6-2 was in the mid-80s (°F), and the coldest day was in the low-60s (°F). Within this range, temperature variations are not expected to significantly influence the results and conclusions.

6.4 Tractor ESC and Trailer RSC Operation

As discussed in Chapter 2, tractor ESC and trailer RSC have similar but different control objectives. Tractor ESC prioritizes tractor yaw stability according to the vehicle speed, yaw rate, lateral acceleration, and driver inputs (throttle, steering angle, steering rate, etc.). Tractor ESC applies and modulates brake pressure to individual wheels via the ABS control valve, slowing the vehicle and producing yaw moments to redirect the tractor's yaw motion in favor of the driver's inputs. For example, if the tractor begins to oversteer, ESC might brake the outside drive wheel to nudge the tractor into a more neutral steering orientation. ESC also reduces engine throttle to help slow the vehicle. Within each maneuver in this study,

the driver did not provide any throttle input and the transmission was in neutral, so ESC intervention only modulated brake pressures. Figure 6-8 plots the tractor drive axle brake pressures and ESC activation relay for a DLC maneuver. At the beginning of the maneuver, as the vehicle was steered left ESC braked the right drive axle wheels, then as the vehicle was steered right ESC braked the left drive axle wheels. ESC may have been modulating the tractor steer axle wheels, but they were not measured due to limited available channels in the data acquisition system. It was decided to measure the drive axle brake pressures over the steer axle because the drive axles provide significantly more braking force to the vehicle.

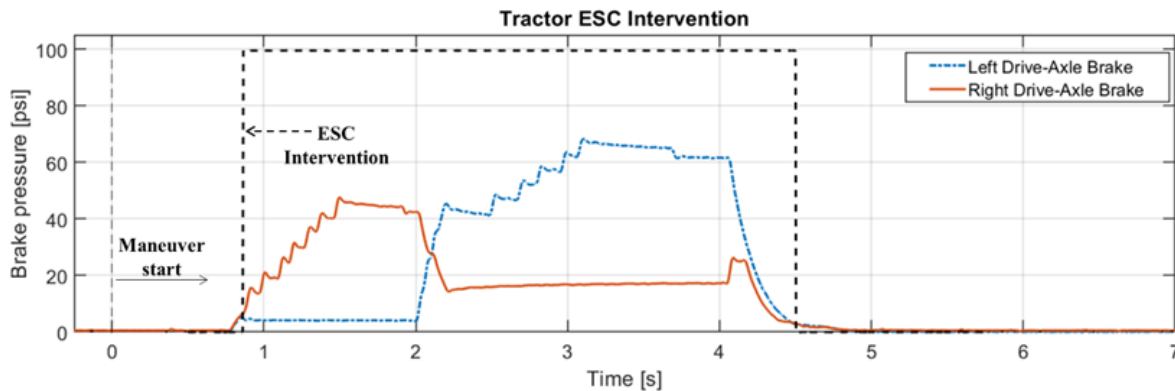


Figure 6-8. Example tractor ESC intervention and resultant brake pressures during a DLC maneuver.

Trailer RSC's primary control objective is to prevent trailer rollover. There is no communication between control systems on different vehicle units – ESC to RSC, front to rear trailers, etc. Lateral acceleration threshold and rate of change are the primary control triggers and are specific to the vehicle unit equipped with the RSC module. These thresholds are adjusted as the RSC module estimates the corresponding trailer roll stability according to an estimated mass from air spring pressure and reaction to small brake pressure pulses during moderately fast maneuvers. Figure 6-9 plots the brake pressure time trace during a DLC maneuver for the front and rear trailers with RSC interventions. Notice that brake pressure amplitude is highest for the wheel on the outside of the turn. At the beginning of the maneuver, the vehicle steers left, causing lateral load transfer onto the right wheel. Because the right wheel supports greater vertical load at that time, it can accept greater brake pressure before wheel lockup. Some brake pressure is applied to the inside wheel to help slow the vehicle and allow the RSC module to determine wheel lift according to the lightly braked wheel's speed response. When comparing ESC intervention in Figure 6-8 and front and rear RSC intervention in Figure 6-9 it is easy to see that the stability control systems activate in order of the corresponding vehicle's position within the maneuver. Because the tractor is first to enter the maneuver, it is first to experience significant steering input, yaw motion, and lateral acceleration, triggering ESC. Then the front trailer begins to roll as it turns left, triggering its RSC. Lastly, the rear trailer follows suit

and RSC intervenes when lateral accelerations are high. As RSC on both trailers slow the vehicle through the maneuver, RSC intervention becomes less aggressive and the resulting brake pressures are less at the end of the maneuver than the beginning, especially for the rear trailer. In Figure 6-9 there are some occasions of small brake pressure outside the RSC activation relay's time window. These events occur when the RSC system is testing for wheel lift. For these instances, the controller typically did not activate the relay to the data acquisition system.

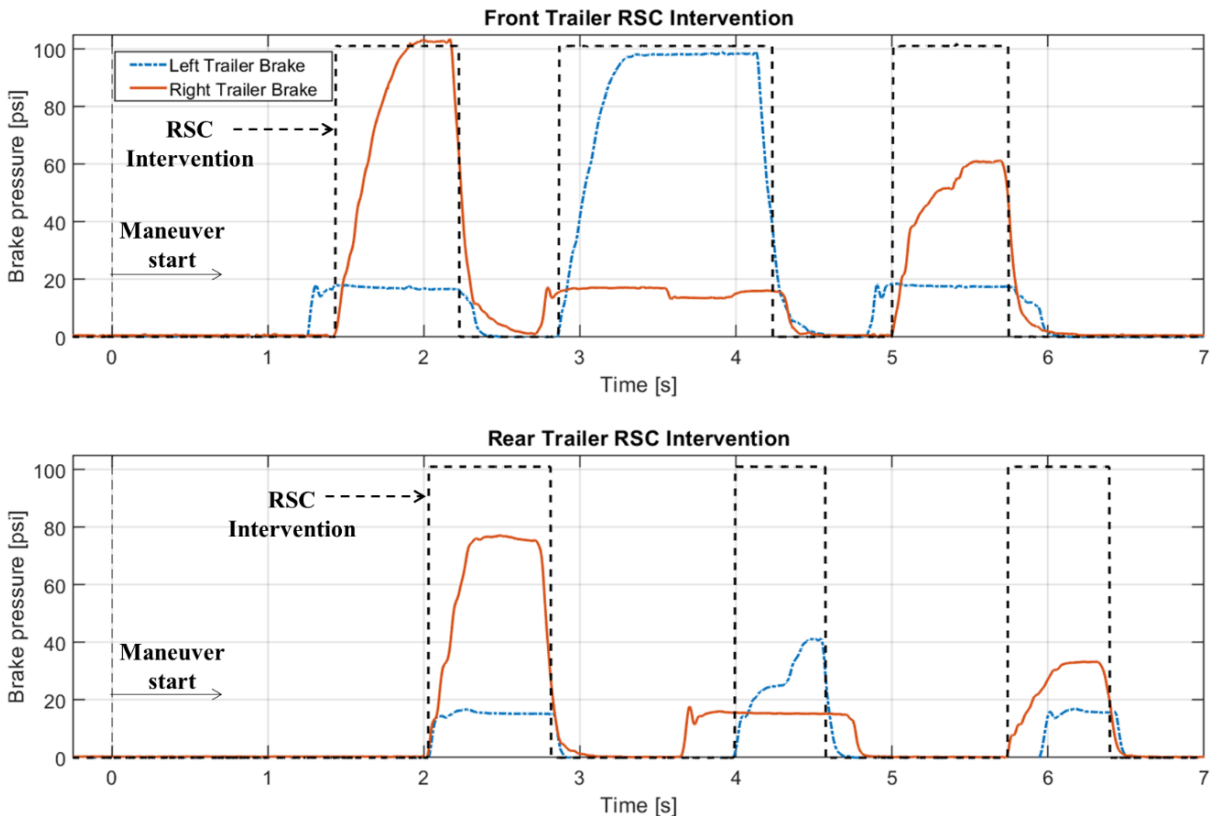


Figure 6-9. Example front and rear trailer RSC interventions and resultant brake pressures during a DLC maneuver.

From Table 5-1 shown earlier, the heavily-loaded 33-ft straight-rail trailer's track width and CG height from the ground are 78 and 77.8 inches, respectively. From Equation (2-6), the SSF is then calculated to be 0.50g. The lateral acceleration just before the first outrigger contact event of a trial is indicative of the trailer's rollover lateral acceleration. The average and median rollover lateral accelerations from all such trials were 0.51g and 0.50g, respectively. These values closely match the SSF's estimates. It is important to understand that these values are inherent to the vehicle unit and are not affected by the stability control systems. Instead, the stability control systems reduce the trailer's rollover propensity by slowing the vehicle during the maneuver before the lateral acceleration becomes too great.

6.5 Results & Analysis

6.5.1 Driver J-turn

Figure 6-10 presents the outrigger wheel contact force with respect to each tested entry speed for the Driver J-turn maneuver. The stock condition had outrigger contacts at 33 mph and 34 mph that just barely surpassed the 400 lbf threshold at both speeds. According to Table 6-1, the rollover threshold is 33 mph. Although the impact force was relatively low, the impacts were clearly strong enough to dramatically change the trailer's lateral acceleration as it leaned onto the outrigger as shown in Figure 6-11. The maneuver start in Figure 6-11 is at 0 sec and corresponds to when the driver typically initiated steering about 0.5 sec before passing the spatial trigger. The RSC only condition had outrigger contacts for speeds at 34 mph and higher. From Table 6-1, the RSC rollover threshold is 35 mph. ESC performed slightly better than RSC, with a rollover threshold of 36 mph. Furthermore, having both RSC and ESC provided the best roll-stability with a rollover threshold at 37 mph. Although this speed was not tested, it is easy to see a trend in the outrigger contact force data from 34 to 36 mph. There were contacts at 36 mph, but not so great in force to determine that as the rollover speed according to Table 6-1, thus it was suspected that the rollover speed is 37 mph. The rollover speeds are plotted in the Section 6.6 along with the other maneuvers.

Figure 6-12 presents the speed drop through the maneuver for each trial and stability control configuration. The speed drop was determined as the entry speed (0.5 second before passing starting gate for driver-steered maneuvers) minus the speed just before the shortest trial was complete 11 seconds later for the Driver J-turn. The speed drop for the stock configuration (no stability enhancement) is much less than any other configuration because no vehicle unit applied brake pressure. ESC and RSC+ESC configurations resulted in about the same speed drop and were higher than RSC by itself. However, RSC+ESC exhibited a higher rollover threshold because there was greater deceleration to slow the vehicle before a trailer could experience a larger lateral acceleration. This is discussed further in Section 6.6 summarizing all maneuvers.

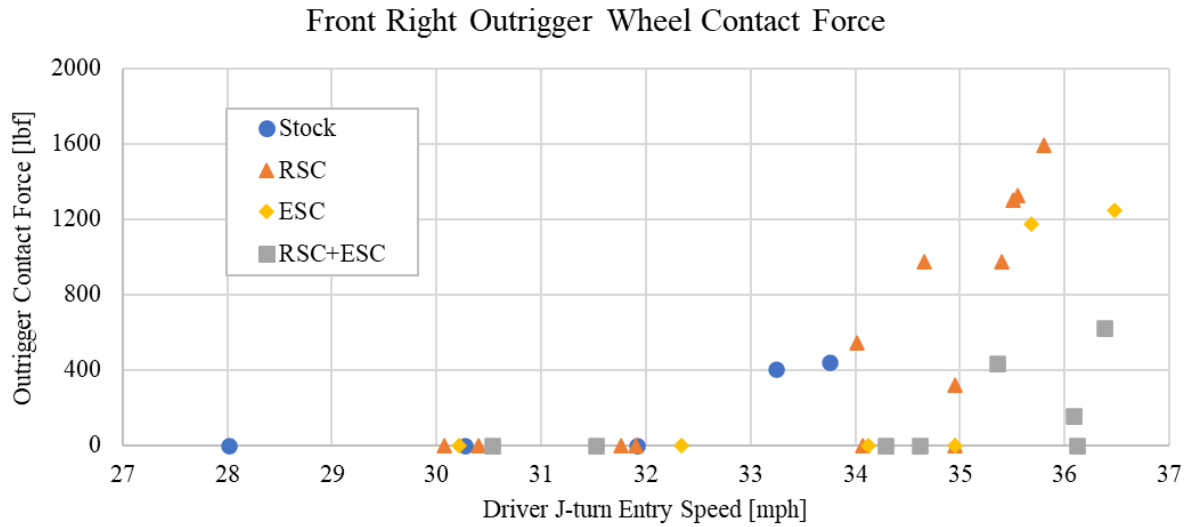


Figure 6-10. Front trailer right side outrigger wheel contact force for the driver J-turn maneuver.

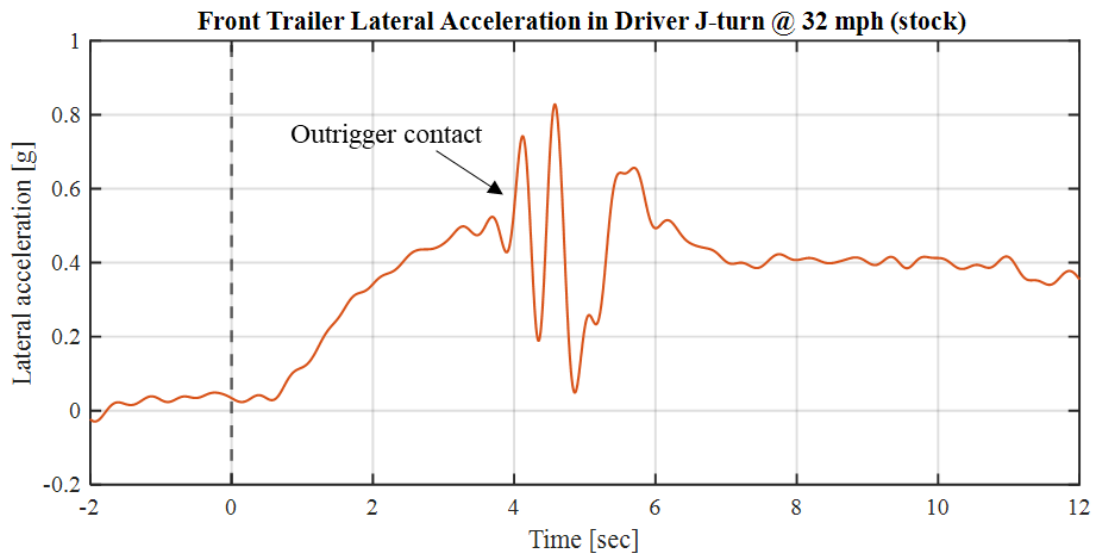


Figure 6-11. Front trailer lateral acceleration during driver J-turn maneuver in stock condition.

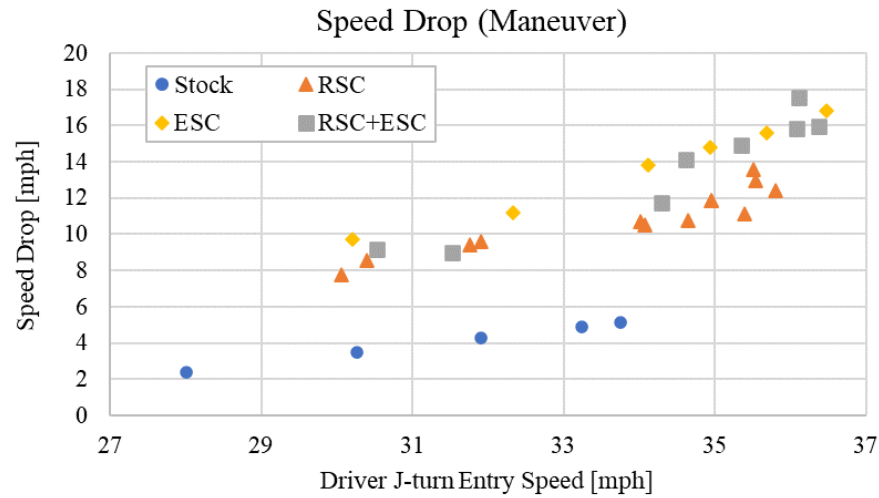


Figure 6-12. Speed reduction during driver J-turn maneuver for each stability control system implementation.

Figure 6-13 presents the maximum tractor-trailer and trailer-trailer articulation angles during the driver J-turn maneuver. Although the data spread is not clear-cut, two trends stood out: (1) ESC intervention tended to increase articulation angle between the tractor and front trailer, and (2) stock configuration resulted in noticeably larger articulation angles between the front and rear trailer than any stability control system. With respect to ESC, other studies on long combination vehicles have observed similar results in that ESC activation can induce larger articulation angles and increase risk of jackknifing [34]. However, ESC's effect on yaw dynamics in this testing do not appear as severe as other studies. ESC's effect on articulation angles can be influenced by driver behavior, vehicle loading condition, road surface friction, among many other factors. The focus of this dissertation is on the effects of stability control systems on trailer roll-dynamics, and the comments here on yaw-dynamics are for completeness. It is recommended that future studies conduct a parametric analysis on ESC tuning effects on long combination vehicle yaw dynamics for different loading conditions, etc., especially on a low-friction surface (wet or snowy road) where a vehicle has substantially less stable yaw dynamics than a high-friction surface (warm, dry asphalt). Figure 6-14 provides an example of articulation angles through the Driver J-turn at 34 mph entry speed for each tested stability control implementation. Only the stock condition experienced rollover at this entry speed. Articulation between the two trailers is typically higher than between the tractor and front trailer because (1) the dolly connecting the trailers adds to the measured articulation angle and (2) the front trailer's hitch path is at a tighter radius than the tractor's hitch path, resulting in larger inboard off-tracking at these speeds as detailed in Chapters 3 and 4; thus, a greater articulation between vehicle units.

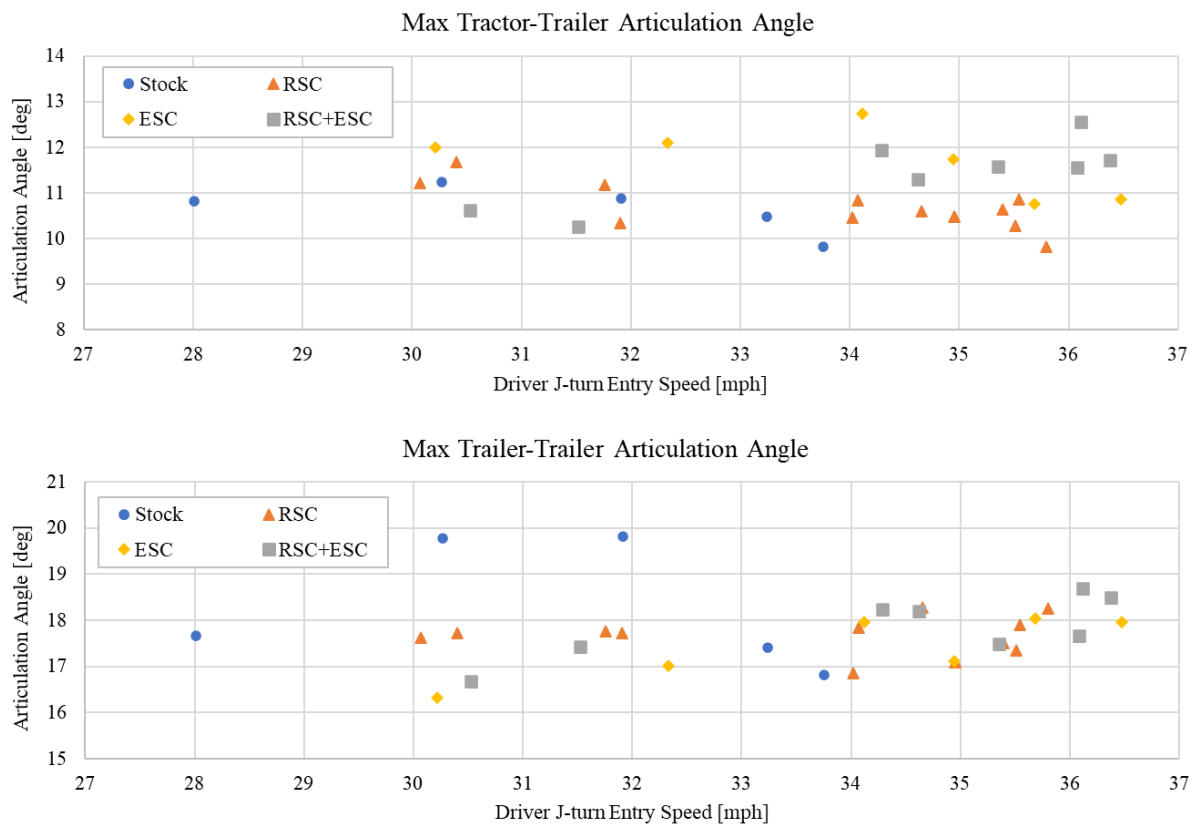


Figure 6-13. Peak articulation angles during driver J-turn maneuver.

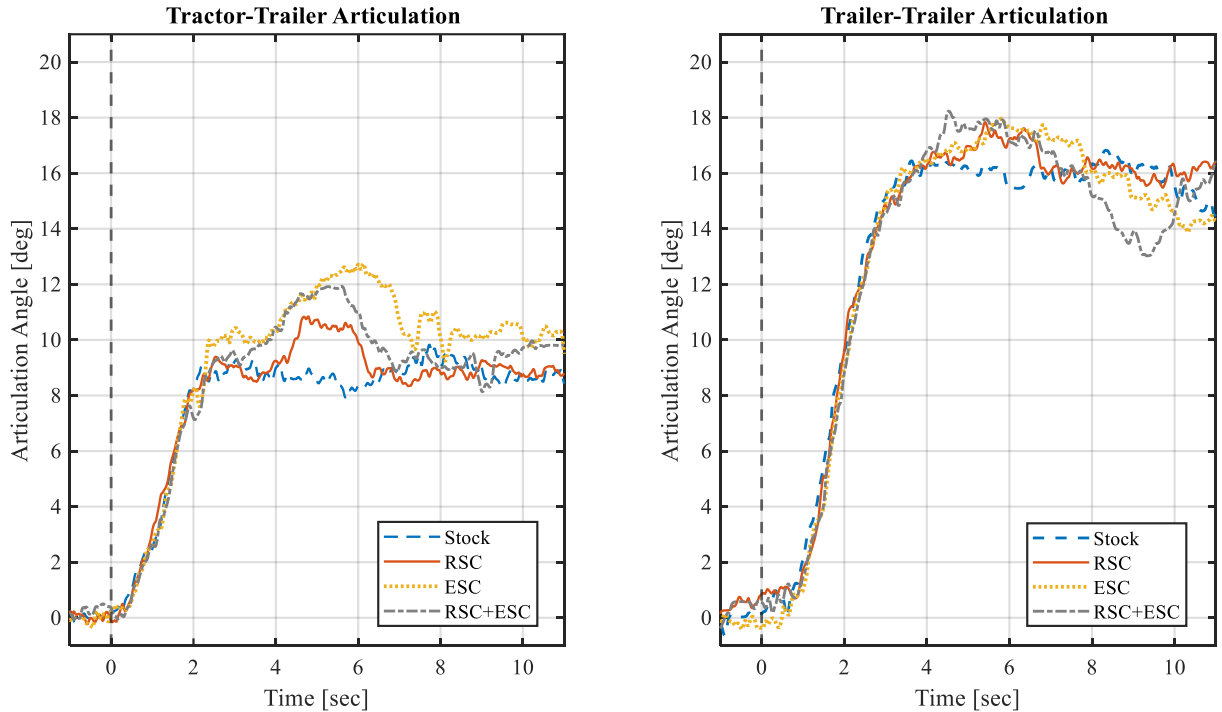


Figure 6-14. Example comparison of articulation angle during driver J-turn maneuver for different stability control system implementations at same entry speed (34 mph).

6.5.2 Robot J-turn

Figure 6-15 presents the outrigger wheel contact forces of the Robot J-turn tests. Outrigger contacts only occurred on the right side of the front trailer. As expected, the stock condition resulted in the lowest rollover threshold speed at 33 mph. Next, a clear trend appeared for the RSC only configuration where outrigger wheel contact force increased approximately linearly with speed from 32 mph to 35 mph. According to Table 6-1, the rollover threshold for RSC only is 34 mph. The ESC only condition didn't have outrigger wheel contact until about 36 mph. This force was high enough to identify the rollover threshold as 36 mph. At 37 mph, the outrigger wheel contact force was very high (>1000 lbf). At first glance, the RSC+ESC results are less clear than other configurations. For all tested speeds, three trials had outrigger contact and only two of which were large enough for the instrumentation to detect. One trial at about 36 mph resulted in an outrigger wheel contact force of about 170 lbf – not large enough to identify the speed as the rollover threshold. Three trials were conducted around 35 mph, where one test trial resulted an outrigger wheel contact force >1000 lbf, one without contact, and another with immeasurably small contact force. From Table 6-1, it is concluded that the rollover speed is greater than 35 mph. The >1000 lbf outrigger force occurred for the one trial because ESC intervention was abnormally brief for that trial despite having similar entry speeds and identical steering profiles. This was the only trial from all tests

where ESC seemed to not perform correctly. The cause for this discrepancy is unknown. This abnormality is highlighted by Table 6-3 for the three trials at 35 mph. Notice that ESC activated around the same time for all trials, but for the trial with hard outrigger contact ESC was only active about 1/10th the duration as the other trials. Because this maneuver is robot-steered, there was not significant difference in the steering profile. RSC intervention was not substantially different among the three trials in Table 6-3.

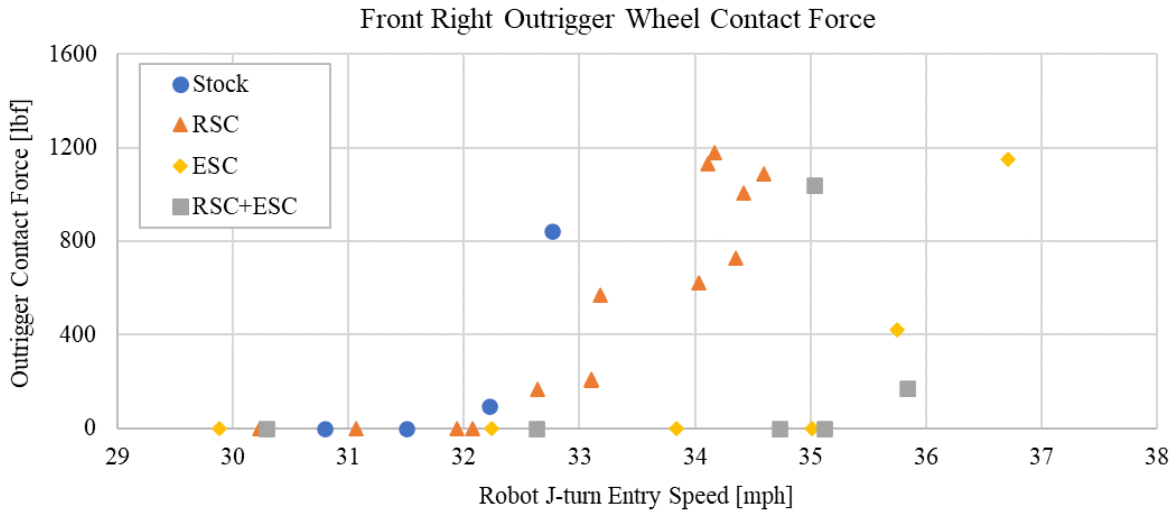


Figure 6-15. Front trailer right side outrigger wheel contact force for the robot J-turn maneuver.

Table 6-3. ESC activation analysis for RSC+ESC configuration during robot J-turn at 35 mph entry speed.

	Entry Speed	ESC Start Time After Entry	ESC Duration	Speed Drop During Any ESC/RSC Intervention	Outrigger Contact Force
Trial 1	35.03 mph	1.128 sec	0.688 sec	10.41 mph	>1000 lbf
Trial 2	34.73 mph	1.196 sec	6.344 sec	15.04 mph	No contact
Trial 3	35.11 mph	1.116 sec	6.844 sec	15.87 mph	Immeasurably small (kiss)

Figure 6-16 presents the speed drop throughout the entire maneuver with start time when passing the spatial trigger and the robot begins to steer to an end time 10.5 seconds later. As expected, the stock condition had the lowest speed reduction and RSC had second lowest. Surprisingly, RSC+ESC resulted in slightly less speed drop than ESC alone. Although the speed drop was less for RSC+ESC than ESC, ESC was less active when RSC was included as illustrated in Figure 6-17. Despite this, a higher rollover threshold was achieved with RSC+ESC than ESC alone because RSC+ESC provided stronger

deceleration, reducing the vehicle’s speed quicker early on and reducing the front trailer’s peak lateral acceleration for the same entry speed as ESC only. This is further discussed in Section 6.6.

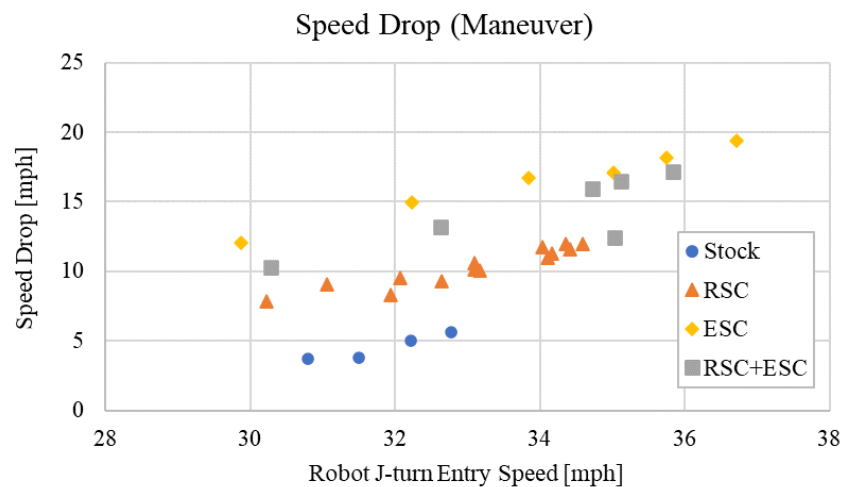


Figure 6-16. Speed reduction during robot J-turn maneuver for each stability control system implementation.

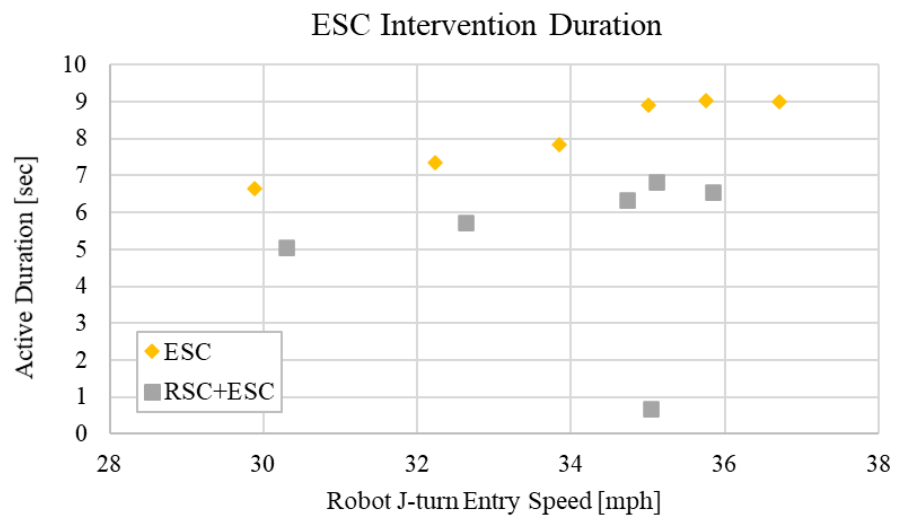


Figure 6-17. ESC intervention duration during the robot J-turn maneuver for ESC only and RSC+ESC configurations.

Figure 6-18 shows the peak articulation angle during the robot J-turn maneuver for all test trials. Just as with the driver J-turn, the stock condition had the lowest articulation angles and ESC intervention resulted in the highest articulation angles between the tractor and trailer. Figure 6-19 presents the articulation angle for the different stability control configurations for test trials of most similar speed ranging between 32

and 33 mph. For both tractor-trailer and trailer-trailer articulation, the stock conditions had less articulation than when stability control systems were present.

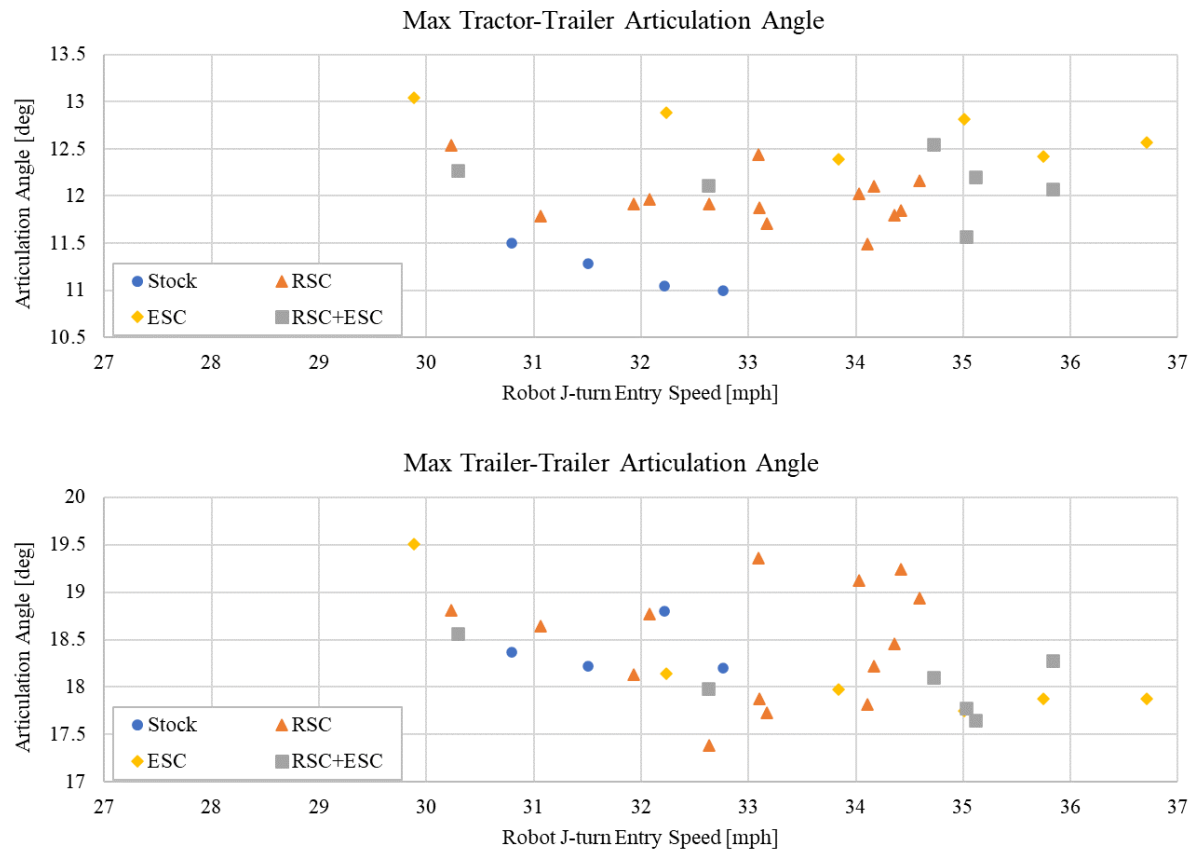


Figure 6-18 . Peak articulation angles during robot J-turn maneuver.

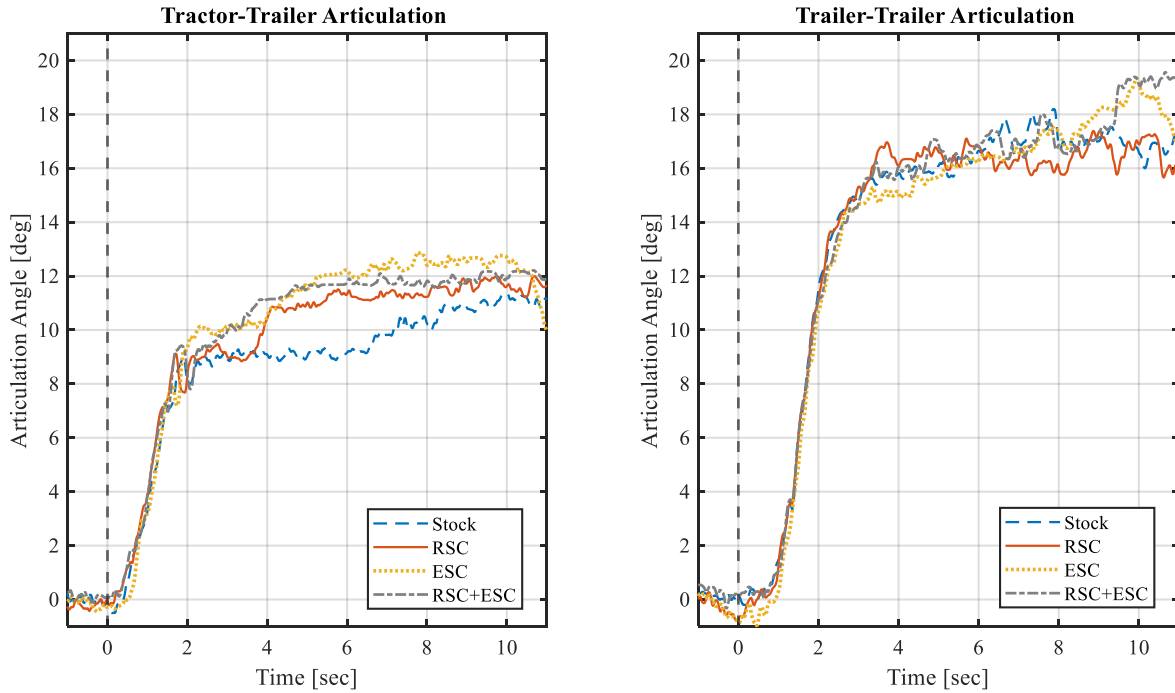


Figure 6-19. Example comparison of articulation angle during robot J-turn maneuver for different stability control system implementations at similar speeds (32 – 33 mph).

The general trend for all data shown in Figure 6-18 is that as vehicle speed increases, articulation angles decrease. This is caused by increased tire slip on the steer axle resulting a larger path radius for constant steer angle as well as increased tire slip on the trailers in response to the centrifugal forces of the trailers negotiating the curve. Figure 6-20 provides an example data set for the RSC+ESC condition where increasing speed clearly resulted in an increased path radius. In the driver J-turn, the driver could alter the steering profile to accommodate increased steer axle tire slip and maintain the desired path of constant radius.

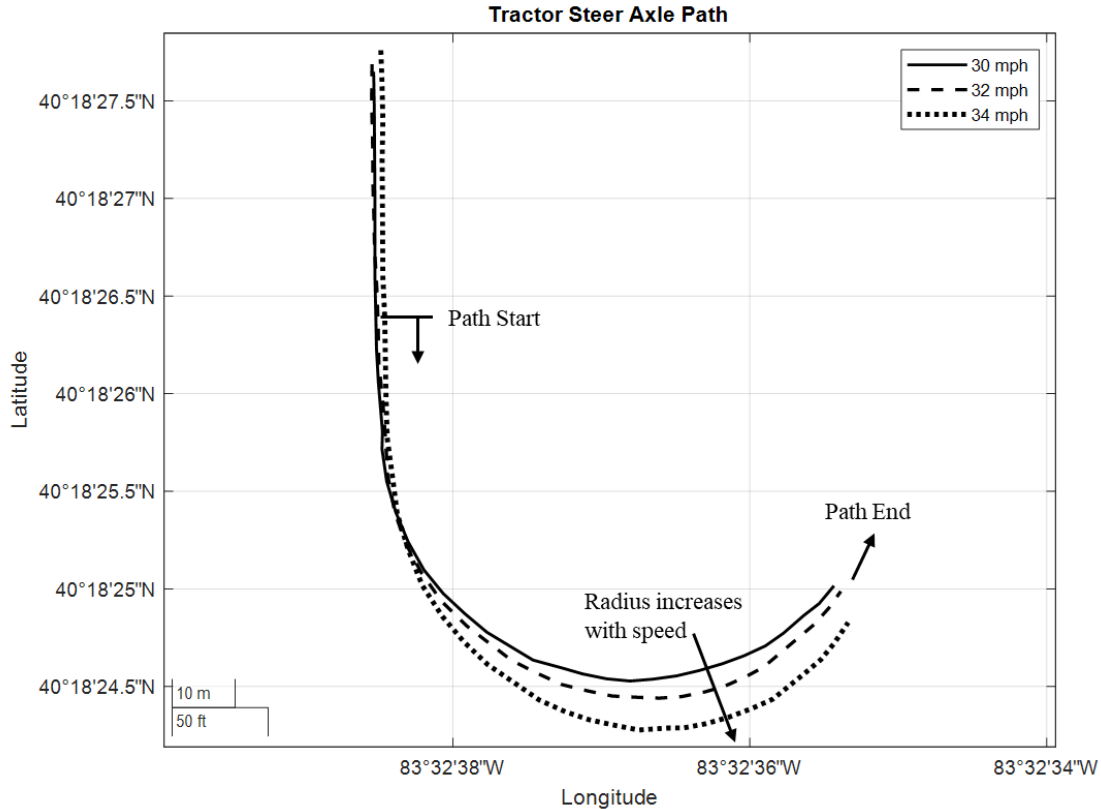


Figure 6-20. Tractor steer axle path through the robot J-turn maneuver with RSC+ESC.

6.5.3 Double-lane Change

Table 6-4 presents the peak outrigger contact forces for the DLC maneuver. Unfortunately, the GPS receiver unknowingly had been disconnected just prior to the stock tests with no RSC. However, the entry speed could be estimated from the driver's phone displaying its GPS-measured speed. This is not as accurate as the standalone GPS receiver because (1) the time delay is not known, (2) the phone only displayed integers, and (3) the tractor cab and glass impair measurement accuracy; however, it is sufficient for this study. It is easy to see that the rollover speed was 42 mph for the stock condition, as it was the only tested entry speed with consistent outrigger contacts. The rear trailer left outrigger had a hard contact in one of the three trials at 40 mph, but the other two trials did not have any contact at that speed. For the RSC only condition, there were no outrigger contacts at 45 mph and very hard contacts at 47 mph. Although some trials were targeting 46 mph, none were measured to be at 46 mph. One trial was at 46.6 mph (rounds to 47 mph) and resulted in a rear trailer left outrigger wheel contact force >1000 lbf. Comparing this data with no contacts at 45 mph suggests that the rollover threshold for RSC only was 46

mph. For ESC only, there were no outrigger contacts at 46 mph and consistently large contact forces at 47 mph. The ESC rollover speed was 47 mph.

Surprisingly, from Table 6-1 and Table 6-4, the RSC+ESC rollover speed was 46 mph, 1 mph lower than ESC by itself. At 45.7 mph, there were no contacts, but at 46.4 mph, there was a very large outrigger contact force (>1000 lbf) on the rear trailer left side for the RSC+ESC condition. The DLC maneuver inherently has the most variability in driver behavior that may impact the results. Had the driver's steering behavior been more consistent, it is suspected that the RSC+ESC condition would have equal or greater rollover speed than ESC alone. This claim is supported by results of the J-turn maneuvers discussed previously as well as the speed and lateral acceleration reduction through the DLC as discussed in the following.

Table 6-4. Peak outrigger contact forces for DLC maneuver.

Stability Control	Entry Speed [mph]	Peak Outrigger Contact Force [lbf]			
		Front Left	Front Right	Rear Left	Rear Right
Stock (none)	41	0	0	0	0
	42*	821	0	1085	2109
	42*	1360	0	1792	2269
	42*	1018	0	1208	0
RSC	45	0	0	0	0
	46*^	-	-	-	-
	47	640	0	1825	K
	47	406	0	1635	0
ESC	46	0	0	0	0
	47*	155	0	854	0
	47*	146	0	1068	K
	48	155	0	1562	361
RSC+ESC	46*	0	0	0	0
	46*	0	0	1537	K
	47	0	0	K	0
	47	1116	0	2374	144
<i>yellow</i> = below threshold (<400 lbf contact force) <i>red</i> = above threshold (≥400 lbf contact force) K = kiss: contact occurred but force was too small to be detected by instrumentation * = rollover speed ^ = no test performed					

The driver’s steering behavior changed for each stability control system, especially through the first and third turn as illustrated in Figure 6-22. These differences, however, cannot be solely attributed to the stability systems themselves. Considering the stability systems were tested on different days and considering the substantial number of DLC test trials with other trailer and loading configurations between the stability control system tests with the heavy straight-rails, it is wrong to suggest that the driver’s changing steering behavior as only response to the stability control systems. Another observation was that peak steering angles tended to reduce in turn 3 as the entry speed increased. Figure 6-21 provides example steering profiles for each stability control configuration at approximately the same entry speed of 40 mph.

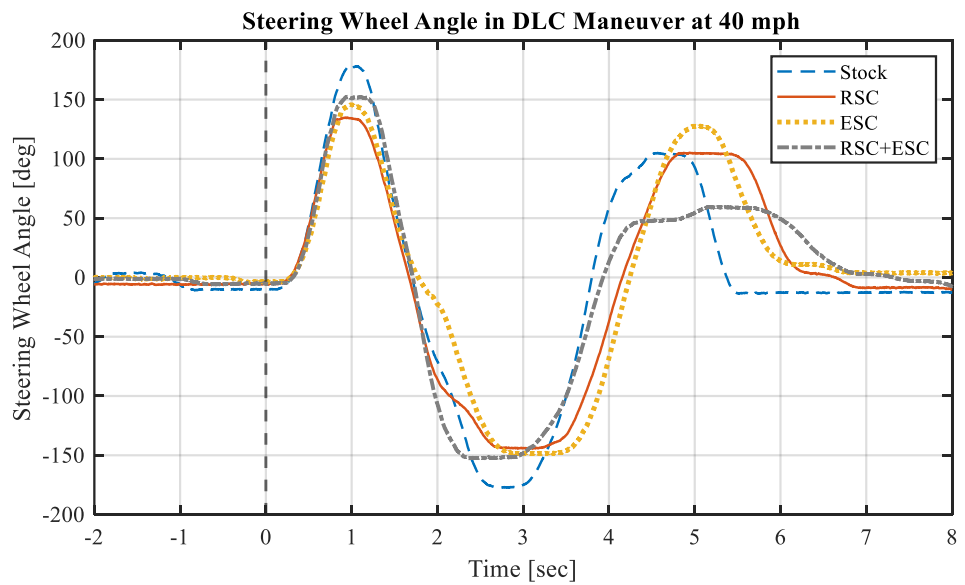


Figure 6-21. Steering wheel angle examples in DLC maneuver at 40 mph.

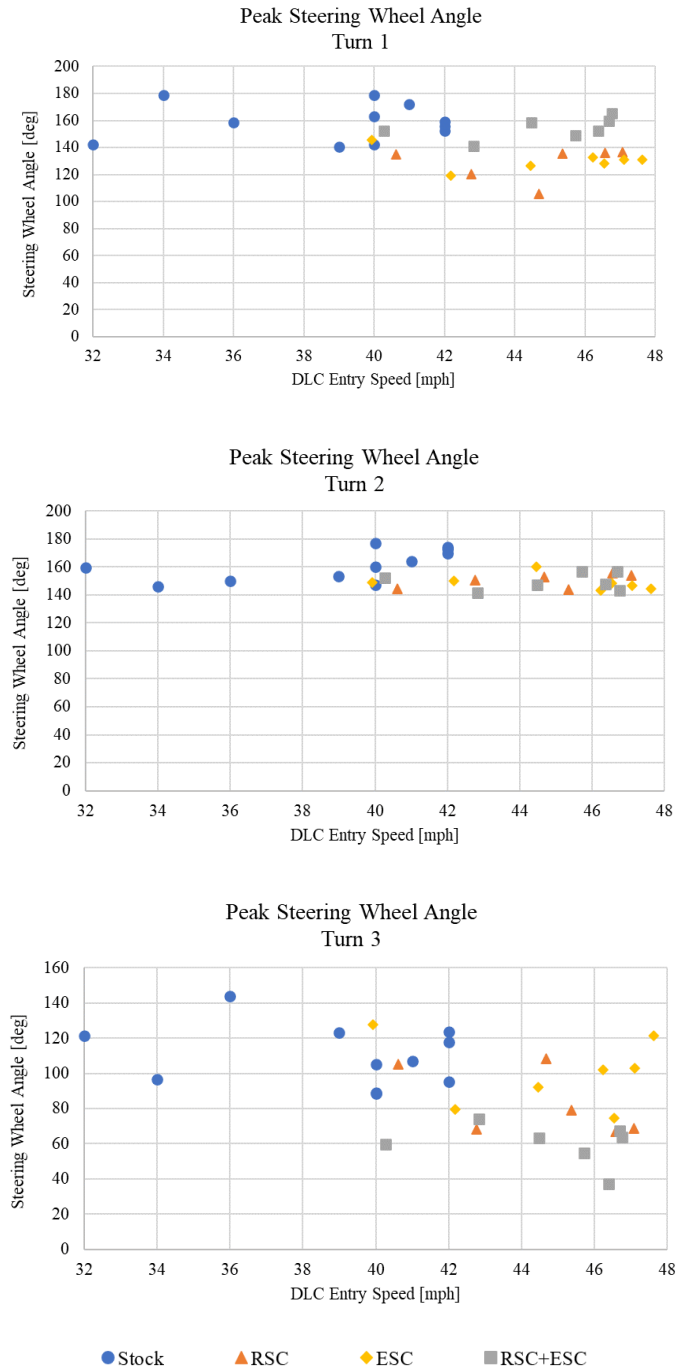


Figure 6-22. Peak steering wheel angles in turn 1, 2, and 3 of the DLC maneuver.

Figure 6-23 presents the speed drop through the maneuver taken as the difference in speed from the maneuver start to 7 seconds later. Just as with the J-turn maneuvers, the stock condition had the least speed reduction whereas RSC+ESC had the most. RSC and ESC only conditions resulted in similar speed drops and were both less than RSC+ESC but more than stock. Figure 6-24 illustrates how each stability control

system’s speed reduction decreases peak lateral acceleration for each vehicle unit at turns 2 and 3. Turns where outrigger contact occurred are omitted for the corresponding vehicle units rolling over. There was not a significant difference among systems in turn 1. At turn 2, the stability control system’s ability to shed speed and reduce lateral acceleration emerges, and by turn 3, each stability control system’s effect on lateral acceleration is clearly evident. By turn 3, RSC+ESC had the lowest lateral accelerations. RSC only and ESC only reduced lateral accelerations to similar degrees, but as mentioned in the outrigger contacts, ESC appeared to improve roll stability more than RSC. In short, because RSC+ESC reduced vehicle speed and trailer peak lateral accelerations more than ESC alone, it is fair to suggest that although the resulting rollover speed of RSC+ESC was less than ESC alone, more test trials and consistent steering profiles would have likely resulted in RSC+ESC having equal or greater rollover speed than ESC alone.

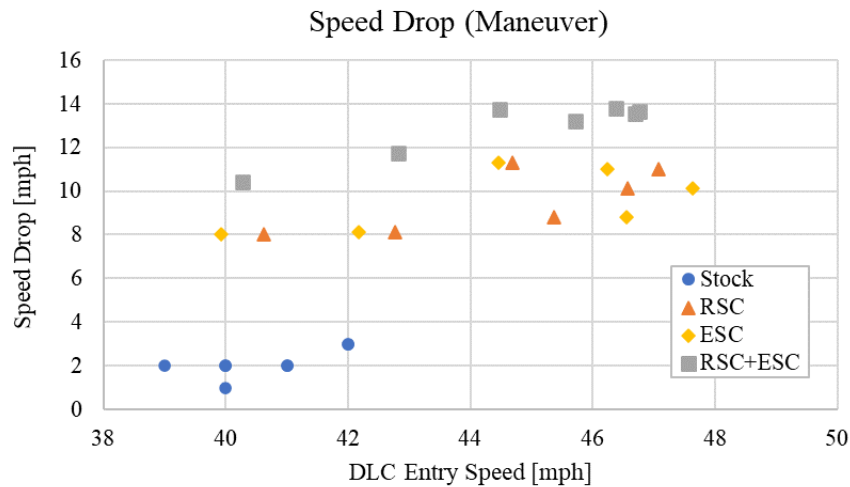


Figure 6-23. Speed drop through DLC maneuver.

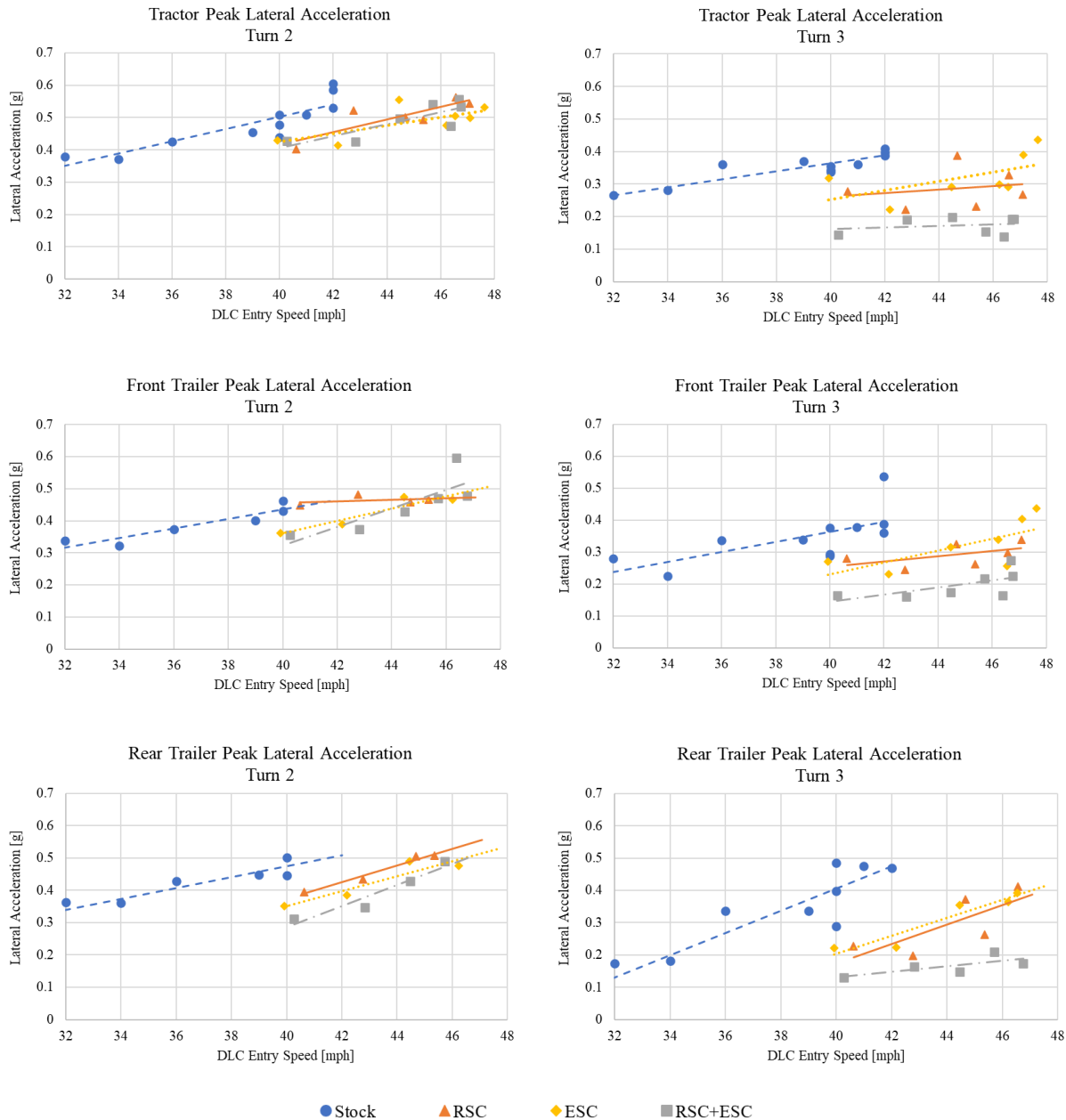


Figure 6-24. Peak absolute lateral accelerations in turn 2 and turn 3 of the DLC maneuver.

Figure 6-25 presents the peak articulation angles at turns 1, 2, and 3. Opposite to the J-turn maneuvers, ESC intervention reduced tractor-trailer articulation, but increased trailer-trailer articulation. This trend was consistent through all three turns. This suggests that ESC reduces likelihood of tractor-trailer jackknifing but may increase likelihood of trailer-trailer or trailer-dolly jackknifing. RSC alone resulted in slightly less articulation angles than the stock configuration in all three turns, especially between the tractor and front trailer. Interestingly, although ESC and RSC alone reduced tractor-trailer articulation angles,

RSC+ESC resulted tractor-trailer articulation angles greater than ESC alone and about the same or occasionally less than RSC alone.

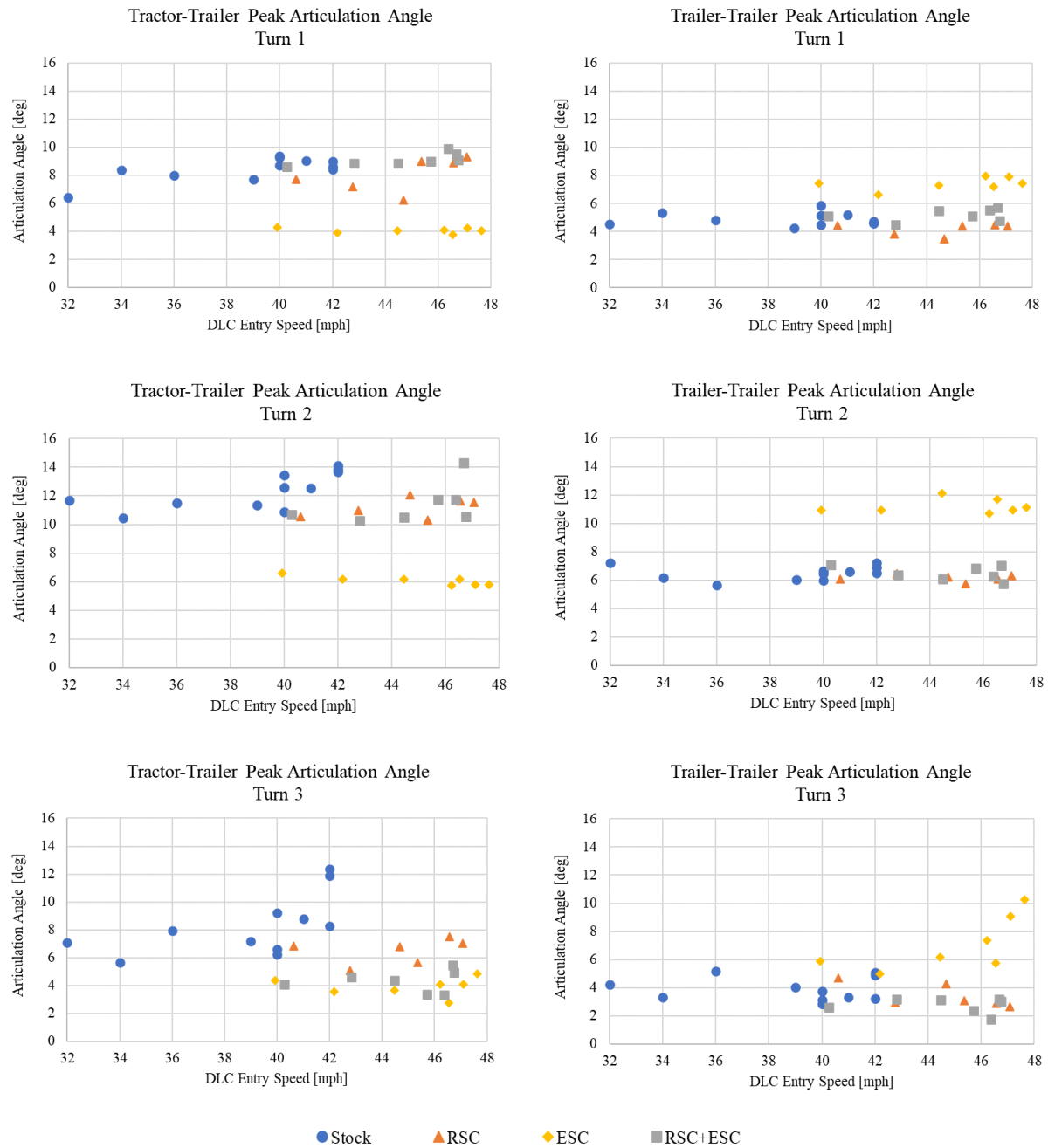


Figure 6-25. Peak articulation angles through the DLC maneuver.

6.5.4 Sine-with-Dwell

As mentioned in Section 6.1.4, the entry speed for all 0.25-Hz SWD tests was 45 mph and the rollover threshold was determined by the steering amplitude instead of speed using the same scheme in Table 6-1. Table 6-5 presents the peak outrigger contact force for the SWD maneuver. It is easy to see that the steering percentage rollover threshold for the stock condition was 60%, and for all others (RSC, ESC, and RSC+ESC) 65%. Unfortunately, the front trailer RSC module was not operating during the RSC+ESC tests, reducing the total vehicle's stability control effectiveness and skewing the results. This was not noticed until too late in the analysis to redo the test trials. Both trailer RSC modules operated properly for the RSC only condition. At the 65% steering percentage, outrigger contact forces were highest for RSC only and lowest for ESC only. The RSC+ESC outrigger contact forces were less than RSC only and just above ESC only. The outrigger contact force differences between ESC and RSC+ESC are within measurement uncertainty. In short, ESC and RSC+ESC with only rear trailer RSC operating, performed very similarly in terms of outrigger contact forces and resulting rollover speed. However, it is suspected that if the front trailer RSC was operable during the RSC+ESC condition, then it would have had a slightly larger rollover steering percentage, likely 70% instead of 65%.

Table 6-5. Outrigger contact forces in SWD maneuver.

Stability Control	Steering Percentage	Peak Outrigger Contact Force [lbf]			
		Front Left	Front Right	Rear Left	Rear Right
Stock (none)	55%	0	0	0	0
	60%*	745	0	718	455
	60%*	539	0	486	244
RSC	50%	0	0	0	0
	60%	0	0	0	159
	65%*	1182	476	1479	868
	65%*	1121	162	888	641
	65%*	1337	219	1530	832
ESC	60%	0	0	0	0
	65%	320	0	0	K
	65%	803	0	0	K
	70%	1315	423	1492	279
RSC+ESC	60%	0	0	0	0
	65%*	938	K	0	198
	65%*	920	201	0	325
	70%	1737	1316	1678	1052
yellow = below threshold (<400 lbf contact force) red = above threshold (≥400 lbf contact force) K = kiss: contact occurred but force is too small to be detected by instrumentation * = rollover steering percentage ^ = no test performed					

As previously discussed for the other maneuvers, RSC intervention typically reduces ESC intervention duration for the RSC+ESC condition compared to the ESC only condition. This was also observed for the SWD maneuver as illustrated in Figure 6-26, but the difference is significantly less clear. These two data sets are so similar for the SWD maneuver because the front trailer RSC was not active in the RSC+ESC condition. As demonstrated in [34], it is more beneficial for the front trailer to have RSC if only one trailer has RSC because the front trailer will then begin braking before the rear trailer would.

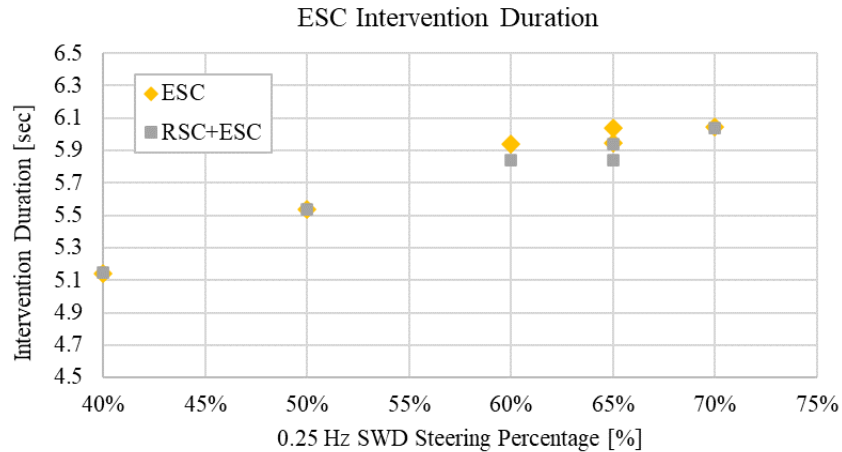


Figure 6-26. ESC intervention duration in SWD maneuver.

Although the rear trailer's RSC intervention was not clearly observed by reducing outrigger contacts nor dramatically affecting the ESC's operation, the speed drop through the maneuver was greater for RSC+ESC than ESC or RSC alone as indicated by Figure 6-27. The speed drop was determined from the start of steering and ending 7 seconds later once the rear trailer exited the maneuver. The tractor's steering profile was approximately 6 seconds long. Notice that for all tested steering amplitudes, the order from least to most speed drop was: stock, RSC, ESC, RSC+ESC. These results coincide with those in previously discussed maneuvers. The distinction is clearer and more consistent for the SWD maneuver compared to others because the entry speed was constant, and the robot provided repeatable steering profiles. Figure 6-28 provides an example of the speed profiles of the different stability control implementations at the 60% steering percentage. Notice that the tractor's speed reduced quicker for ESC only than RSC only. The tractor's speed reduces at about the same rate for the ESC and RSC+ESC conditions until about 5 sec into the maneuver. In the latter half of the maneuver, the rear trailer's RSC is highly active and further slows the vehicle than ESC can alone. However, the first outrigger contact typically occurs on the front trailer and within the first two seconds of the rear trailer's RSC activation. This is not much time to slow the vehicle and prevent trailer rollover. Deceleration during any stability control intervention is discussed in Section 6.6.

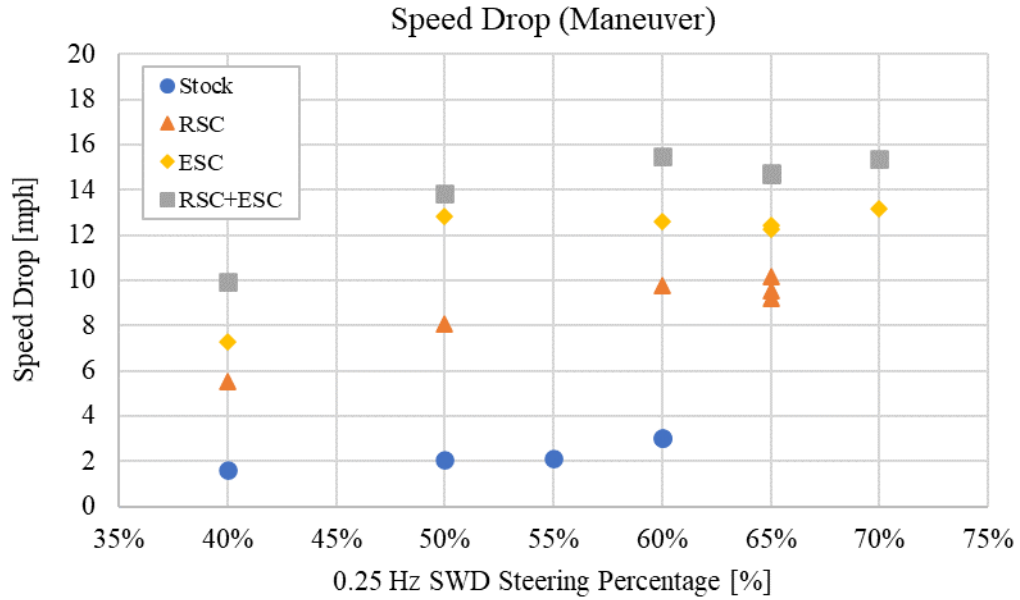


Figure 6-27. Speed drop through the SWD maneuver.

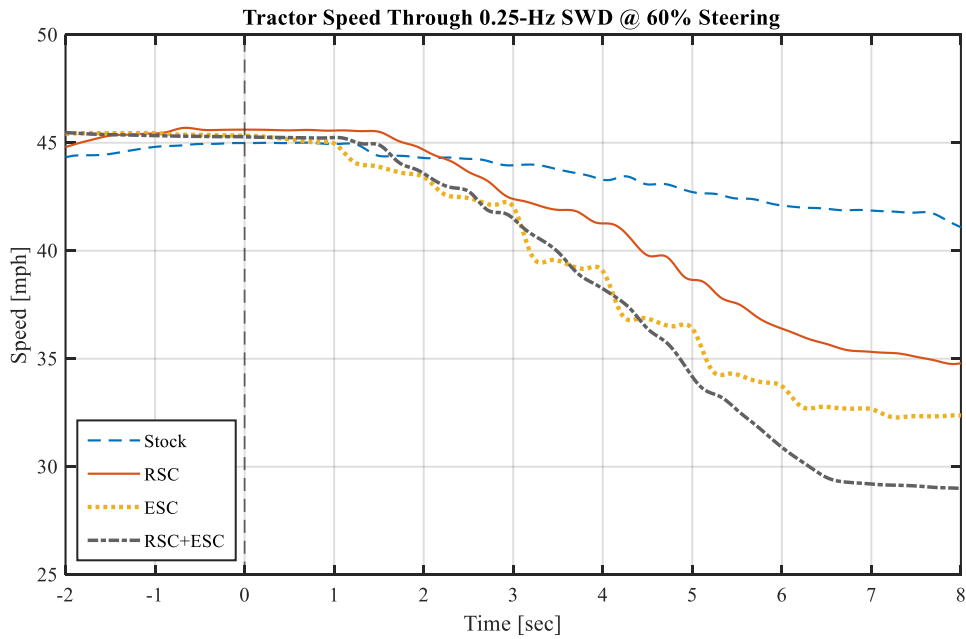


Figure 6-28. Example of tractor speed through the SWD maneuver at 60% steering.

As expected, articulation angles between vehicle units increases with increasing steering amplitude. For a given steering amplitude, the order from smallest to largest peak articulation angles for the different stability control systems in both tractor-trailer and trailer-trailer articulation was: stock, RSC, ESC, and

RSC+ESC. At high speeds, large articulation angles may be indicative of yaw instability, however, for the SWD maneuver, the articulation angles of a given steering amplitude are also affected by variation in the tractor’s path altered by its speed and steering profiles. An LCV with large articulation angles is more susceptible to jackknifing than one with small articulation angles. Notice the order of least to most articulation in Figure 6-29 is the same as least to most speed reduction in Figure 6-27 and Figure 6-28. The stability control systems had significantly more time to reduce the vehicle speed before turn 2 than turn 1. Thus, the difference in articulation angles between stability control systems is more apparent in turn 2 than turn 1. Just as with the J-turn maneuvers, the articulation between the two trailers was greater than between the tractor and front trailer.

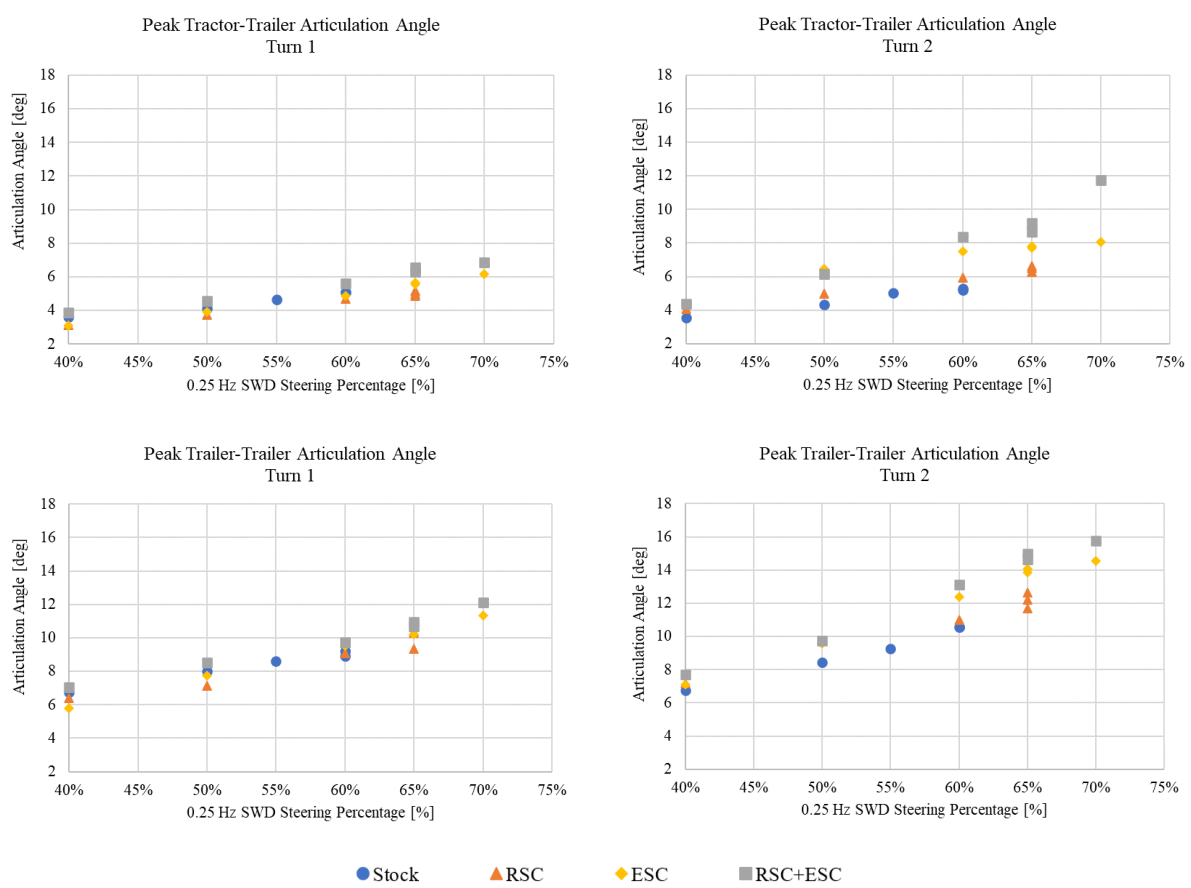


Figure 6-29. Peak articulation angles in SWD maneuver.

6.6 Conclusions

Four unique maneuvers were tested to evaluate rollover propensity of the heavily-loaded double 33-ft trailers with a variety of stability control system implementations – trailer RSC, tractor ESC, and RSC+ESC. As illustrated in Table 6-6, two maneuvers were driver-steered and two were robot-steered.

Similarly, two maneuvers were representative of an exit-ramp and the other two obstacle avoidance. The maneuvers were selected based upon past studies conducted by CVeSS as well as NHTSA. The steering robot provided repeatability in the steering profile whereas the driver provided repeatability in the traveled path. For each trial, the driver entered the maneuver at a prescribed speed then released the throttle and put the transmission into neutral just before passing the starting gate. A spatial trigger flagged recorded data so that individual trials could be fairly compared in time from the maneuver start.

Table 6-6. Roll-stability test maneuver categorization.

	Driver-steered	Robot-steered
Exit-ramp	Driver J-turn	Robot J-turn
Obstacle-avoidance	Double-lane-change	0.25 Hz Sine-with-Dwell

The three tested stability control system implementations (RSC, ESC, and RSC+ESC) noticeably reduced vehicle speed through each tested maneuver, reducing vehicle lateral accelerations to reduce likelihood of rollover compared to the stock vehicle. The rollover threshold for each maneuver was determined as the entry speed or programed steering amplitude that resulted in significant outrigger wheel contact force suggesting high likelihood of complete rollover. Figure 6-30 presents the rollover threshold for each of the tested maneuvers. For the DLC maneuver, the raw test data first suggests that RSC+ESC had less roll-stability than the ESC only configuration. However, a deeper analysis discussed in Section 6.5.3 suggests that this result was caused by significant variation in the driver's steering behavior for different test days among other environmental factors. Had there been more consistent steering input and path trajectory for the DLC maneuver, it is suspected that RSC+ESC would have a rollover threshold equal to or higher than ESC alone. Similarly, for the SWD maneuver, the RSC+ESC condition resulted in the same rollover threshold as RSC only and ESC only. This is likely because the front trailer's RSC was mistakenly inactive during the RSC+ESC condition. If the front trailer RSC was active during the SWD with RSC+ESC, it is suspected that the steering percentage rollover threshold would then be 70%.

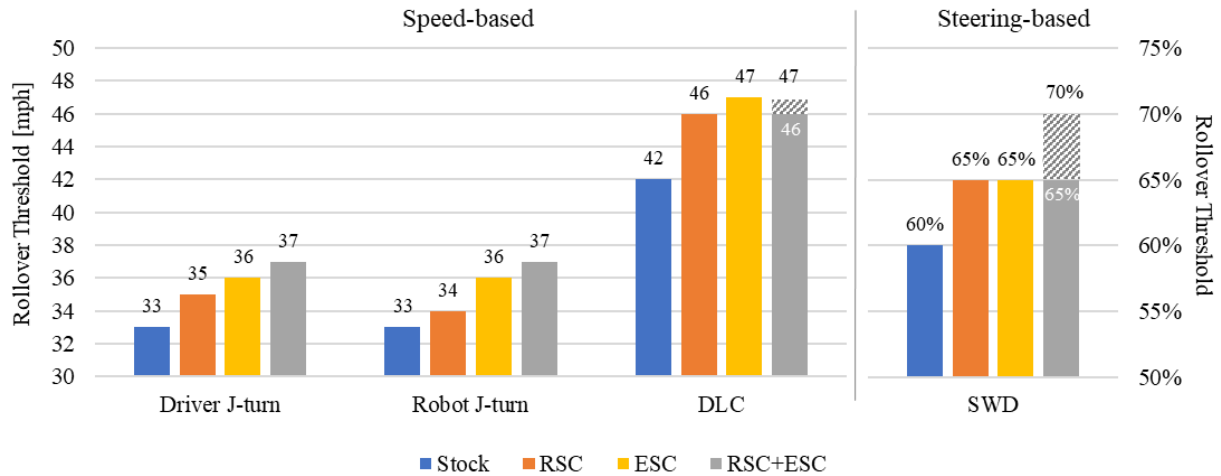


Figure 6-30. Rollover speed and steering percentage for all tested maneuvers and stability control configurations.

Results in Figure 6-30 may be illustrated more pragmatically in terms of roll-stability improvement for exit-ramp and obstacle-avoidance type maneuvers as demonstrated by Figure 6-31. Here, the exit-ramp roll stability is taken as the average rollover threshold increase compared to stock condition for the robot and driver J-turn maneuvers. Similarly, obstacle avoidance roll-stability is taken as the average rollover threshold increase compared to stock condition for the DLC and SWD maneuvers, where 5% increase in SWD steering corresponds to a 1 mph increase in rollover speed. It is fair to equate a +5% in SWD steering to +1 mph in the DLC entry speed because they result very similar changes in lateral acceleration. Specifically, for all stock test trials, the average change in trailer peak lateral acceleration in the three distinct turns of the DLC maneuver was 0.0366g per +1 mph in entry speed, and the average change in trailer peak lateral acceleration in the two distinct turns of the SWD maneuver was 0.0374g per +5% in steering. The vector magnitude – RSS of exit-ramp and obstacle avoidance roll-stability improvement – of Figure 6-31 is shown in Figure 6-32. Notice that RSC+ESC has the highest improvement, then ESC, then RSC. Suspected results of RSC+ESC accounting for the inconsistencies in tests during the SWD and DLC maneuvers is also included.

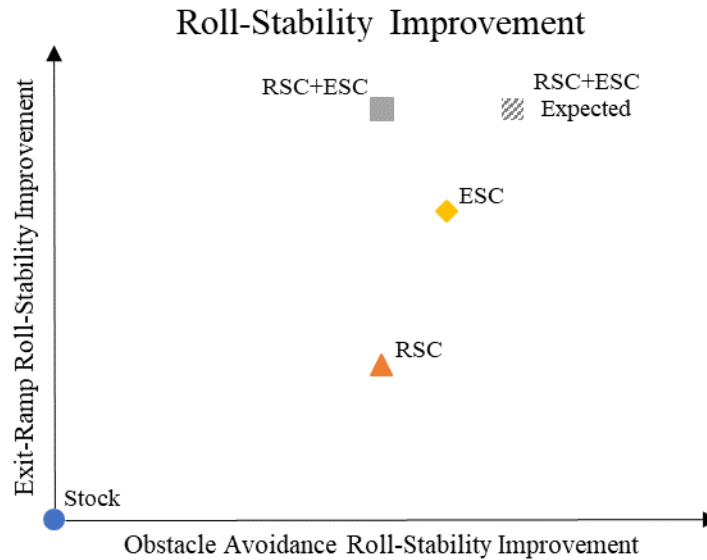


Figure 6-31. Roll-stability improvement trends by maneuver type for each stability control system.

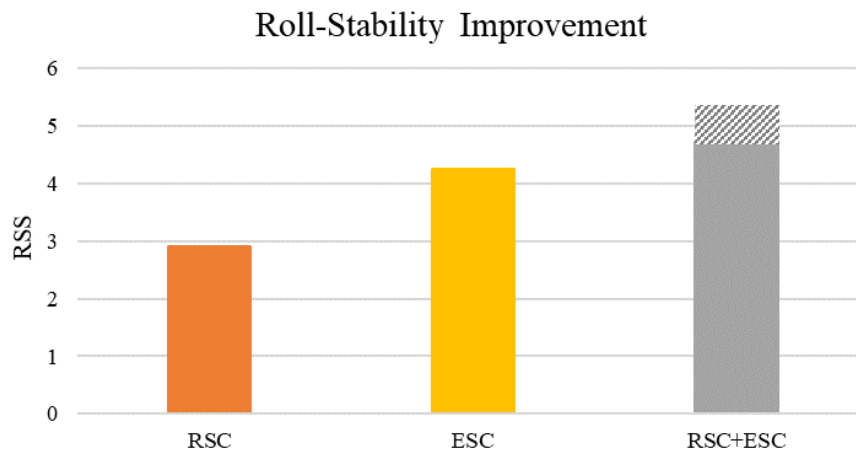


Figure 6-32. Overall roll-stability improvement for each stability control configuration.

To prevent the vehicle from rolling over, the stability control systems activate brake pressure to slow the vehicle (RSC and ESC) and/or correct yaw motion to better follow the driver's steering input (ESC). The stability control systems of one vehicle unit operate independently of those on other units in the same combined vehicle. Stability control intervention occurs as the corresponding vehicle experiences triggering dynamics such as higher lateral acceleration or rate of steering wheel angle. Thus, tractor ESC activates before front trailer RSC, and front trailer RSC activates before rear trailer RSC. Because of this order, ESC can scrub more speed from the vehicle than RSC such that the entry speed may be higher before a

trailer rolls over later in the maneuver. Although ESC may be able to reduce the vehicle speed more through the maneuver simply by being active for longer, the vehicle tends to slow down quicker with RSC by itself than ESC by itself as shown in Figure 6-33. As expected, RSC+ESC results the greatest deceleration for all maneuvers.

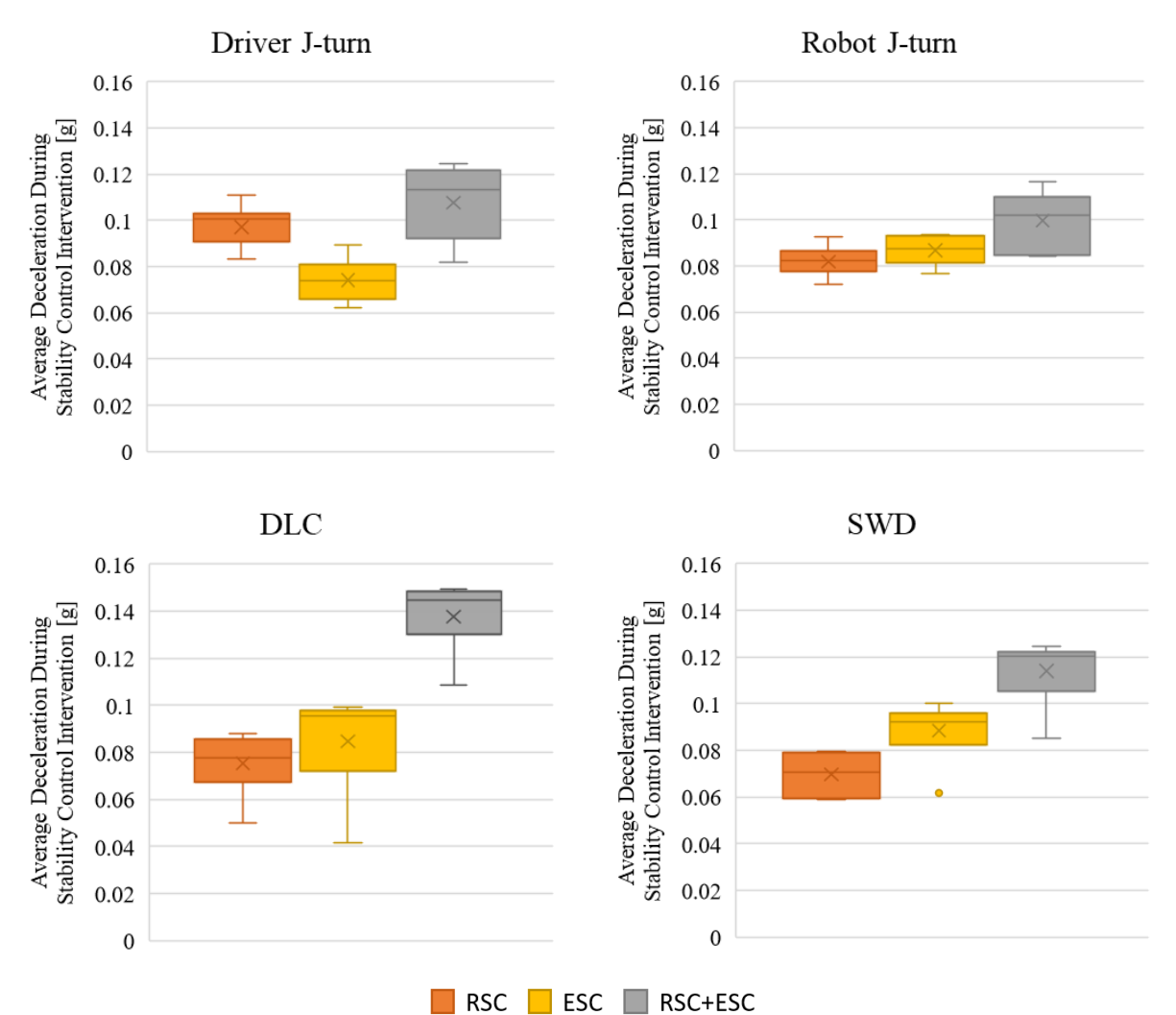


Figure 6-33. Average deceleration while stability control systems are intervening.

Although the focus of this study was on trailer roll-stability, another factor briefly investigated was the stability control’s effect on each vehicle unit’s directional stability as detected by the relative articulation angles between successive vehicle units. Large articulation angles increase likelihood of jackknifing. Table 6-7 summarizes the trends observed for all maneuvers. ESC alone appeared to have the greatest effect on articulation angles, especially in driver-steer maneuvers.

Table 6-7. Stability control system effect on articulation angles.

	Exit-Ramp			
	Driver J-turn		Robot J-turn	
	Tractor-trailer	Trailer-trailer	Tractor-trailer	Trailer-trailer
RSC	slight increase	decrease	slight increase	no effect
ESC	significant increase	decrease	slight increase	no effect
RSC+ESC	increase	decrease	slight increase	no effect
	Obstacle Avoidance			
	Double-Lane-Change		0.25 Hz Sine-with-Dwell	
	Tractor-trailer	Trailer-trailer	Tractor-trailer	Trailer-trailer
RSC	decrease	slight decrease	no effect	no effect
ESC	significant decrease	significant increase	slight increase	slight increase
RSC+ESC	decrease	slight decrease	increase	increase

Chapter 7

Trailer Brake Type Performance Evaluation

This chapter details the test procedure, results analysis, and discussion comparing the performance of two different types of trailer brake systems – drum and disc brakes – on a long combination vehicle. Most studies on commercial vehicle braking have focused on vehicles with a single trailer [86,88,89,90,91,92,93,94,96]. The comparative testing was conducted with the same 33-ft A-double straight-rail trailers in a heavy loading condition (~15,000 lb payload each) as detailed in Chapter 5 and tested in Chapter 6. The tractor and dolly were equipped with drum brakes while the double trailers were tested with drum brakes then disc brakes as shown in Figure 7-1. Both front and rear trailers' brakes were swapped at the same time, so there was no testing with mixed brake type for the two trailers. The drum brakes had been used throughout the roll-stability testing but were thoroughly inspected and serviced before brake testing. The drum brake shoes still had well over half their life left. The disc brakes were brand new and had not been burnished due to time and logistical constraints. Brake burnishing is a method to “break-in” new brakes and improve the surface friction between the brake pad (shoe) and disc (drum). Burnishing is well documented to improve braking performance and consistency [72,95,99,100,101,102]. Considering the disc brakes were not burnished, they are suspected to perform less than their optimal capability. This will be discussed in more detail later as it pertains to the specific tests, but in general is not thought to be a significant factor in the studies' findings. In the trailer brake type performance study, three tests were conducted to assess the braking performance, as follows:

- Straight-line stopping distance
- Roll stability during braking-in-turn
- Effect of brake type on roll-stability control performance

The following sections detail the procedure and results for each study. Background on the two braking systems was detailed in Chapter 2.

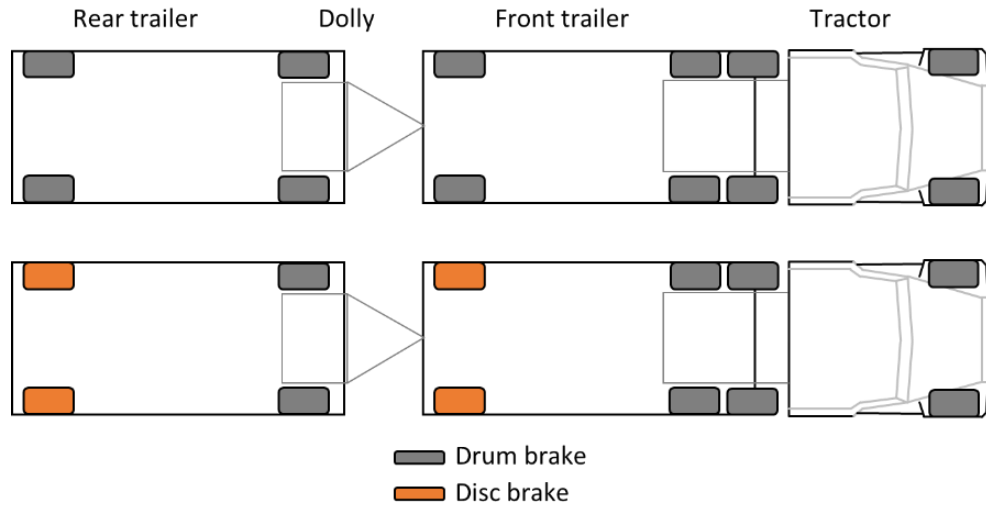


Figure 7-1. Brake type at each axle for testing trailer brake type performance.

7.1 Straight-line Braking Tests

7.1.1 Test Setup and Procedure

The straight-line braking test objective was to determine the influence of trailer brake type on the braking distance of the 33-ft double-trailer truck. The braking distance was measured as the distance traveled from the moment the brake pedal was applied to the vehicle's stopped position. This definition eliminates the human reaction time that varies widely depending on the driver's alertness, age, gender, fatigue, and environmental factors [87]. The braking robot was used during testing, and its design and operation were detailed in Chapter 5. Key highlights of the braking robot include spatial triggering to provide fast, accurate, and repeatable brake application, and regulated air pressure for consistent brake pedal force that guaranteed the brake pedal reached its full stroke. The straight-line brake test speeds ranged from 20 to 65 mph. The spatial trigger was used to determine brake pedal activation position and the stopped position was determined using a forward-facing LiDAR and laser-distance meter to measure the distance to a target placed at a known distance from the spatial trigger target, as illustrated in Figure 7-2. The end target was a free-standing 4x8 ft sheet of plywood covered in highway reflective tape. The tape's high reflectivity improved the LiDAR's measurement accuracy for the challenging environment – in direct sunlight and up to about 120 ft away. The spatial trigger and end target's staged locations for each test speed was marked on the test surface so that they could be placed in the same position for different days of testing for fair comparison between brake types.

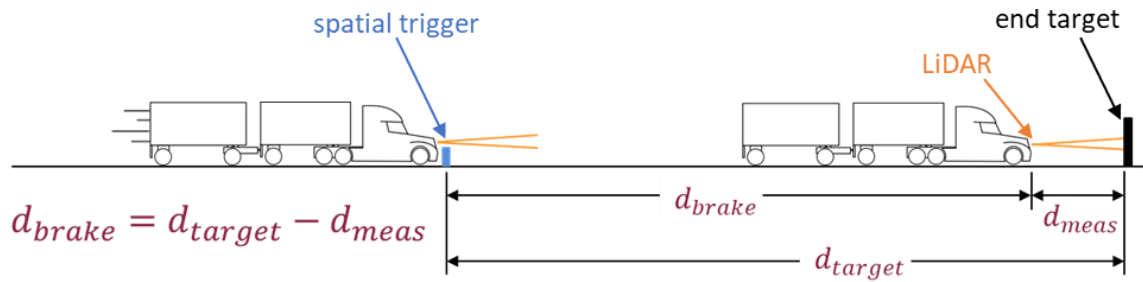
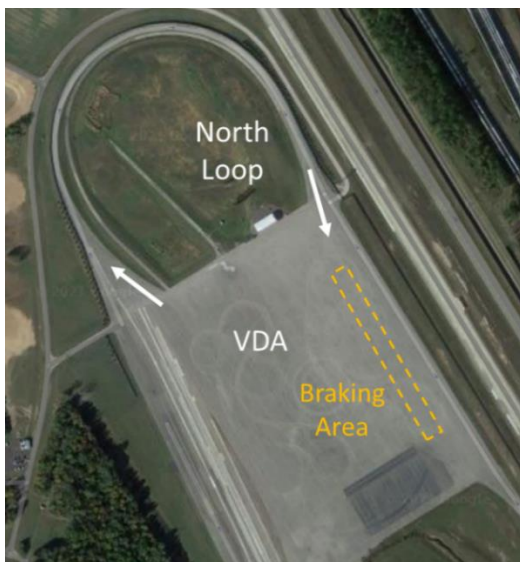


Figure 7-2. LiDAR was used to measure the stopped distance from the reference end target. The reference end target was positioned at a known distance from the starting gate's spatial trigger.

A scheme of alternating test speeds and cool-down laps between trials was followed to prevent significant brake fade. After each speed was tested three times consecutively, brake temperatures were measured at each drum/disc by an infrared thermometer. For entry speeds of 40 mph and higher, the north loop (NL) was used to get the vehicle up to speed rather than staying within the vehicle dynamics area (VDA) between trials. Taking the north loop also provided convective cooling for the brakes. For entry speeds of 50 mph and higher, an additional cool-down lap around the north loop was added between each test trial. The test speeds were alternated in the following order: 20, 60, 30, 55, 40, 50, 45, 65 mph. The scheme of test speed, cooldown laps, and brake temperature measurements is tabulated in Figure 7-3.



(a)

Set	Speed [mph]	Qty	Entrance Route	Cool-down Lap Between Trails
1	20	x3	VDA	No
				Check temperatures
2	60	x3	NL	Yes
				Check temperatures + cool-down lap
3	30	x3	VDA	No
				Check temperatures
4	55	x3	NL	Yes
				Check temperatures + cool-down lap
5	40	x3	NL	No
				Check temperatures + cool-down lap
6	50	x3	NL	Yes
				Check temperatures + cool-down lap
7	45	x3	NL	No
				Check temperatures + cool-down lap
8	65	x3	NL	Yes
				Check temperatures

(b)

Figure 7-3. (a) Satellite image of vehicle dynamics area (VDA) and north loop (NL) used for testing. (Maxar Technologies, State of Ohio, 2022). (b) Test scheme of alternating speeds, measuring brake temperatures, and cool down laps between trials and test speed sets.

For each trial, the test driver approached the braking area at the desired entry speed using a GPS speedometer with a digital display. Just before passing the spatial trigger, the driver disengaged the clutch, and the passenger set the braking robot on standby. As soon as the spatial trigger was passed, the braking robot's pneumatic actuator automatically struck and held the brake pedal at the pedal's maximum stroke. The delay between passing the spatial trigger and the actuator striking the brake pedal was approximately 40 ms as illustrated previously in Figure 5-29. This results in traveling about 3.8 ft past the spatial trigger before the brake pedal was pressed during a 65-mph test. An appropriate delay was accounted for in all trials. Tests with each brake type were conducted on different days but had very similar weather conditions at the time of testing: sunny, occasional low winds up to 7 mph and around 50°F. The wind was coming from different directions for the two days but is believed to cause negligible impact on the straight-line braking distance due to the low wind speeds. In short, environmental factors during the test days are believed to be negligible for comparing the brake type's performance on straight-line braking.

7.1.2 Braking Distance, Time, and Deceleration

The braking distance for all trials of both brake types are shown in Figure 7-4. To best compare the brake types, the two sets of data for drum and disc brakes were fitted with 2nd-order polynomials with R-squared values of 0.9954 and 0.9976, respectively. A higher R-squared value loosely suggests higher repeatability and consistency in braking performance. A 2nd-order polynomial curve fit is appropriate for this dataset because the dominant dynamics that dictate the braking distance are accurately modeled as 2nd-order polynomials with respect to entry speed. From an energy conservation perspective, the work done by brakes to stop a vehicle on a flat road (zero potential energy: $\Delta PE = 0$) is equal to the vehicle's change in kinetic energy (ΔKE) minus nonconservative work done by road loads such as aerodynamic drag and rolling resistance (W_{rl}). This relationship is illustrated by Equations (7-1) to (7-3). The kinetic energy includes the total vehicle mass velocity and the rotational inertia of the wheels and axles. Although nonconservative forces may exist in the wheel bearings and other areas, aerodynamic drag and tire rolling resistance are the dominant loads. The change in kinetic energy is approximately two orders-of-magnitude more than the road load work [69], making the road load negligible.

$$W_{brakes} = \Delta KE - \Delta PE - W_{rl} \quad (7-1)$$

$$KE = \frac{1}{2}mv^2 + \frac{1}{2}\sum I\omega^2 \quad (7-2)$$

$$W_{rl} = \int \left(\frac{1}{2} \rho C_d A v^2(x) + mg(C_{rr1}v(x) + C_{rr0}) \right) dx \quad (7-3)$$

Figure 7-5 shows the braking distance reduction from trailer drum brakes to disc brakes according to the polynomial curve-fits of the experimental data shown in Figure 7-4. Notice that at 70 mph, the braking distance is projected to shorten by nearly two passenger car lengths by using disc brakes in place of drum brakes on the trailers. Based on the results of Figure 7-4 and Figure 7-5, the tested trailer disc brakes resulted significantly shorter braking distances than the drum brakes for speeds of 50 mph and higher. For speeds less than 50 mph, there was not an evident difference in stopping distance between the two brake types.

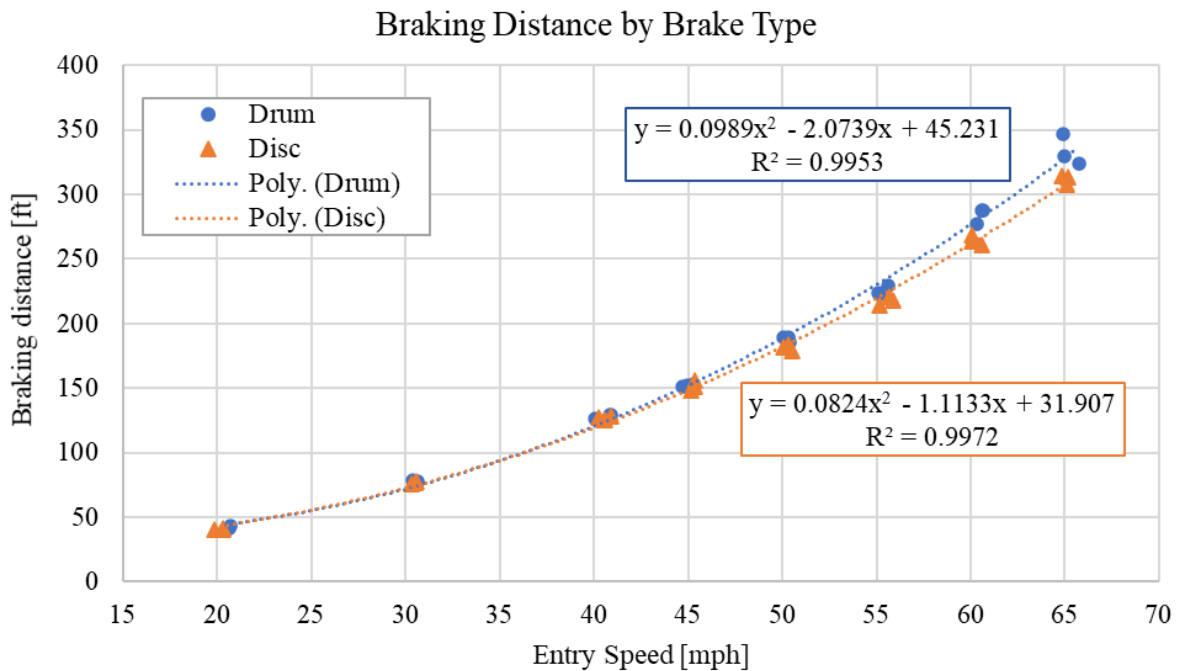


Figure 7-4. Comparison of measured braking distance between disc and drum brakes at different speeds.

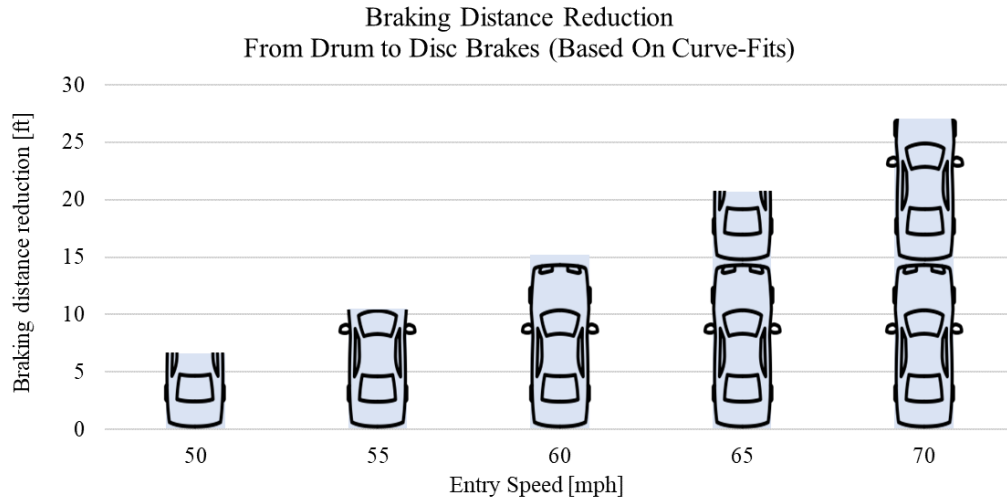


Figure 7-5. Expected braking distance reduction in terms of average passenger car length by switching trailer brakes from drum to disc according to 2nd-order polynomial curve-fits from test data.

The average deceleration for the entire braking period of the three trials at the same entry speed is shown in Figure 7-6. These data points were calculated by Equation (7-4) and include pneumatic transport delay in the braking system. These values are the average of the three trials shown in Figure 7-4. Just as observed in the braking distance, the drum and disc brakes resulted in similar average decelerations for speeds <50 mph and the disc brakes resulted in more deceleration than the drum brakes at speeds ≥ 50 mph. The average deceleration increases with an increase in speed up until about 55 mph. For speeds greater than 55 mph, the average deceleration decreases with an increase in speed. These two trends suggest that: (1) the pneumatic delay is a substantial portion of the braking time and distance for lower speeds, and (2) the disproportionate increase in kinetic energy, especially for higher entry speeds, dominates the change in average deceleration for a step increase in entry speed. The average measured brake time delays are listed in Table 7-1. These values are the time delay from the moment the braking robot first contacts the brake pedal to the first rise in brake chamber pressure for all tested entry speeds and trials. These values are similar to the data used in a pneumatic brake circuit model detailed in [69] but include the time delay of the actuator, brake pedal stroke, and treadle valve. The pneumatic transport delay itself is independent of entry speed and brake type.

$$a = \frac{1}{3} \sum_{i=1}^3 \frac{v_i^2}{2\Delta x_i} \quad (7-4)$$

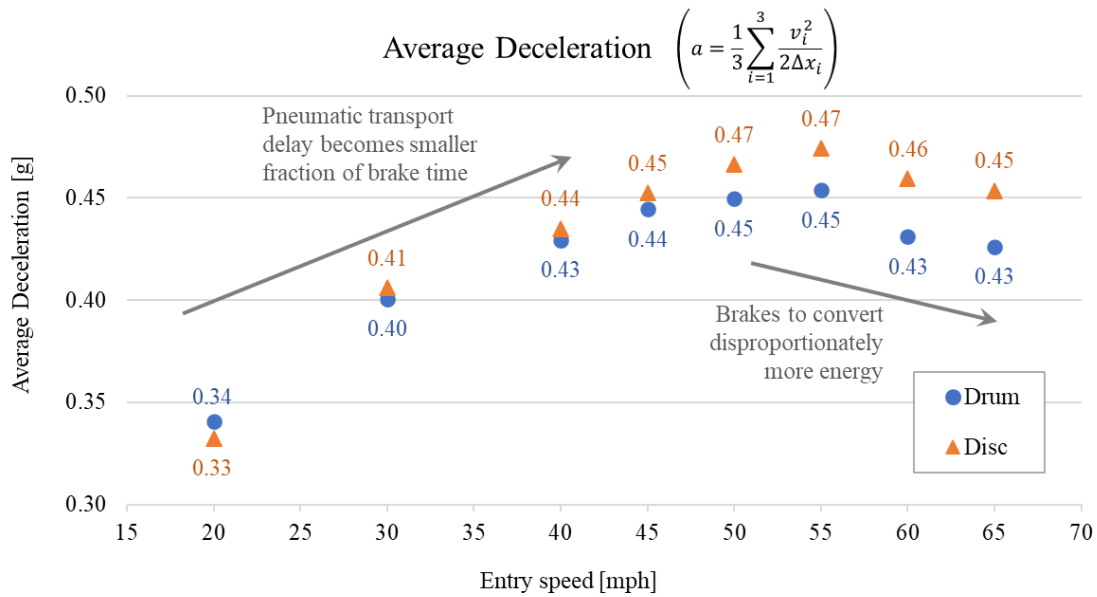


Figure 7-6. Average deceleration for entire braking maneuver for each tested entry speed.

Table 7-1. Brake chamber pressure time delay from brake pedal initiation to start of brake chamber pressure rise.

Vehicle Unit	Brake Delay [sec]
Tractor Drive Axle	0.09
Front trailer (33-ft)	0.26
Rear trailer (33-ft)	0.52

Figure 7-7 and Figure 7-8 clearly illustrate how the pneumatic transport delay's fraction of braking time and braking distance reduces as the entry speed increases. This is expected because the brake delay is independent of vehicle speed and the brake time increases with entry speed. The plotted lines are exponential curve fits of all test data. Exponential functions in the form of $y = Ae^{-b}$ were chosen because the pneumatic transport delay's percentage of total brake time approaches a horizontal asymptote along the x-axis as the entry speed approaches infinity. Each curve-fit's R-squared value was >0.98 . There is an evident difference between the brake types' percentages of total brake time and distance because the disc brakes had shorter brake times and distances than the drum brakes. However, the pneumatic delay from the moment the brake pedal is pushed to the moment pressurized air first reaches the brake chamber is equal between the two brake types. This is because only the brakes themselves were changed between tests. The control lines, relay valves, etc. on the test vehicle and trailers did not change. The disc brakes required slightly longer (by 1-2 ft) hoses from the brake controller to the brake chamber than the drum

brakes; however, this was negligible compared to the total distance traveled by air through the entire air braking system and there was no evident difference in the brake delay between brake types.

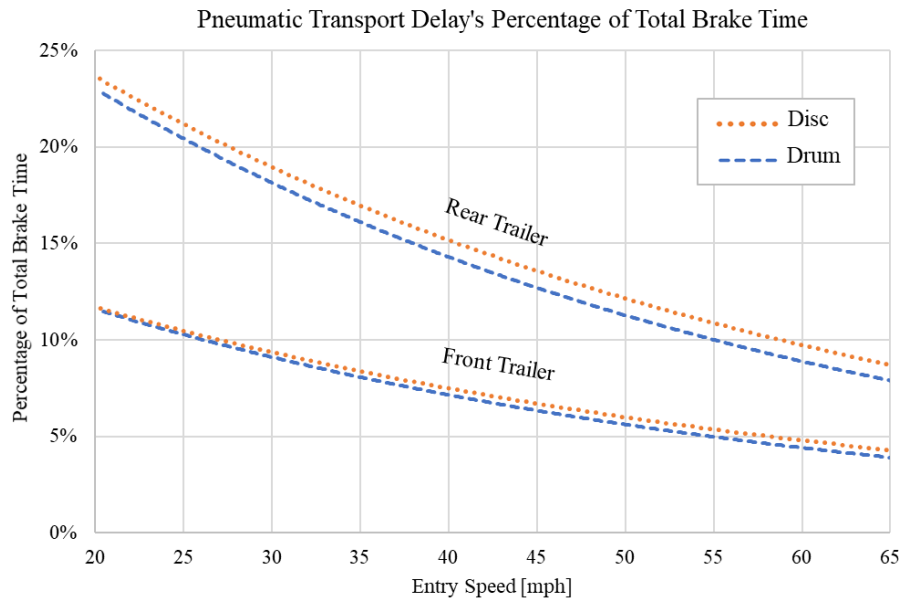


Figure 7-7. Pneumatic transport delay as percentage of total brake time.

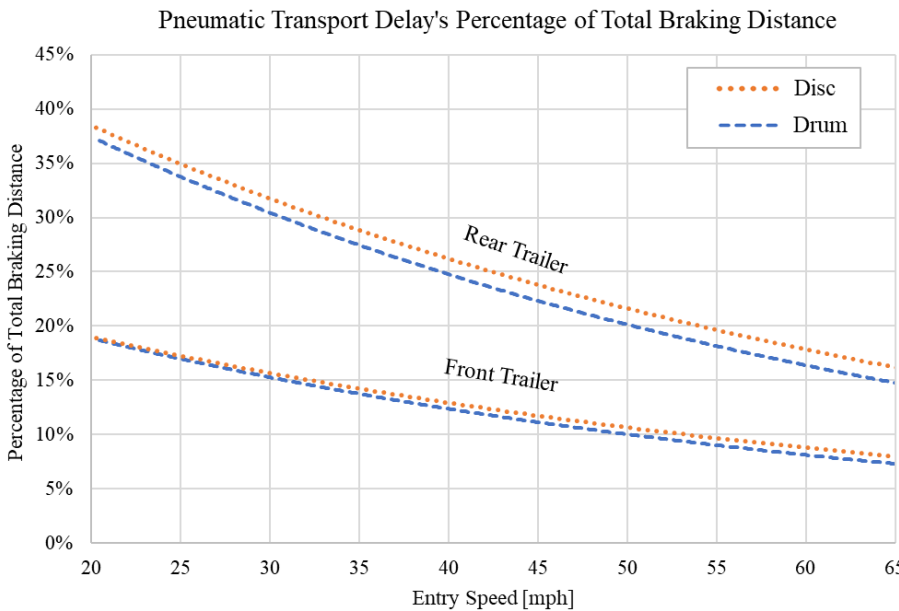


Figure 7-8. Pneumatic transport delay as percentage of total braking distance.

To better represent the trailer brake types' influence on braking performance of the entire vehicle, the average deceleration during full-braking is compared in Figure 7-9. In this plot, full-braking is considered to occur when all tractor and trailer axles receive maximum brake pressure or ABS intervention. This definition eliminates the pneumatic response in the control line and more clearly accentuates the difference in trailer brake type performance. Just as illustrated before, when equipped with disc brakes, the trailers experienced a higher average deceleration for entry speeds of 50 mph and higher. The average deceleration during full braking was approximately 0.63-0.64g for both brake types for speeds less than 50 mph. At higher speeds, the average deceleration decreased when the entry speed increased. The deceleration decreases with an increasing entry speed because, as shown earlier in Equation (7-1), the brakes are required to convert disproportionately more kinetic energy into heat when slowing down from higher speeds. For example, decelerating from 65 to 60 mph requires about 47% more energy to be dissipated by the brakes than decelerating from 45 to 40 mph. Even though the speed difference is the same, higher entry speeds require significantly more energy dissipation than lower entry speeds.

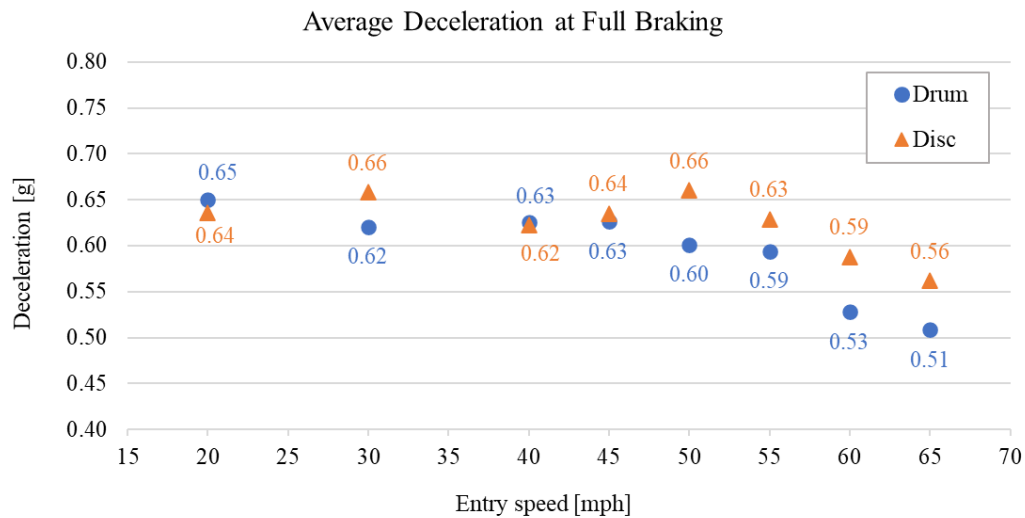


Figure 7-9. Average deceleration at full braking for straight-line testing.

Another observation from the tests was that the disc brakes tended to perform more consistently than the drum brakes during repeated straight-line braking tests. This is loosely suggested by the lower R-squared value for the drum brakes' polynomial curve-fit compared to the disc brakes' in Figure 7-4, but the consistency is better illustrated by changes in braking distance range (longest distance minus shortest distance) for the three trials of each test speed set as shown in Figure 7-10. Because the entry speed for each test did not perfectly match the intended speed (i.e. one trial may have been 64 mph while another closer to 66 mph), the braking distance range for each trial was normalized by dividing by the average

entry speed of their respective three trials. The average normalized range of braking distance for test speeds of 50 mph and higher were 3.9% and 2.8% for the drum and disc brakes, respectively. Lower speeds (<50 mph) were omitted from this calculation because the difference in braking distance between the two brake types was insignificant. The drum brakes' higher normalized braking distance range and the braking distance ranges shown in Figure 7-10 suggest that the drum brakes performed less consistently than the disc brakes during the repeated tests.

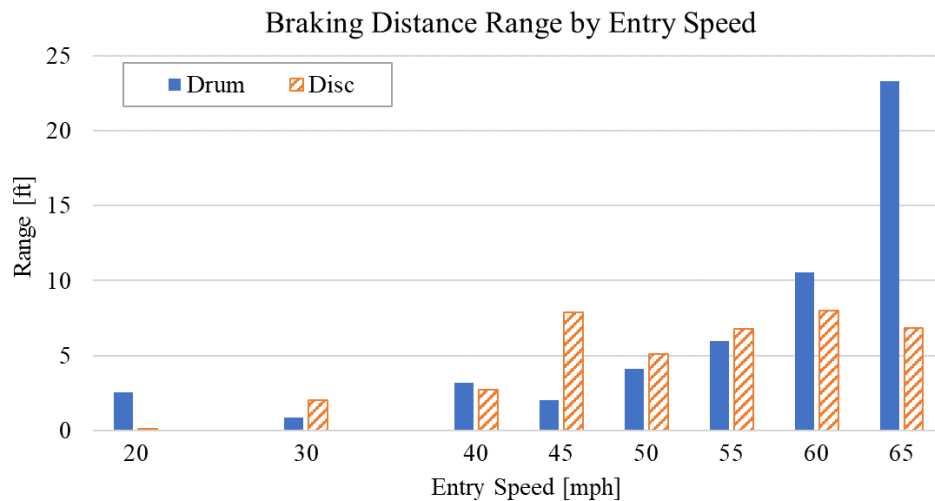


Figure 7-10. Range of braking distance of among the three trials for each test speed.

For all data, the average entry speed range of the three trials for a given target entry speed was 0.45 mph. The largest entry speed range was 0.97 mph and occurred during the 65-mph test set with drum brakes. At first glance, one might hypothesize that this large entry speed range contributed to the large braking distance range of the 65-mph test set with drum brakes. However, this was not the case. The first and fastest of the three trials had the shortest braking distance and the slowest and last trial had the longest braking distance for the drum brakes. The disc brakes' braking distance only increased slightly through each consecutive trial within the same test speed set. These differences are presented in Figure 7-11 where the three trials' speeds and braking distance of both brake types for the 65-mph test speed set are plotted. Notice that the order of each test trial is also listed in the plot and that the third drum brake test had a braking distance 23 ft longer than the disc brake's third test for approximately the same entry speed. The increase in braking distance for each consecutive trial is indicative of brake fade. Brake fade is caused by factors including, but not limited to, thermal expansion of the brake drum, distortion of brackets and mechanisms under thermal and mechanical stress, and reduced friction coefficient between the brake shoe (pad) and drum (disc) [85].

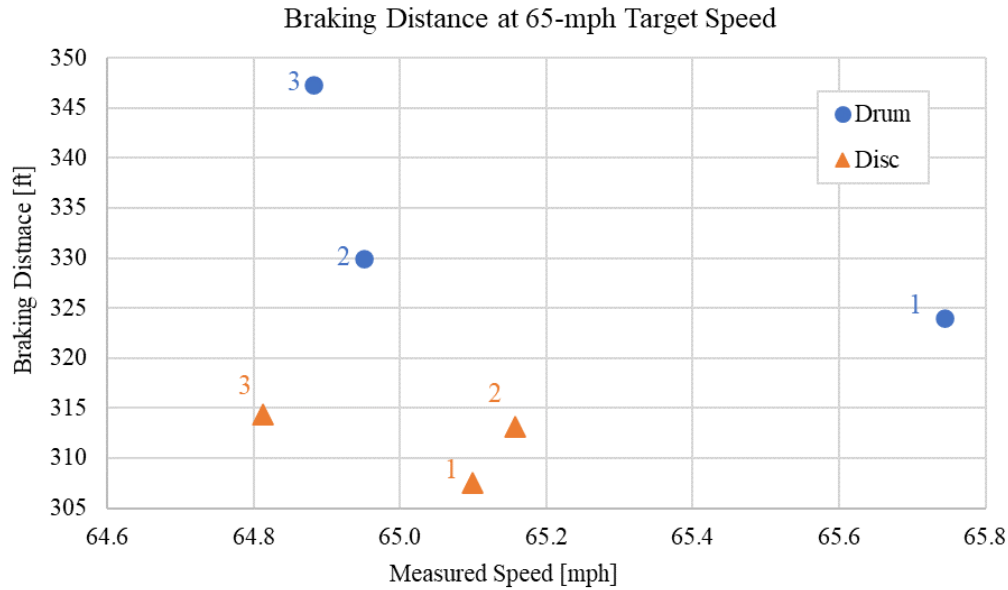


Figure 7-11. Comparison of braking distance at the 65-mph test speed set for disc and drum brakes.

7.1.3 Brake Temperature ABS Intervention

After conducting three trials at each test speed, tractor and trailer brake temperatures were measured with an infrared thermometer. The measured temperatures are shown in Figure 7-12. Notice that all temperatures were relatively consistent between test speeds, thanks to the scheme shown earlier in Figure 7-3. Each bar denotes the average of the left and right wheels' brake temperatures for a given axle. There was no significant or consistent difference in brake temperature from the left or right side of a given axle.

For nearly all test speeds on both drum and disc brakes, the front trailer axle had the highest brake temperature. The front trailer with disc brakes reached significantly higher temperatures than any other axle for all tests. Excluding 20 mph tests due to the substantially lower temperatures, Figure 7-13 presents the total average brake temperatures for each axle of both trailer brake types. The front trailer axle was the only axle that showed significant differences in brake temperatures between brake types. The tractor and dolly axles' brake temperatures decreased negligibly when the trailer brakes were switched from drums to discs. Because the tractor and dolly did not swap equipment when changing the trailer brakes, the similar temperatures on the tractor and dolly from testing the trailer drums to trailer discs demonstrates the consistency in the test procedure between the two test days.

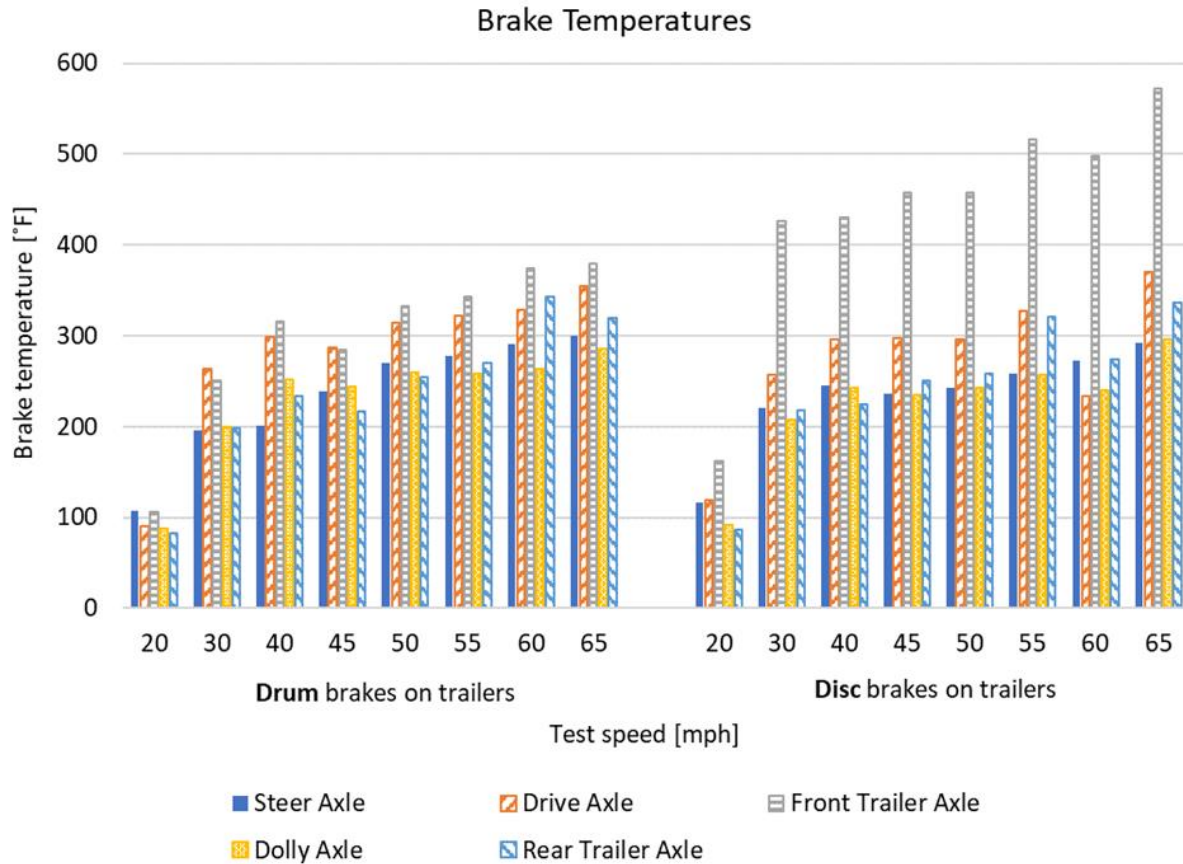


Figure 7-12. Brake temperatures measured after each test speed test.

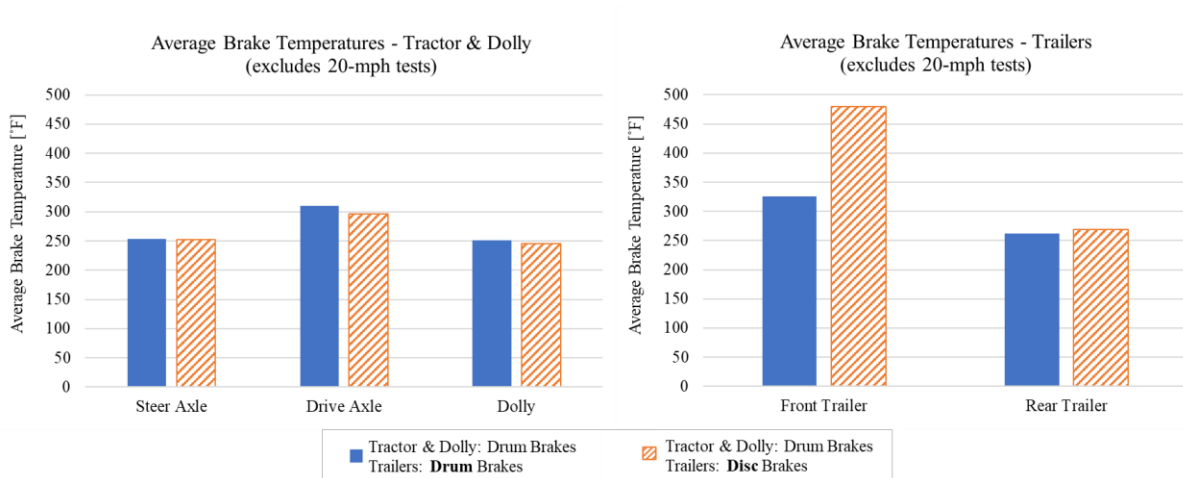


Figure 7-13. Total average axle brake temperatures for all speeds excluding 20-mph.

Observing higher rotor temperatures on disc brakes than drum temperatures on drum brakes has been documented in literature [70,77]. There are many factors that can affect the brake rotor/drum temperature including, but not limited to, material heat capacitance, thermal mass, and heat transfer properties from the testing environment and aerodynamics around the brakes and wheels. Because this study was not designed to identify the exact thermal properties of trailer brakes, discussion on brake temperatures here are only included to present the test results and show that brake temperatures were accounted for in the test scheme to generate conditions that fairly compare the different brake types while accommodating track time constraints. Note that increased brake temperature reduces braking performance by reducing the friction coefficient and through thermal expansion and mechanical strain [31,85,87,88]. Refer to [78-83] for more information on brake rotor/drum thermal properties under hard braking.

Although the disc brakes ran hotter than drum brakes, the discs demonstrated better and more consistent braking performance. This is partly attributed to the direction the force is applied on the brake type's friction pair. On the drum brakes, the shoes push outward against the inside of the brake drum. As the brake drum heats up, it expands outwardly, reducing the contact pressure against the brake shoe. In contrast, the brake pads of a disc brake squeeze the sides of the rotor. As the rotor heats up, it expands toward the brake pads, increasing the contact pressure.

Another factor that can greatly influence braking performance and consistency is the brake pad/shoe friction material. The friction material typically consists of four components: fiber, binder, filler, and additive. Fibers are the primary friction product and are typically glass, metal (steel, brass, copper), aromatic polyamide, or ceramic. Binders are designed to ensure structural integrity of the friction material and bind the different components together. These must have high thermal stability and often include silicone or epoxy modified resin. The most common resin binder is phenolic resin. Fillers are included to reduce the friction material cost and improve manufacturability. The choice of filler materials depends on the binders and fibers used. Common inorganic fillers include barium sulphate, mica, vermiculite, and calcium carbonate. Common organic fillers include cashew dust and rubber. Most fillers help suppress noise and have low thermal conductivity. The last component of the friction material is the additive. Additives are either lubricants or abrasives. Lubricants stabilize but reduce the friction coefficient between the friction material and rotor/drum and typically include graphite. Abrasives increase the friction coefficient but are often susceptible to significantly change with temperature [84].

Manufacturers do not openly share their proprietary friction material compositions, so the exact friction material composition of either tested brake type is not known. However, the tested brakes are standard

issue equipment for the tested 33-ft straight-rail box-van trailers. There may be some variations in maximum capable brake torque among different models of drum and disc brakes, but as shown in literature and demonstrated in this study, drum brakes are typically more susceptible to brake fade than disc brakes.

7.1.4 ABS Intervention

For both brake types, the front trailer exhibited the highest temperatures. This is because the front trailer brakes significantly harder (more brake chamber pressure) for a longer duration than the rear trailer. Figure 7-14 illustrates the difference in front and rear trailer brake pressures for both brake types. The front trailer can apply higher braking forces than the rear trailer because the rear trailer and dolly pitch motions increase the front trailer's axle load, increasing the maximum amount of available braking torque before wheel lockup. To prevent lock-up, ABS intervenes and modulates the brake chamber pressure as demonstrated in Figure 7-14 for the disc brakes and later presented in Figure 7-15. Time $t = 0$ sec was the moment the spatial trigger was passed.

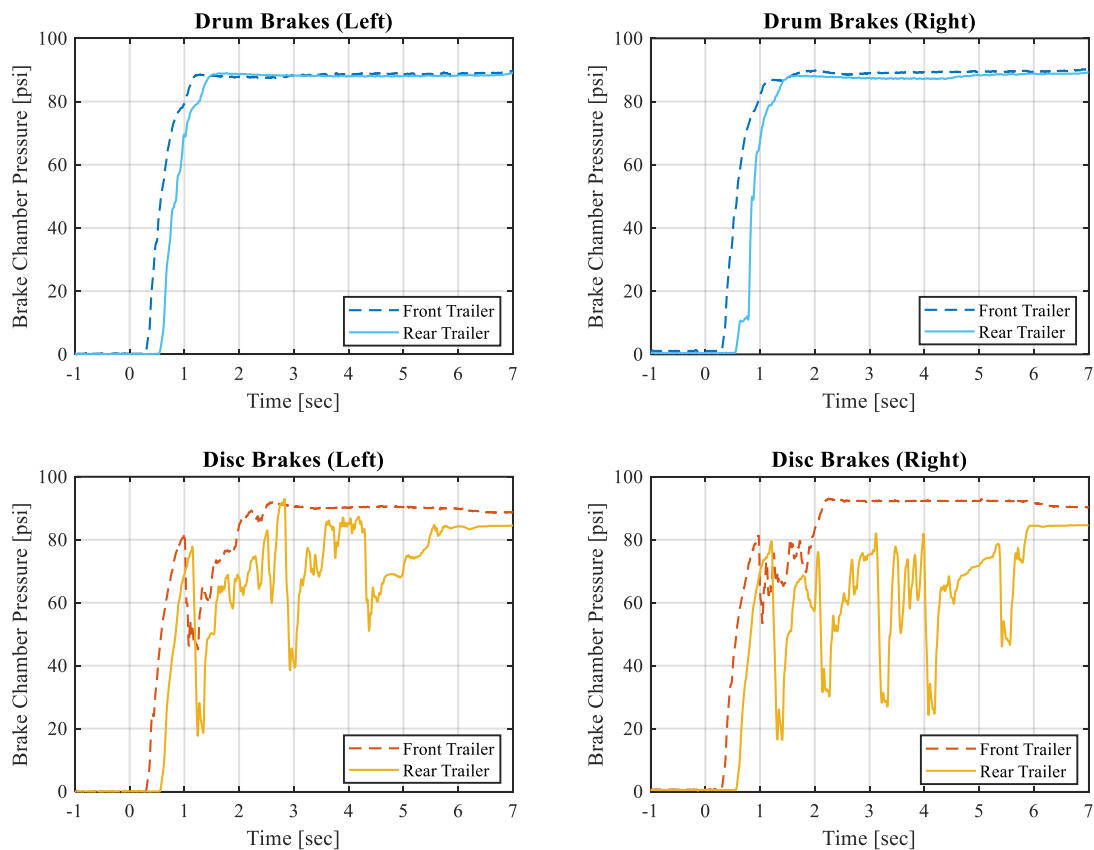


Figure 7-14. Comparison of front and rear trailer brake pressures during a 65-mph straight-line braking test.

For trials shown in Figure 7-14, ABS intervention was very limited for the drum brakes whereas ABS was extremely active for the disc brakes, especially the rear trailer. This is in part due to the disc brakes having greater torque than the drums and more quickly reaching the tire's braking limit. The rear trailer's ABS was active for most test trials for both brake types. Each vehicle unit's average ABS intervention duration for the three trials of each test speed is shown in Figure 7-15. When the trailers were equipped with drum brakes, trailer ABS intervened for significantly less time than when the trailers were equipped with disc brakes. When equipped with disc brakes, both trailers' ABS systems intervened for nearly all test speeds. ABS intervention is dependent on tire slip that is affected by weather conditions, tire properties, road surface, and tire loading. Because the weather for both test days was similar, the tire models and conditions were the same, and the test vehicle unchanged, it is suspected that the longer trailer ABS activation duration of the disc brakes is a direct indication of the disc brakes producing higher torque than the drum brakes. This coincides with the shorter braking distances and higher average decelerations presented earlier. Because the tractor was tested with the same equipment throughout, the ABS duration did not change substantially at a given test speed for different trailer brake types. This again demonstrates the test procedure's consistency between the two test days.

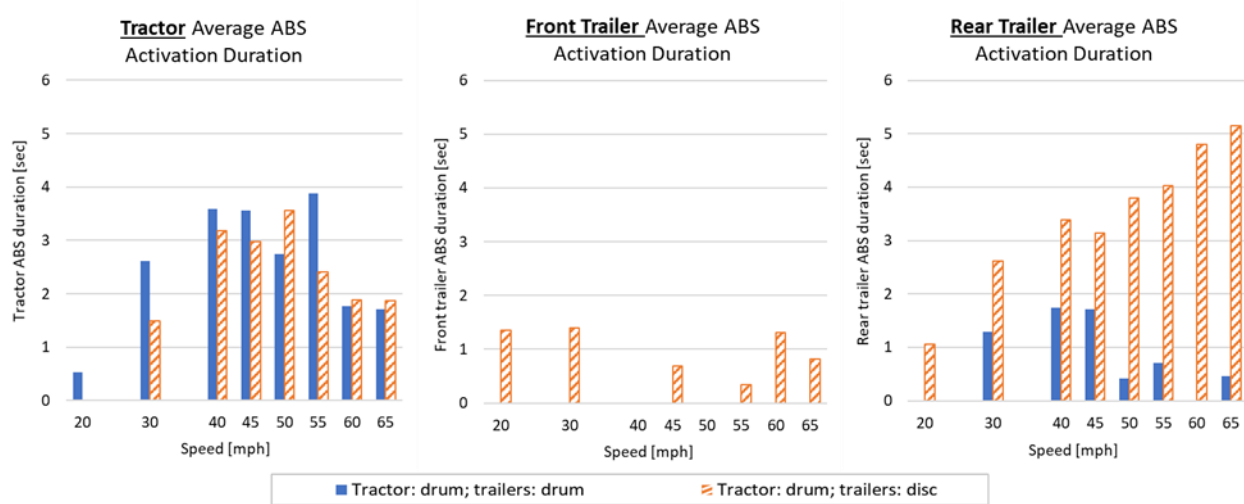


Figure 7-15. Average ABS activation duration of the three trials for each test speed set during straight-line braking.

7.1.5 Braking-Induced Suspension Jounce

The trailing unit's forward pitch motion and load transfer is caused by their high C.G. position relative to the braking force at the ground and resistive forces from a leading unit's braking. Under steady-braking and assumed constant deceleration, each vehicle unit, its suspension, and its axle can be modeled as a single rigid body. This does not capture suspension reactions to large brake modulations and pitch

oscillations but is useful to illustrate a difference in the front and rear trailer axle loads caused by longitudinal load transfer in a quasi-steady-state. Wheel and axle rotational inertia is neglected for simplicity. Figure 7-16 illustrates each trailing unit's free-body diagram with appropriate forces. The force arrows are drawn in their positive direction during braking. The inertial force that resists deceleration is positive in the forward direction. Because we assume quasi-steady conditions and rigid bodies, force and moment summations provide equations of each unit's axle load relative to braking forces and load transfer. From the free-body diagrams, moments about each trailer kingpin are summed according to Equations (7-5) and (7-6). The dolly's fifth wheel hitch is above the axle such that the rear trailer kingpin latches into position directly above or just forward of the dolly's axle and the bulk of vertical force F_{Ky_2} is transmitted directly to the dolly axle rather than the front trailer. However, during a braking event, the longitudinal force from the rear trailer onto the dolly's fifth wheel hitch generates a counter-clockwise moment, increasing the vertical load onto the front trailer's pintle hitch. Because the dolly drawbar is noticeably lower to the ground than the dolly fifth wheel hitch, the clockwise moment from the longitudinal force along the drawbar is less than the counter-clockwise moment mentioned previously. This occurs even though the longitudinal force on the fifth wheel hitch is less than on the drawbar by a magnitude of the vehicle's deceleration and dolly's mass.

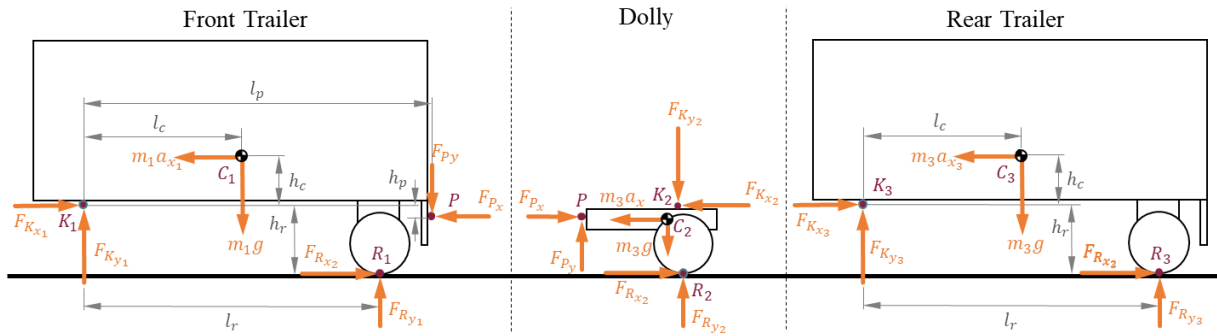


Figure 7-16. Dolly and trailers free-body diagrams during steady braking and constant deceleration.

$$\Sigma M_{K_1} = m_1 a_{x_1} h_c - m_1 g l_c + F_{R_{x_1}} h_r + F_{R_{y_1}} l_r - F_{P_y} l_p - F_{P_x} h_p = 0 \quad (7-5)$$

$$\Sigma M_{K_3} = m_3 a_{x_3} h_c - m g l_c + F_{R_{x_3}} h_r + F_{R_{y_3}} l_r = 0 \quad (7-6)$$

Equations (7-5) and (7-6) can be rearranged into Equations (7-7) and (7-8), respectively. Equation (7-7) shows that the front trailer axle's vertical load decreases from the front trailer's deceleration but increases as the dolly and front trailer transfer load onto the pintle hitch represented by forces F_{P_x} and F_{P_y} . The front

trailer wheels' vertical load could decrease or increase during a braking event depending on each vehicle unit's dimensions, mass, and CG position. Equation (7-8) shows that the rear trailer wheels' vertical load only decreases during the braking event.

$$F_{R_{y_1}} = \left(\frac{1}{l_r}\right) \left(-m_1 a_{x_1} h_c + m_1 g l_c - F_{R_{x_1}} h_r + F_{P_y} l_p + F_{P_x} h_p\right) \quad (7-7)$$

$$F_{R_{y_3}} = \left(\frac{1}{l_r}\right) \left(-m_3 a_{x_3} h_c + m_3 g l_c - F_{R_{x_3}} h_r\right) \quad (7-8)$$

Because the rear trailer's axle is unloaded during braking, we might expect the rear trailer's suspension air pressure to decrease and the spring to extend; however, this was not observed in the test data. Both front and rear trailer suspensions compressed when the brakes were activated. This phenomenon at first appears counterintuitive but can be quickly explained by the suspension's geometry and reaction to brake torque and forces. Figure 7-17 presents free-body diagrams of the suspension trailing arms for drum and disc brake axles assuming steady conditions for a steady straight-line braking event. The angle from the horizontal line denoted 0° to a vector connecting pivot P to some point X is denoted by θ_x and is taken as positive in the clockwise direction. The trailing arm pivots about a rotational joint P in the front and is nominally (no sprung mass pitch or roll) sloped downward to the axle housing A . The suspension force, F_S , pushes the trailing arm downward. For the drum brakes, the torque M_B about the axle housing is the reaction to friction between the brake shoes and rotating drum. Although the leading and trailing brake shoes inside the drum generate slightly different braking forces, for simplicity the reaction forces onto the axle housing are modeled as a pure couple. The torque rotates the trailing arm counter-clockwise as illustrated, pushing upward against the suspension air spring. Similarly, the brake caliper exerts a force F_B where it is attached to the trailing arm as a reaction to the brake torque applied to the wheel. Notice that F_B is tangent to a circle centered at the axle A . The point SP is the string potentiometer's connection to the trailing arm. Point SP is shown only for reference in relating the suspension and axle motion with respect to the string potentiometer's measurement. Note that the suspension force F_S and disc brake force F_B orientations with respect to the trailing arm angle are fixed and thus rotate with suspension jounce and rebound. The axle and pivot forces' orientations are fixed with respect to the inertial frame, as indicated by their subscripts representing x and y directions.

The free-body diagram's forces and moments in Figure 7-17 can be summarized by Equations (7-9) through (7-11) for the drum brakes and Equations (7-12) through (7-14) for the disc brakes. These equations clearly indicate that the suspension force is a function of the vertical load on the axle, brake

torque and forces on the trailing arm, and the reaction forces on the trailing arm's pivot. Even though the sprung mass's forward load transfer during a braking event unloads the rear trailer's axle decreasing F_S , the braking torque and force on the trailing arm increases F_S . The air spring force on the trailing arm is countered by the vertical force on the pivot, the braking torque and force on the trailing arm, and the vertical force on the axle.

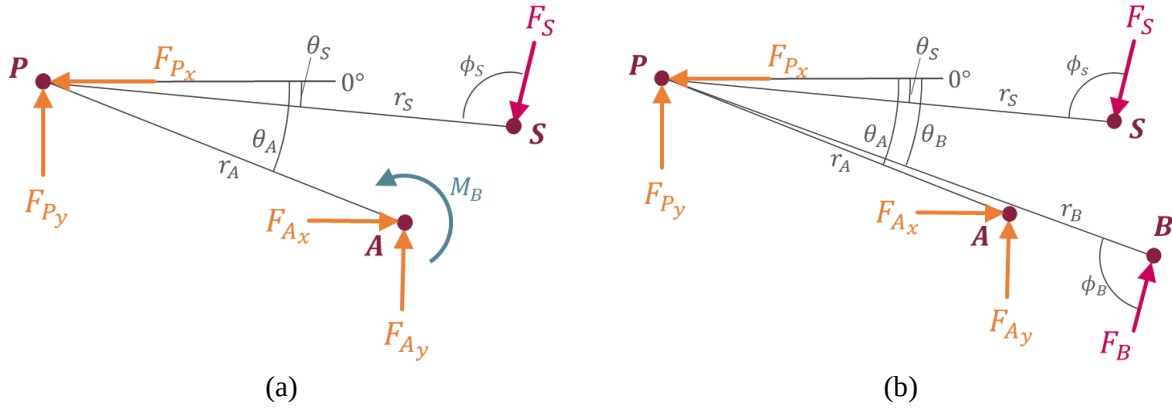


Figure 7-17. Free-body diagrams of trailer suspension trailing arm for an axle with (a) drum brakes and (b) disc brakes.

Relevant force and moment equations for trailing arm with drum brakes as shown in Figure 7-17 (a).

$$\Sigma F_X = F_{Ax} - F_{Px} - F_S \sin(\theta_S + \phi_S - 90^\circ) = 0 \quad (7-9)$$

$$\Sigma F_y = F_{Ay} + F_{Py} - F_S \cos(\theta_S + \phi_S - 90^\circ) = 0 \quad (7-10)$$

$$\Sigma M_P = F_{Ax} r_A \sin(\theta_A) + F_{Ay} r_A \cos(\theta_A) + M_B - F_S r_S \cos(\phi_S - 90^\circ) = 0 \quad (7-11)$$

Relevant force and moment equations for trailing arm with disc brakes as shown in Figure 7-17 (b).

$$\Sigma F_X = F_{Ax} - F_{Px} - F_S \sin(\theta_S + \phi_S - 90^\circ) + F_B \cos(\phi_B - 90^\circ) \sin(\theta_B) = 0 \quad (7-12)$$

$$\Sigma F_y = F_{Ay} + F_{Py} - F_S \cos(\theta_S + \phi_S - 90^\circ) + F_B \cos(\phi_B - 90^\circ) \cos(\theta_B) = 0 \quad (7-13)$$

$$\Sigma M_P = F_{Ax} r_A \sin(\theta_A) + F_{Ay} r_A \cos(\theta_A) + F_B r_B \cos(\phi_B - 90^\circ) - F_S r_S \cos(\phi_S - 90^\circ) = 0 \quad (7-14)$$

Equation (7-11) can be rearranged into Equation (7-15) to equate the drum brake torque to the forces on the trailing arm from the axle loads and suspension air spring. Similarly, Equation (7-14) can be rearranged to Equation (7-17) after substituting Equation (7-16) for the brake force term. Equation (7-16) determines the disc brake torque about the axle where r_{AB} is the radius from the axle center to the brake pad centroid.

$$M_{B_{drum}} = -F_{Ax}r_A \sin(\theta_A) - F_{Ay}r_A \cos(\theta_A) + F_S r_S \cos(\phi_S - 90^\circ) \quad (7-15)$$

$$M_{B_{disc}} = F_B r_{AB} \quad (7-16)$$

$$M_{B_{disc}} = \left(\frac{r_{AB}}{r_B \cos(\phi_B - 90^\circ)} \right) \left[-F_{Ax}r_A \sin(\theta_A) - F_{Ay}r_A \cos(\theta_A) + F_S r_S \cos(\phi_S - 90^\circ) \right] \quad (7-17)$$

Forces in the x-direction on the trailing arm are very small compared to the vertical forces necessary to support the sprung mass for both tested brake types in the configurations illustrated by the free body diagrams in Figure 7-17. Furthermore, angle θ_A is small. From these two points, it is fair to neglect the F_{Ax} term altogether in Equations (7-15) and (7-17). During braking, the longitudinal forces acting on the rear trailer wheel F_{Rx_3} shown in Figure 7-16 is not directly transmitted to the trailing arm in the same direction but is dependent on the location of the brake's reaction force onto the trailing arm. For example, the drum brake is assumed to generate a pure couple about the axle, eliminating a direct reaction force from the friction pair to the trailing arm. The disc brake's reaction force is primarily in the vertical direction due to the caliper's position. If the caliper were directly above the axle, the reaction force would be in the vehicle's traveling direction and not affect suspension dynamics significantly. If the caliper were directly in front of the axle, the reaction force would cause suspension jounce.

Assuming steady deceleration, the vertical force F_{Ay} onto the axles in Equations (7-15) and (7-17) is simply the force F_{Ry_3} from the total vehicle free-body diagram in Figure 7-16 minus the weight of the wheels, tires, and axle (about 1,500 lbf total). The brake torque applied to a wheel determines the longitudinal force along the tire-ground contact patch as

$$F_{Rx} = \frac{M_B - I\alpha}{r_r} \quad (7-18)$$

where M_B is the brake torque, I is the wheels, tires, and axle total moment of inertia, α is the angular deceleration of the rotating components, and r_r is the tire rolling radius [31]. F_{Rx} in Equation (7-18)

corresponds to $F_{R_{x_1}}$ and $F_{R_{x_3}}$ in Equations (7-7) and (7-8), respectively. Using the same axles, tires, and wheels, for the two brake types, terms I , α , and r_r are unchanged.

To best demonstrate the brake type and position effect on suspension dynamics, let's compare the two brake types for identical trailers slowing at the same steady deceleration. By virtue of the same vehicle mass, geometry, inertial properties, and assumed constant deceleration, the longitudinal force along the tire contact patch must be the same (i.e.: $F_{R_{x_{drum}}} = F_{R_{x_{disc}}}$ from Equations (7-7) and (7-8)), resulting equal brake torques $M_{B_{drum}}$ and $M_{B_{disc}}$ from Equation (7-18). Although the brake torques and vehicle deceleration may be the same for the two brake types, the suspension dynamics will defer. This is determined by the factor $\left(\frac{r_{AB}}{r_B \cos(\phi_B - 90^\circ)}\right)$ that is less than one in Equation (7-17) and not part of Equation (7-15). Thus, the air spring force must be higher for the disc brakes than the drum brakes for the same brake torque. This conclusion in tandem with the disc brake's higher brake torque demonstrated by shorter braking distances clearly explains why suspension jounce was greater when the trailers were equipped with disc brakes than drum brakes. Figure 7-18 presents the air spring pressures for the drum and disc brakes on the front and rear trailers during a 65 mph stopping distance tests to illustrate this conclusion.

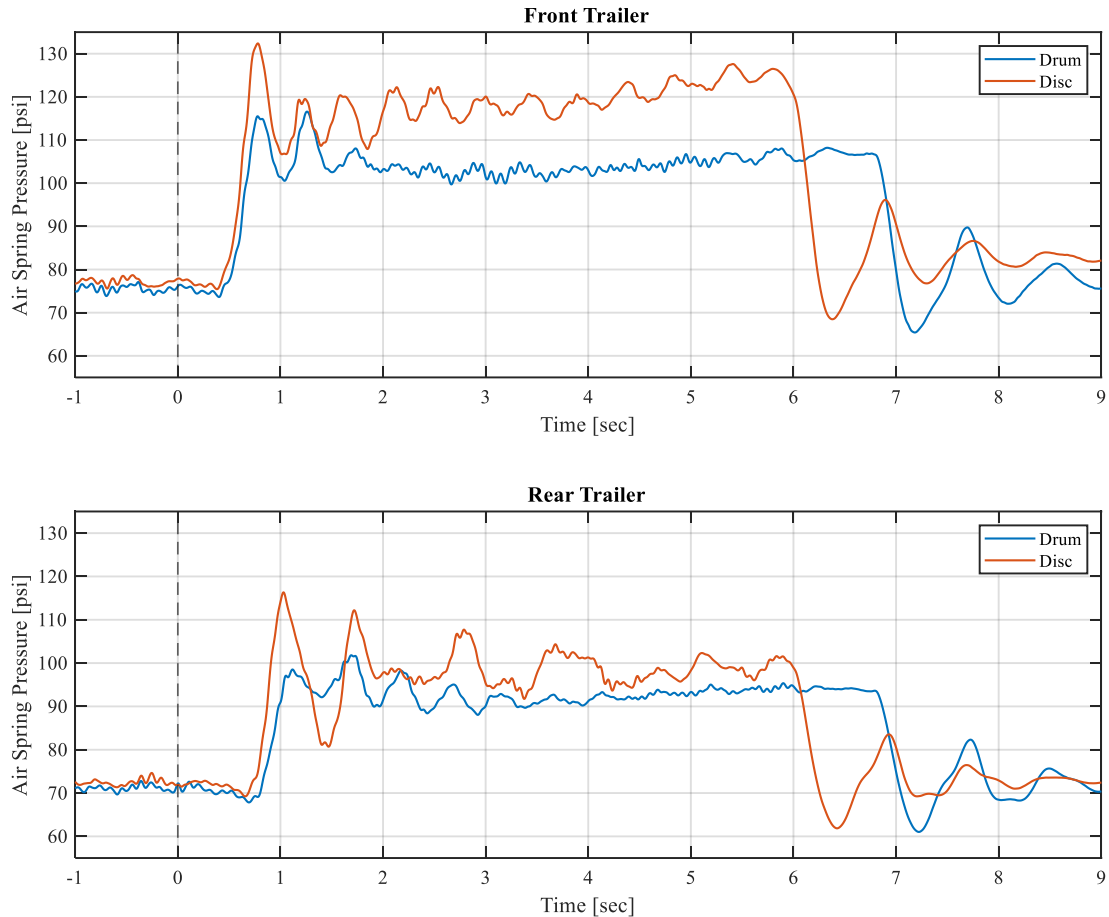


Figure 7-18. Front and rear trailer air spring pressures for drum and disc brakes during 65-mph straight-line braking.

7.2 Braking-in-turn Maneuver Testing

The objective of this study was to test if the different trailer brake types provide significantly different amounts of time for a driver to initiate braking to prevent trailer rollover during a turn. Tests were conducted using both steering and braking robots with specified time delays of brake initiation relative to the start of steering. The braking-in-turn maneuver shares the same steering wheel angle profile as the Robot J-turn, but instead of increasing the entry speed by 1-2 mph until a rollover event, the driver enters the maneuver at the same speed and the braking robot delay relative to the start of steering was increased. Brake delays were tested in 0.125 sec increments at two different brake chamber pressure levels that are labeled in this document as full and partial. Just as in the straight-line braking distance test, full braking was achieved by the robot's pneumatic actuator pushing and holding the brake pedal to its maximum stroke. Partial braking was achieved by limiting the braking robot actuator stroke via a hard stop. Brake chamber air pressure during partial braking was approximately 40% of the full-braking brake chamber air

pressure as shown earlier in Figure 5-31. Entry speeds were tested at 35, 40, and 45 mph for both full and partial braking as well as 50 mph for full braking. The north loop route (shown earlier in Figure 7-3) was taken between trials to cool down the brakes and was also necessary to get the vehicle up to speed for some high-speed trials. The tractor's ESC and both trailers' RSCs were disabled. ABS was active on all axles. At the time of these tests, available track time was very limited, so the total number of trials at each test speed and brake delay are scarce compared to straight-line braking tests.

7.2.1 Results Analysis of Full-Braking J-turn

Figure 7-19 shows the brake delays that resulted in measurable outrigger contact forces with respect to different entry speeds for the full-braking J-turn tests. The bubble area is proportional to the peak outrigger contact force. A larger bubble area represents a higher contact force and a more severe rollover event (higher likelihood of actual complete rollover). As the brake delay (relative to the start of steering) increases for a given entry speed, the outrigger contact force increases. Outrigger contacts for the drum and disc brakes occur around the same entry speed and brake delay with approximately the same sized bubble marker.

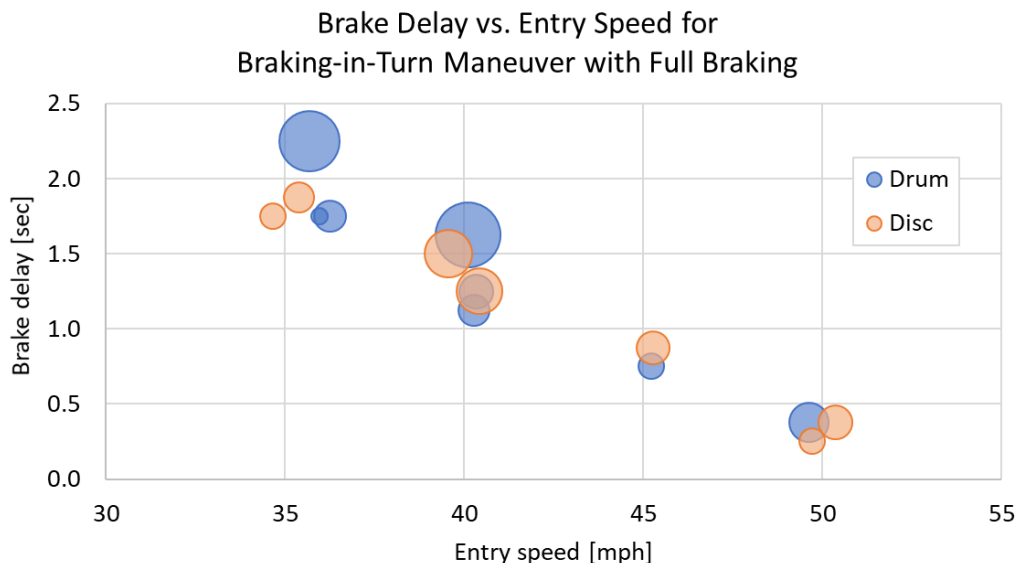


Figure 7-19. Tested brake delay vs entry speed for the braking-in-turn maneuver with full braking (the bubble areas are proportional to the peak outrigger contact forces).

Figure 7-20 presents the full-braking J-turn rollover conditions for brake delays at and before the rollover threshold was met for each entry speed. Trials with entry speeds more than 1 mph above or below the targeted speed are omitted. Trials with brake delays well above the brake delay that met the rollover

condition are also omitted for brevity. Hard outrigger contacts are designated by peak contact forces above 400 lbf, under which condition the trailer is very likely to rollover. Light outrigger contacts have measurable contact forces but are below the 400 lbf threshold for rollover. Kiss outrigger contacts occur when the outrigger wheel contacts the ground, but the impact force is below the instrumentation noise floor. The trailer is extremely unlikely to rollover under this condition. Lastly, Figure 7-20 also lists when the front trailer left side wheel lifts off the ground but there is no outrigger contact (no rollover). At 35, 45, and 50 mph, the drum and disc brakes met the rollover criteria (hard outrigger contact) for the same brake delay. At 40 mph, both brake types resulted in outrigger contact at 1.125 sec brake delay. The drum brakes resulted in hard outrigger contact whereas the disc brakes resulted in a kiss; however, the disc brakes' entry speed was more than 1 mph slower than the targeted 40 mph. If the entry speed was correct, the disc brakes likely would have resulted in a hard outrigger contact. This is noted in the figure. From the test data, the type of trailer brake does not appear to provide more than a 0.125 sec window for driver response to begin braking with full brake pressure through a turn to avoid trailer rollover. In short, the drum and disc brakes performed similarly. This result coincides with the straight-line braking distance tests for speeds up to 50 mph.

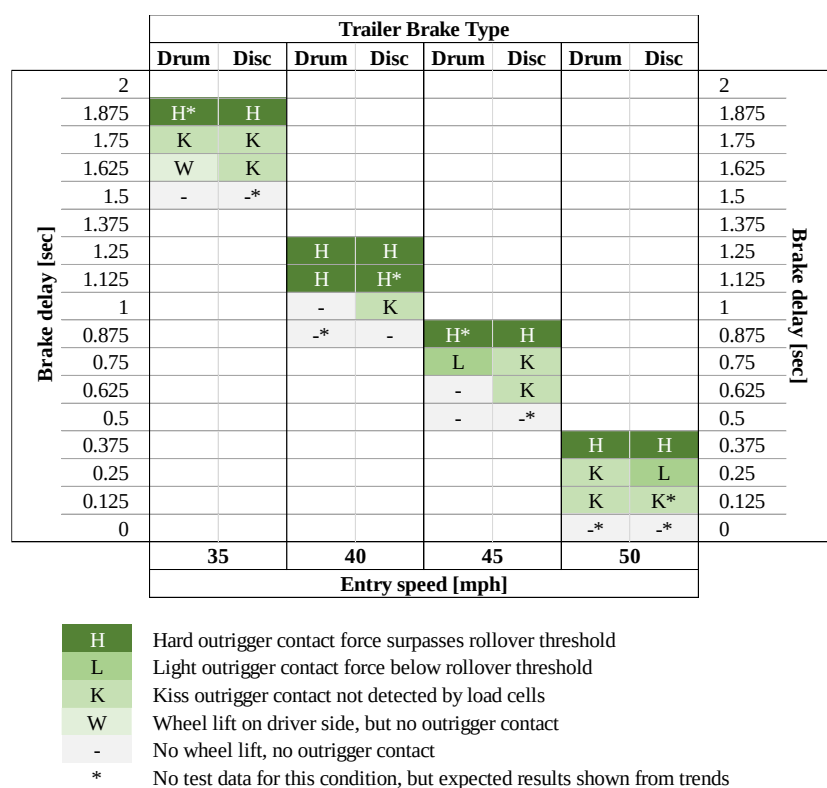


Figure 7-20. Summary of outrigger contacts for full-braking J-turn maneuver.

Figure 7-21 presents ABS activation time delay from brake pedal application during full-braking J-turn tests. The ABS activation time delay increases from the tractor to the trailers because the trailer brakes are activated much later than the tractor due to pneumatic transport delay and resistance in the air brake system. The average difference (drum minus disc) in ABS delay between the two brake types is 0.004, 0.017, and 0.216 sec for the tractor, front trailer, and rear trailer, respectively. The tractor's difference is nearly zero because it uses the same drum brakes for all testing. The front trailer's difference is very small because the roll dynamics dominate the inside tire's loading and, thus, braking capability. ABS activates if at least one wheel begins to slip on a given axle but modulates the brake pressure for each side independently. The disc brakes initiate ABS slightly sooner than the drum brakes on the rear trailer during the full-braking J-turn maneuver. This is likely because the rear trailer does not experience enough lateral acceleration to cause wheel lift and because the disc brakes are quicker to apply higher braking torque as demonstrated in the straight-line braking test.

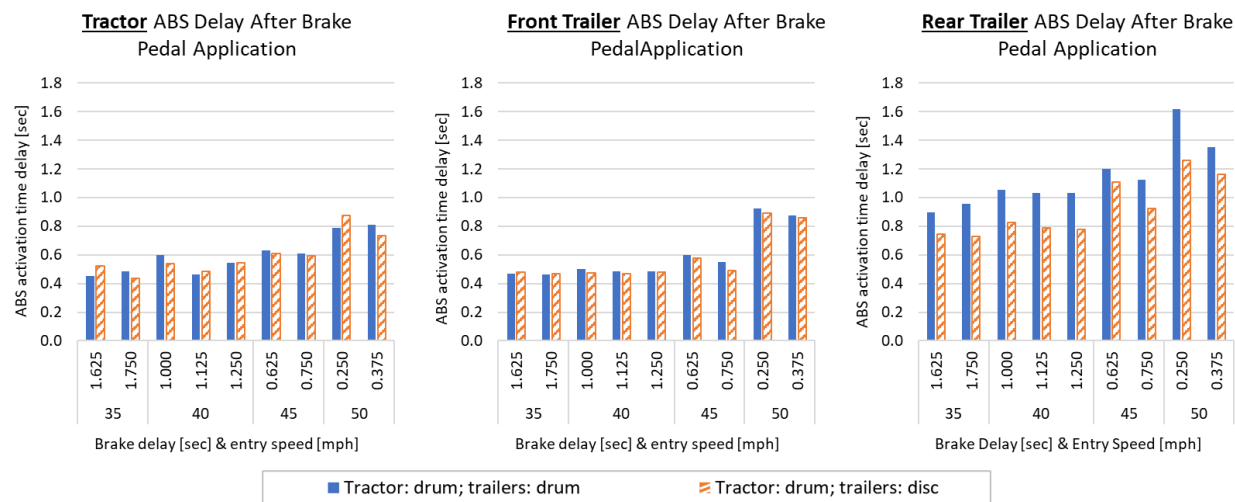


Figure 7-21. ABS activation time delay from brake pedal application during full-braking J-turn tests.

7.2.2 Result Analysis of Partial-Braking J-turn

The partial-braking J-turn tests were conducted similarly to the full-braking J-turn tests at entry speeds of 35, 40, and 45 mph. Figure 7-22 compares the severity of outrigger contact for the drum and disc brake systems for different tested entry speeds and brake delays. The bubble area is proportional to the outrigger contact force. At 40 and 45 mph, both brake types had outrigger contact at around the same brake delay, but some of the differences between brake types may have been due to slight differences in the entry speeds. At the 35-mph entry speed, the trailers with the disc brakes rolled over at a shorter brake delay than the drum brakes.

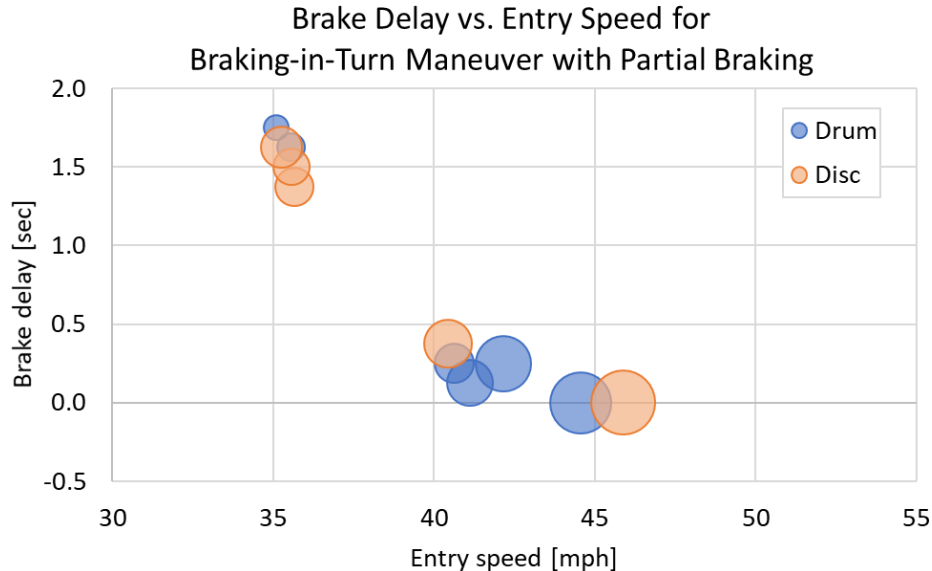


Figure 7-22. Tested brake delay vs entry speed for the braking-in-turn maneuver with partial braking (the bubble areas are proportional to the outrigger contact forces).

Figure 7-23 presents the partial-braking J-turn rollover conditions for brake delays at and before the rollover threshold was met for each target entry speed. Trials which had entry speeds more than 1 mph above or below the target speed are omitted. For the case of 35 mph, a noticeable difference occurred between trailer brake types. The disc brakes first made outrigger contact with a 1.375 sec brake delay whereas the drum brakes' first outrigger contact was at 1.625 sec brake delay. The disc brake's first rollover occurred at the 1.5 sec delay for the disc brakes and are projected to be 1.875 sec for the drum brakes. Opposite to the 35 mph entry speed tests, the drum brakes met the rollover condition at a shorter brake delay than the disc brakes for the 40 mph entry speed tests. The first outrigger contacts for both brake types occurred at 0.25 sec. At 45 mph and 0 sec brake delay, both trailer brake types experienced hard outrigger contacts. The rear trailer right side outrigger also contacted the ground for both trailer brake types for the 45-mph entry speed and 0 sec brake delay condition. This was the only tested condition in the braking J-turn turn tests where a rear trailer outrigger contacted the ground. The difference in brake delays that resulted in hard outrigger contacts for the brake types may be largely due to differing nonlinear properties of each brake type mechanism's conversion of brake chamber air pressure into braking torque. For these reasons and because the order of brake delays with hard contact for the brake types was not consistent among the different entry speeds, there is not sufficient evidence to suggest that one trailer brake type will consistently provide more time before the trailer rollover under partial braking.

		Trailer Brake Type							
		Drum	Disc	Drum	Disc	Drum	Disc		
Brake delay [sec]	2							2	Brake delay [sec]
	1.875	H*						1.875	
	1.75	L						1.75	
	1.625	L	H					1.625	
	1.5	-	H					1.5	
	1.375	_*	L					1.375	
	1.25	-	-					1.25	
	1.125							1.125	
	1							1	
	0.875							0.875	
	0.75							0.75	
	0.625							0.625	
	0.5							0.5	
	0.375			H*	H			0.375	
	0.25			H	K			0.25	
0.125			W	-			0.125		
0			_*	_*	H	H	0		
		35	40		45				
		Entry speed [mph]							

H	Hard outrigger contact force surpasses rollover threshold
L	Light outrigger contact force below rollover threshold
K	Kiss outrigger contact not detected by load cells
W	Wheel lift on driver side, but no outrigger contact
-	No wheel lift, no outrigger contact
*	No test data for this condition, but expected results shown from trends

Figure 7-23. Summary of outrigger contacts for partial-braking J-turn maneuver.

To further investigate the effect of trailer brake type on the rollover of 33-ft A-doubles for the partial-braking J-turn maneuver, the front and rear trailer lateral accelerations for a 35-mph entry speed and 1.25-sec brake delay are analyzed, as shown in Figure 7-24. This brake delay was chosen because it was tested just below the first outrigger contact for either brake type. The lateral acceleration was nearly identical between brake types for both trailers, suggesting similar performance in the brake types for the shown trials. This finding coincides with the straight-line braking tests for low speeds (<50 mph). Additionally, trailer roll dynamics during the braking-in-turn maneuvers impact the maximum braking capability for a trailer axle, blurring the difference in braking performance between the two trailer brake types.

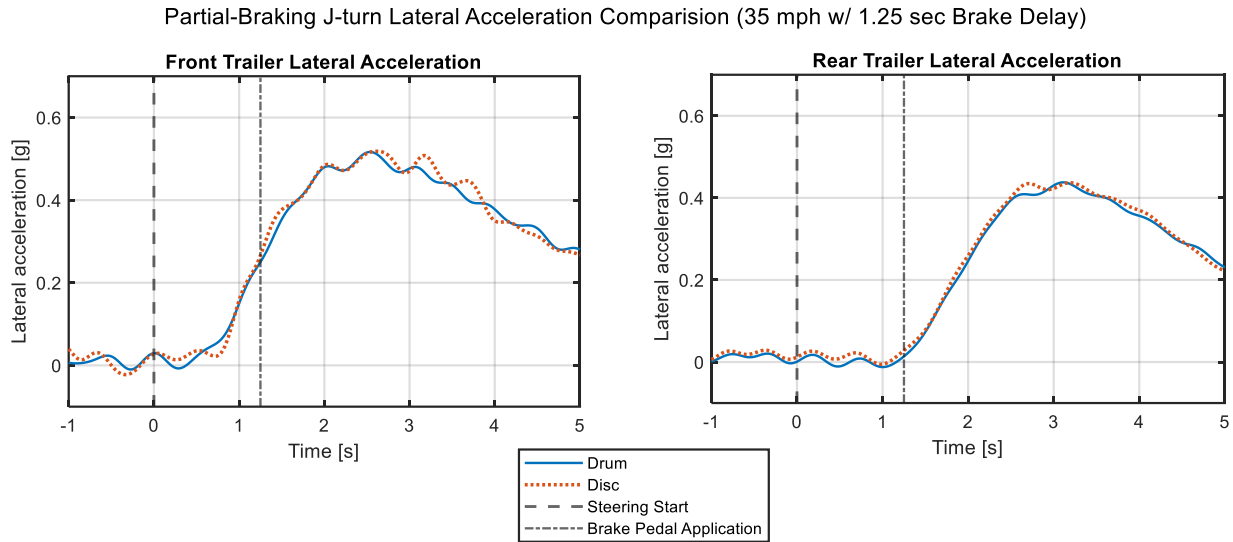


Figure 7-24. Lateral acceleration comparison of drum and disc trailer brakes for partial-braking J-turn maneuver with 1.25 sec brake delay and 35 mph entry speed.

Figure 7-25 presents the ABS activation time delay after the brake pedal is applied for the partial-braking J-turn tests. Just as the same as the full-braking tests, the ABS delay increases from the tractor to the front trailer then to the rear trailer due to the actuation delay of the pneumatic braking system. The ABS time delay with respect to brake pedal application also decreases for an increase in brake delay at a given entry speed for the rear trailer. At the point in time when the rear pneumatic brakes receive pressurized air, the rear trailer has a higher lateral acceleration that causes unloading of the inside wheel. As mentioned before, this unloading causes ABS intervention on the wheel inside of the turn to prevent the wheel from locking up. The average difference in delay for the two brake types is less than 0.055 sec for all vehicle units. This is very small, especially when compared to the rear trailer during the full-braking J-turn tests, because the roll-dynamics of both trailers and tractor dominate the braking capability and ABS activation of the wheels on the inside of the turn.

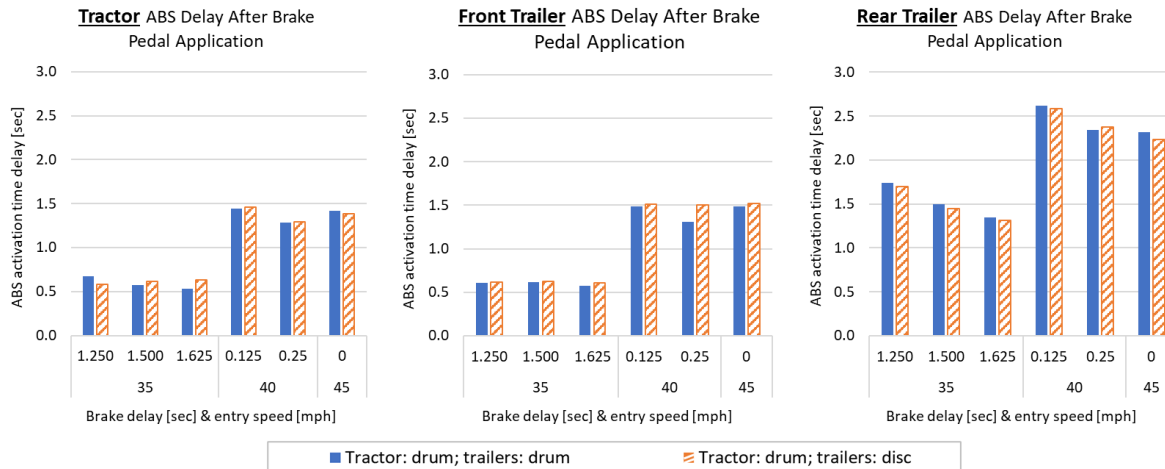


Figure 7-25. ABS activation time delay from brake pedal application during partial-braking J-turn tests.

7.2.3 Conclusions from Braking-in-turn Maneuvers

In summary, the full-braking and partial-braking J-turn tests did not present a substantial difference in the brake type's performance. Wheel tip-up is the predominant factor for ABS activity in most tests. Similarities in outrigger contacts for given brake delays and entry speeds between the brake types for both the partial-braking and the full-braking J-turn tests are largely due to the tractor's braking contribution and the trailers' limited braking capability due to roll dynamics unloading the inside wheel. Some inconsistencies in the test results, particularly the 35 mph entry speeds with partial braking, are best estimated to be caused by environmental factors and are not directly indicative of the different brake type performances. These findings coincide with the low-speed (50 mph and less) braking distance tests.

7.3 Trailer Brake Type Effect on RSC Performance

The purpose of this study is to evaluate how the different trailer brake types impact the brake-based RSC system performance. The same four maneuvers (Driver J-turn, Robot J-turn, Double Lane Change, and 0.25-Hz Sine with Dwell) used to evaluate the rollover propensity for different stability control implementations were repeated with disc brakes in place of drum brakes on the heavily loaded 33-ft straight-rail trailers. The scheme to determine the rollover threshold for each maneuver used here is the same as in Chapter 6 (Table 6-1).

7.3.1 Robot J-turn

For a left-hand J-turn maneuver, the front trailer's right-side outrigger is first to contact the ground during a rollover event for the tested trailer and loading configuration. Figure 7-26 shows the peak outrigger

contact force for both brake types at the tested entry speeds. The trailers with drum brakes had a rollover speed of 34 mph while the trailers with disc brakes had a rollover speed of 35 mph. At 33 mph with the drum brakes, there was one outrigger contact force hard enough to be considered as a high likelihood of a rollover. However, because this only occurred for one of the three trials at 33 mph the rollover speed for the drum brakes was determined to be 34 mph.

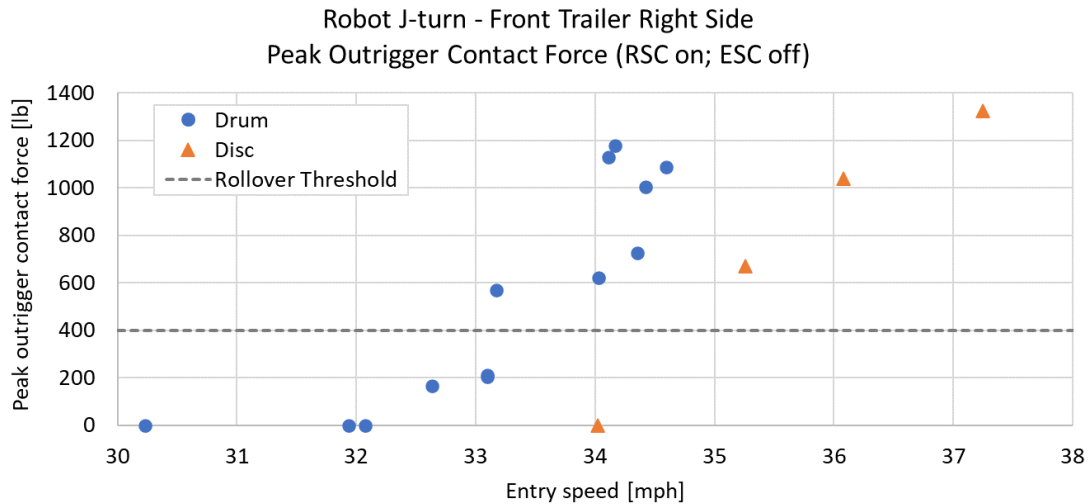


Figure 7-26. Peak outrigger wheel contact force for Robot J-turn with RSC on for both tested trailer brake types

7.3.2 Driver J-turn

Similar to the results of the Robot J-turn maneuver, the only outrigger contacts during the Driver J-turn tests occurred at the front trailer's right-side outrigger. Figure 7-27 shows the peak outrigger contact forces at different entry speeds. Both trailer brake types had a rollover speed of 35 mph. The trailer with drum brakes experienced a hard outrigger contact for one of three trials at 34 mph (two of the trials had the same entry speed at 34.1 mph, making them appear as a single data point in Figure 7-27), but two out of three trials at 35 mph had hard outrigger contacts. The disc brakes resulted in outrigger contact for entry speeds of 35 mph.

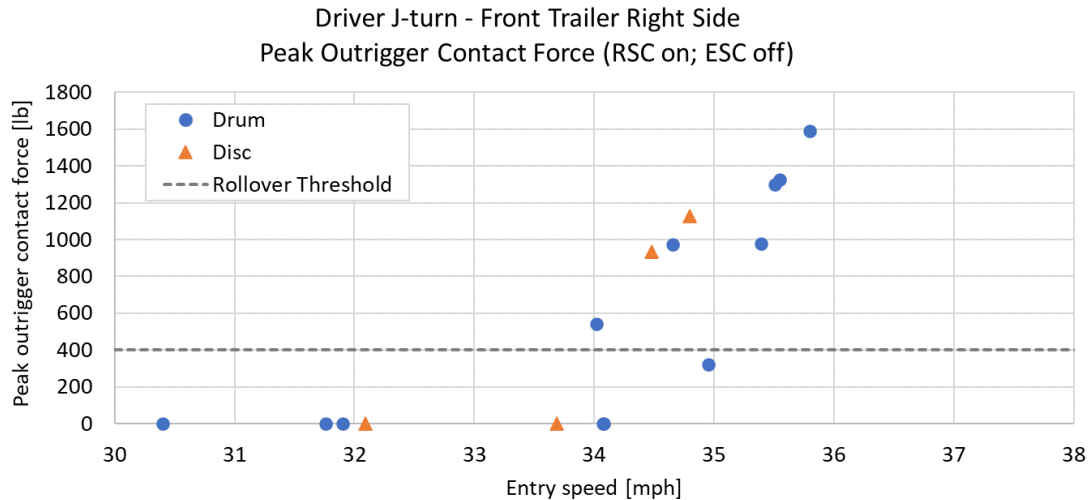


Figure 7-27. Peak outrigger wheel contact force for Driver J-turn with RSC on for both tested trailer brake types.

7.3.3 Double Lane Change (DLC)

The Double Lane Change maneuver has a more complex steering profile compared to the J-turn maneuvers and often had multiple outrigger contacts rather than just one. Figure 3-24 shows the peak outrigger contact forces with respect to entry speeds for the left side outriggers on the front and rear trailers. The front trailer right side outrigger did not make contact for either brake type for any tested entry speed. The front trailer left side outrigger was typically the first outrigger to contact the ground, however, due to rearward amplification, the rear trailer left side outrigger contact force was consistently much higher than the front trailer left side outrigger contact force. As detailed in Section 6.5.3 and shown in Table 7-2, the rollover speed for the drum brakes was 46 mph. Because only one of two trials at 47 mph with the disc brakes resulted in a hard outrigger contact just above the force threshold whereas the contact force at 48 mph was much higher than the contact force threshold, the rollover speed for the disc brakes was 48 mph. The disc brakes resulted in a 2 mph higher rollover speed than the drum brakes and the outrigger contact forces were much smaller for the disc brakes at the respective rollover speeds. Thus, the results suggests that the disc brakes performed better than the drum brakes through the DLC maneuver with RSC.

Table 7-2. Peak outrigger wheel contact forces for DLC with RSC on for both tested trailer brake types.

Stability Control	Entry Speed [mph]	Peak Outrigger Contact Force [lbf]			
		Front Left	Front Right	Rear Left	Rear Right
Drum	45	0	0	0	0
	46*^	-	-	-	-
	47	640	0	1825	K
	47	406	0	1635	0
Disc	46	0	0	0	0
	47	0	0	0	0
	47	0	0	472	0
	48*	K	0	1025	540
	49	1210	0	2628	1904
<i>yellow</i> = below threshold (<400 lbf contact force) <i>red</i> = above threshold (≥400 lbf contact force) K = kiss: contact occurred but force is too small to be detected by instrumentation * = rollover steering percentage ^ = no test performed					

7.3.4 Sine with Dwell (SWD)

For the SWD maneuver, some steering percentages cause all four outriggers to contact the ground during the same trial, as shown in Table 7-3. When equipped with drum brakes, both trailers clearly experienced a rollover condition at 65% steering. When equipped with disc brakes, only the front trailer experienced a rollover condition at 70% steering. Although no tests were conducted at 65% steering for the disc brakes, it is expected that if outriggers were to contact the ground during a 65% steering test, the contact force would be below the rollover threshold and thus be very unlikely for a trailer to rollover. This is because the highest contact force for the disc brakes at the 70% steering percentage was approximately 600 lbf which is relatively close to the rollover threshold and well below the contact forces from the drum brakes.

Table 7-3. Peak outrigger wheel contact forces for SWD with RSC on for both tested trailer brake types.

Stability Control	Steering Percentage	Peak Outrigger Contact Force [lbf]			
		Front Left	Front Right	Rear Left	Rear Right
Drum	50%	0	0	0	0
	60%	0	0	0	159
	65%*	1182	476	1479	868
	65%*	1121	162	888	641
	65%*	1337	219	1530	832
Disc	60%	0	0	0	0
	65%^	-	-	-	-
	70%*	292	635	0	246
	75%	899	1250	1336	1274
	75%	952	1358	966	1029
<i>yellow</i> = below threshold (<400 lbf contact force) <i>red</i> = above threshold (≥400 lbf contact force) K = kiss: contact occurred but force is too small to be detected by instrumentation * = rollover steering percentage ^ = no test performed					

7.3.5 Comparing Brake Type Effect on Rollover Speed & Deceleration with RSC On & ESC Off

The rollover speed and steering percentage for the four test maneuvers are summarized in Table 7-4 for both brake types. In general, the disc brakes performed better than the drum brakes and amplified the benefits of RSC in improving the roll stability of the vehicle. The rollover speed and steering percentage are higher for the disc brakes than the drum brakes for all maneuvers except the Driver J-turn where both brake types exhibited the same rollover speed. Despite higher entry speeds or steering percentage, the rollover severity in terms of peak outrigger contact force was typically less for the disc brakes than the drum brakes. This further illustrates the performance benefits of trailer disc brakes over drum brakes.

Table 7-4. Rollover speed and steering percentage comparison between drum and disc brake systems for the four tested maneuvers with RSC on & ESC off.

	Robot J-turn	Driver J-turn	DLC	SWD
Drum	34 mph	35 mph	46 mph	65%
Disc	35 mph	35 mph	48 mph	70%
Improvement with Disc Bakes	+1 mph	+0 mph	+2 mph	+5%

The rollover threshold with RSC on and ESC off for the disc brakes is typically higher than that for drum brakes because the disc brakes provide slightly quicker braking response and higher braking torque as demonstrated in the straight-line brake tests. This is confirmed in both the disc brakes' higher average deceleration and larger reduction of lateral acceleration while RSC is active. Figure 7-28 plots the spread of average decelerations while RSC is active on either or both front and rear trailers through all maneuvers. This data is determined by the GPS's measured speed reduction from the start of RSC intervention to the end. The average and median decelerations of the disc brakes are both 0.02g higher than those of the drum brakes.

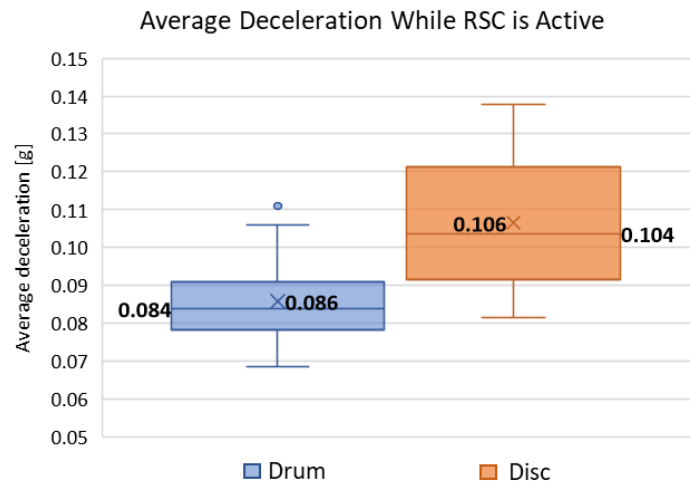


Figure 7-28. Box and whisker plot of average deceleration while RSC is active on either or both trailers for all maneuvers.

Because the disc brakes slow down the vehicle slightly faster than the drum brakes, the trailers' lateral acceleration amplitudes are slightly lower for the same maneuver and entry speed. This difference is illustrated in Figure 7-29 that plots the front and rear trailer lateral acceleration for the SWD maneuver with 60% steering. Notice that the difference in lateral acceleration amplitude increases through the maneuver as the disc brakes slow down the vehicle faster than the drum brakes. By midway through the second turn at about 4 and 5 seconds, there is a clear reduction in lateral acceleration amplitude of trailers with disc brakes compared to trailers with drum brakes.

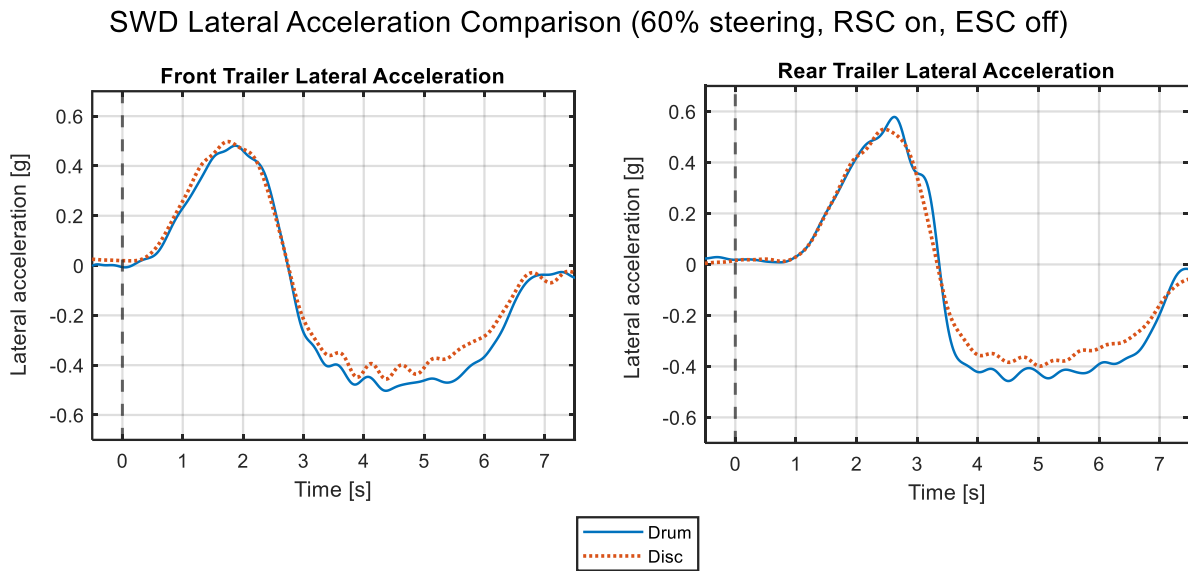


Figure 7-29. Lateral acceleration comparison of drum and disc trailer brakes for 0.25 Hz SWD maneuver for 45 mph entry speed and 60% steering.

7.3.6 Comparing Brake Type Effect on Rollover Speed & Deceleration with Both RSC & ESC On

The four maneuvers used to evaluate roll-stability were tested once again with trailer drum and disc brakes. This time, however, both the tractor ESC and trailer RSC systems were active. The same scheme was used for determining the rollover threshold as before. Although the rollover threshold increased when testing with RSC on and ESC off with trailer disc brakes in place of drum brakes, difference in rollover speed for the two trailer brake types is not substantial when both RSC and ESC are active. For example, both brake types resulted in the same rollover speed for the driver and robot J-turns when RSC and ESC were active. The disc brakes performed slightly better in the DLC maneuver than the drum brakes, however that difference is predominately attributed to changes in driver steering behavior. Unfortunately, the front trailer's RSC was inactive when testing the SWD maneuver with drum brakes, so a fair comparison cannot be made from this maneuver as it would under-report the drum brake's effectiveness. In general, because the tractor ESC provides some braking force to slow down the vehicle and is typically the first to activate because the tractor is the first vehicle unit to enter the maneuver, the tests did not detect a substantial and consistent difference in the trailer brake types in terms of increasing the rollover speed or steering percentage when both trailer RSC and tractor ESC are active. This conclusion coincides with the braking-in-turn test results. The rollover thresholds from the two trailer brake types in the four maneuvers with RSC and ESC active are listed in Table 7-5. Note that the front trailer RSC was not active during the SWD maneuver with drum brakes.

Table 7-5. Rollover speed and steering percentage comparison between drum and disc brake systems for the four tested maneuvers with RSC on & ESC on.

	Robot J-turn	Driver J-turn	DLC	SWD
Drum	37 mph	37 mph	47 mph	65%*
Disc	37 mph	36 mph	50 mph	75%
Improvement with Disc Bakes	+0 mph	-1 mph	+3 mph	n/a
*Front trailer RSC was off				

7.4 Conclusions on Trailer Brake Type Performance Comparison

In the straight-line braking tests, the disc brakes outperformed the drum brakes in both braking distance and repeatability. The disc brakes generated higher braking torque resulting higher suspension compression and more frequent and longer ABS intervention than the drum brakes. The disc brake rotors tended to heat up quicker and to higher temperatures than the drum brake drums. For three trials of each entry speed the braking distance increased significantly in succession for the drum brakes, but negligibly for the disc brakes, suggesting that the drum brakes are more susceptible to brake fade than drum brakes.

During braking-in-turn maneuver tests, there was not a significant nor consistent difference in brake activation delay that caused rollover for the two brake types in partial and full braking. This is primarily caused by the tractor and dolly's braking contribution, muddying the difference between the trailers' braking contributions. Trailer ABS intervention was dominated by the trailer roll dynamics rather than brake torque and did not exhibit a clear difference in ABS intervention or delay for the two brake types for braking-in-turn maneuvers.

The disc brakes increased the roll-stability benefits of trailer RSC as compared to the trailer drum brakes. When RSC was on and ESC off, a small increase in rollover threshold was observed for three of the four test maneuvers. When both RSC and ESC were on, the rollover threshold difference between trailer brake types was small and inconsistent for the test maneuvers.

In conclusion, the trailer disc brakes outperformed the drum brakes in both straight-line braking and RSC effectiveness. At lower speeds while the tractor provides braking such as during straight-line braking (≤ 50 mph), braking-in-turn maneuver, and RSC+ESC tests, the difference between brake type performance is unclear.

Chapter 8

Conclusions and Recommendations

8.1 Summaries and Conclusions

8.1.1 Low-Speed Off-tracking & Maneuverability

A vehicle's off-tracking through a maneuver is indicative of its maneuverability. Increasing vehicle length or reducing turning radius increases off-tracking. Increasing the number of articulation points reduces off-tracking. Long combination vehicles exhibit large off-tracking, requiring significantly more space to negotiate a turn than passenger vehicles. Long combination vehicles' off-tracking has been well-documented in simulation and modeling studies, but few include physical tests and almost none explicitly detail a field test procedure for measuring the uncertainties. Furthermore, off-tracking of double 33-ft trailers had not yet been considered.

As detailed in Chapters 3 and 4, a study was conducted to physically test and measure off-tracking of three common long combination vehicles – single 53-ft, double 28-ft, and double 33-ft trailers – through four real-world maneuvers, namely 50-ft J-turn, 75-ft J-turn, left-turn through a roundabout, and a right-turn through an intersection. To accurately estimate the tractor's orientation and dolly's contribution to the trailer-trailer articulation for the double trailer configurations, a numerical algorithm was developed to model low-speed (<5 mph) motion of a long combination vehicle when tire lateral slip is negligible on trailing axles. This algorithm emulates tractrix curves without needing to solve differential equations. Tractrix curves are well documented for accurately representing low-speed vehicle motion. The algorithm inputs include the steer axle path as a series of cartesian coordinates as opposed to an arch length and curvature suggested in some studies. The algorithm was proven to accurately capture tractrix curves in a robust manner in large and irregular spacing between the input path indices, making it easy to use for field data. The algorithm was then used to determine the peak off-tracking sensitivity to measurement errors in the field tests' procedures and sensors. The results indicate that the peak off-tracking estimate is most sensitive to the tractor's orientation and tractor-trailer articulation angle measurement. The lateral and forward position errors do not impact off-tracking estimates as greatly as the vehicle orientation and articulation angles.

By combination of field measurements and simulation models, the peak off-tracking for each configuration and maneuver was determined. For all tests, the order from smallest to largest off-tracking proved to be double 28-ft, double 33-ft, and single 53-ft trailer configurations. Although double trailers have a longer

total length than single 53-ft trailers, their added articulation points reduce off-tracking. The results agree with both high-order dynamic models for low-speed maneuvers and simple geometric models for a steady-state curve. In summary, a double 33-ft trailer configuration is expected to be able to negotiate any turn that a single 53-ft trailer can. The 33-ft doubles, however, would require slightly larger turning space than 28-ft doubles.

8.1.2 Rollover Propensity and Stability Control Systems

In addition to the stock vehicle roll-stability, three stability control system combinations were evaluated for their effectiveness in reducing rollover propensity of heavily loaded 33-ft straight-rail doubles. The four configurations included: stock (no stability control system), trailer RSC, tractor ESC, and RSC+ESC. For the RSC configurations, both trailer RSC systems were on. The trailer roll-stability was tested in four unique maneuvers consisting of two with driver steering and two with robot steering. The three stability control systems (RSC, ESC, and RSC+ESC) reduced trailer rollover propensity by braking wheels to reduce vehicle speed and, ultimately, reduce the lateral accelerations in the curves. The tested stability control systems did not communicate with other vehicle units and could only activate the installed unit's brakes. Because the tractor is first into a maneuver, the ESC activates first, substantially reducing speed. This resulted in a higher rollover speed/steering for ESC alone than RSC alone in three of the four maneuvers. RSC+ESC resulted in the highest rollover speed and steering, while the stock yielded the lowest. As expected, the highest average deceleration during any stability control intervention was observed for RSC+ESC for all maneuvers. The ESC alone exhibited similar but slightly higher deceleration than the RSC alone for three of the four maneuvers. The stability control systems impacted vehicle directional stability as indicated by the changes in peak articulation angles. The ESC alone tended to effect articulation angles more than the RSC alone. Some key findings are summarized as follows:

- RSC, ESC, and RSC+ESC reduced vehicle speed to reduce rollover propensity compared to stock (no stability control).
- RSC+ESC was most effective at reducing trailer rollover propensity.
- Tractor ESC influenced the articulation angles between vehicle units more than RSC alone or RSC+ESC.

8.1.3 Trailer Brake Type Performance Evaluation

Even though it is well documented in the literature that disc brakes tend to outperform similarly-sized drum brakes for 53-ft trailers, most heavy vehicle trailers in the United States are equipped with drum

brakes. The studies detailed in Chapter 7 were designed to quantify the improvements in stopping distance resulting from disc brakes for 33-ft straight-rail doubles. This study also extends past braking studies to braking-in-turn and to evaluate any effect that disc brakes can have on RSC performance.

The first study tested straight-line braking distance of the heavily-loaded double 33-ft straight-rail trailer configuration for trailer drum and disc brakes. The tractor and dolly brakes were drum type and unchanged throughout testing. A pneumatic actuator with spatial triggering provided accurate and repeatable brake pedal application. A LiDAR unit measured the vehicle's stopped position from a fixed target at a known distance from the start of braking. The entry speeds were from 20 mph to 65 mph. Each target speed was tested three times. The target speeds were alternated between the slowest to fastest after each set to prevent significant brake fade (ex: 20, 20, 20, 65, 65, 65, 30 mph...). Multiple cool-down laps were systematically introduced between high-speed trials and between all test sets to keep the brakes within normal operating conditions. Immediately after each set of three trials, brake temperatures were measured at each wheel.

For speeds greater than 50 mph, the disc brakes exhibited shorter and more consistent straight-line stopping distances. For speed ≤ 50 mph, there was not a significant difference in the braking performance. The disc brakes exhibited more consistent braking distance, even for tests that the entry speed varied by up to 1 mph from the target speed. The disc brake's higher braking torque resulted in more frequent and longer ABS intervention than the drum brakes. Similarly, the suspension jounce due to reaction to larger disc brake forces was greater than from the drum brakes. Beyond the higher disc brake forces, the relative location of the brake calipers to the suspension trailing arm also contribute to the larger jounce.

The second study tested the trailer's rollover propensity during a braking-in-turn maneuver. The tests were conducted with full- and partial-braking pedal application. A steering robot provided precise steering and accurate control of brake delay from the start of steering for a J-turn. The tests were conducted at 35, 40, 45, and 50 mph for full-braking and 35, 40, and 45 mph for partial-braking. Partial braking was achieved by limiting the brake pedal stroke, resulting in approximately 40% brake chamber pressure relative to full-braking. The brake application delay was tested in increasing 0.125 sec increments until a trailer rollover event occurred. The rollover likelihood was assessed according to the magnitude of outrigger contact forces. Largely due to the tractor and dolly's braking contribution, there was not a clear difference in brake delay that resulted in rollover for the two brake types. ABS intervention during the braking-in-turn maneuver was primarily because of the unloading of the inside wheel due to body roll than a difference in brake torque of the two types of brakes.

The third study tested RSC effectiveness for the two trailer brake types. Entry speed and steering amplitude rollover thresholds for four maneuvers were determined similar to the evaluation of roll-stability discussed in Chapter 6 and Section 8.1.2. The disc brake's higher braking torque and quicker braking action improved the trailer RSC effectiveness by more quickly slowing the vehicle. This resulted in a higher rollover threshold for three of the four maneuvers. When the tractor's ESC was active in addition to the trailer RSC, however, there was not a substantial or consistent difference in rollover threshold for the two brakes. The tests were conducted with special consideration to maintain brake temperature at the start of the tests within a small range, to extent possible.

In summary, when the tractor provides some braking action during lower speed maneuvers (≤ 50 mph) (i.e., braking-in-turn or with tractor ESC intervention), the difference between trailer brake types is less evident. At higher speeds and more frequent brake application, the disc brakes performed more consistently and provided substantially shorter stopping distance. The disc brakes marginally improved the trailer RSC effectiveness, compared to drum brakes.

8.2 Recommended Future Work

8.2.1 Tire Slip Effect on Off-tracking

There is little research directly detailing the effect of tire lateral slip on off-tracking. It is known that as a vehicle speed increases, off-tracking reduces due to larger rear sideslip angles. When tire slip is negligible, geometric models accurately capture vehicle motion and off-tracking. When tire slip grows, however, dynamic models are required to accurately estimate off-tracking. The transition from when a geometric model is acceptable to when a dynamic model is required has not been clearly defined for long combination vehicles. Furthermore, the effects of cargo load, turn radii, and tire-road friction coefficient on this have not been investigated. These effects could be analyzed within a simulation study using a dynamic model validated against field tests.

8.2.2 Stability Control Tuning and Driver Behavior

Aggressive RSC tuning results in frequent and hard braking action. Unlike tractor ESC, trailer RSC does not impact the tractor's throttle. Because of this, an aggressive driver may increase throttle to match a desired speed and overcome the braking torque caused by trailer RSC. Frequent and hard braking action will reduce tire, brake pad, and brake rotor/drum life cycles and, more importantly, increase likelihood of brake fade. Brake fade increases braking distance and reduces RSC effectiveness. Naturalistic driving

studies for different RSC tuning programs may provide an insight on ideal RSC tuning that can reduce brake application frequency and amplitude without hindering the RSC effectiveness. There is also a gap in the literature on driver behavior and response to ESC intervention, especially for LCVs. A naturalistic driving study in a closed course may help define how ESC tuning effect driver response and behavior.

8.2.3 Further Analysis on Brake Type Effect on Suspension Dynamics, ABS, and RSC Tuning

As shown in Chapter 7, the trailer disc brakes generated higher braking torque than drum brakes, resulting in shorter stopping distances, more frequent ABS intervention, and greater suspension jounce. The impact of these relationships on suspension dynamics, RSC tuning, and tire wear should be investigated more thoroughly for both single- and tandem-axle trailers with leaf and air spring suspensions. Quick and large brake torque application can cause wheel hop, wheel lockup, and other undesirable dynamics, especially for leaf-spring tandem axles with load-distribution linkages. This topic was briefly introduced and discussed in Chapter 7 within the scope of the program. A more thorough evaluation of such dynamics is warranted in future studies. Furthermore, for a known suspension geometry, brake pad position, brake caliper mounting, air spring effective area, air spring pressure, vehicle deceleration, and vehicle inertial properties, the brake torque may be estimated during a steady straight-line braking event. This should be further investigated in field tests and validated against wheel torque measurements. In theory, such information could be used to continuously monitor brake life and performance.

Brake temperature and fade of the two brake types should be investigated further. The tests described in Chapter 7 were designed to maintain brake temperatures within the normal operating range. Future tests may require brake type durability testing to better analyze brake fade when installed on the vehicle rather than in the dynamometer testing like that in a few past studies. Additionally, computational-fluid dynamics (CFD) analysis for passenger cars should be extended to evaluate and improve trailer brake cooling, especially during frequent and aggressive brake applications.

Lastly, conclusions drawn from the brake type performance evaluation studies detailed in Chapter 7 cannot be extrapolated to all disc and drum brakes. As mentioned earlier, brake performance is largely influenced by the composition of the brake shoe material, brake size, and many other parameters outside of the scope of this study. Although the observed results and conclusions coincide with past studies, a more detailed parametric study of trailer brakes in commercial trucks is suggested in a future study.

References

- 1 2020, "Large Truck and Bus Crash Facts 2018," No. FMCSA-RRA-19-018, Federal Motor Carrier Safety Administration.
- 2 Saxe, C., and Cebon, D., 2019, "Estimation of trailer off-tracking using visual odometry," *Vehicle System Dynamics*, 57:5, pp. 752-776, DOI: 10.1080/00423114.2018.1484498.
- 3 Millar, D.S., and Walton, C.M., 1985, "Off-tracking of the Larger Combination Commercial Vehicles," *Transportation Research Record* 1026, pp. 62-65.
- 4 Erkert., E.W., Sessions, J., and Layton, J.D., 1989, "A Method for Determining Off-Tracking of Multiple Unit Vehicle Combinations," *Journal of Forest Engineering*.
- 5 Sayers, M.W., 1991, "Exact Equations for Tractrix Curves Associated with Vehicle Off-Tracking," *Vehicle System Dynamics*, 20:5, pp. 297-308, DOI: 10.1080/00423119108968991.
- 6 Christensen, T.C., and Blythe, W., 2000, "Off-tracking: History, Analysis, and Simulation," *SAE Transactions*, 2000, Vol. 109, Section 6: *Journal of Passenger Cars: Mechanical Systems Journal*, pp. 592-612.
- 7 Lacy, L., 2018, "Dwight D. Eisenhower and The Birth of The Interstate Highway System", U.S. Army, accessed July 2021, <https://www.army.mil/article/198095>.
- 8 Stevens, H., Tignor, S., and LoJacono, J., 1966 "Off-tracking Calculation Charts for Trailer Combinations," *SAE Transactions*, Vol. 74, pp. 395-413.
- 9 Heald, K., 1986, "Use of the WHI Off-Tracking Formula," *Symposium of Geometric Design for Large Trucks*, *Transportation Research Record* N1052, pp 53-62.
- 10 Fancher, P.S., Segel, L., Winkler, C.B., and Ervin R.D., 1984, "Tracking and Stability of Multi-Unit Truck Combinations," UMTRI-84-25, The University of Michigan Transportation Research Institute.
- 11 Fong, K.T., and Chenu., D.C., 1986, "Simulation of Truck Turns with a Computer Model," *Transportation Research Record* 1100, pp. 20-29.
- 12 Sayers, M., 1986, "Vehicle Off-Tracking Models," *Transportation Research Record* 1052, pp53-62.
- 13 Jindra, F., 1963, "Off-tracking of Tractor-trailer Combinations," *Automobile Engineer*, pp. 96-101.
- 14 Woodrooffe, J., Smith., C., Morisset., L., 1985, "A Generalized Solution of Non-Steady State Vehicle Off-Tracking in Constant Radius Curves," *Truck & Bus Meeting and Exposition*, Chicago, IL, December 9-12, 1985.
- 15 Garber, S., 2008, "Designing for Truck Movements and Other Large Vehicles in Portland," *Portland Office of Transportation*.
- 16 Chen, Y., Zheng, X., Peterson, A., and Ahmadian, M., 2020, "Simulation Evaluation on the Rollover Propensity of Multi-trailer Trucks at Roundabouts," *SAE technical paper*, Art no. 2020-01-5005, doi: 10.4271/2020-01-5005.
- 17 Figliola, R.S., Beasley, D., 2015, *Theory and Design for Mechanical Measurements* 6th Edition, John Wiley and Sons, NY.
- 18 Brunsdon, C., 2007, "Path Estimation from GPS Tracks," *9th International Conference on GeoComputation*.
- 19 "A Brief Introduction to Time-of-Flight Sensing: Part 1 - The Basics," *TeraBee*, accessed October 2021, <https://www.terabee.com/a-brief-introduction-to-time-of-flight-sensing-part-1-the-basics/>

- 20 Zhang, Z., Huang, Y., Wang, H., and Khajepour, A., 2018, "A Comparative Study of Equivalent Modelling for Multi-Axle Vehicle," *Vehicle System Dynamics*, 56:3, pp. 443-460, DOI: 10.1080/00423114.20171387277.
- 21 Dragcevic, V., Korlaet, Z., Stanceric, I., 2008, "Methods for Setting Road Vehicle Movement Trajectories," *The Baltic Journal of Road and Bridge Engineering*, 3(2), pp. 57-64.
- 22 Prince, G.E., and Dubois, S.P., 2009, "Mathematical Models for Motion of the Rear Ends of Vehicles," *Mathematical and Computer Modelling*, 49, pp. 2049-2060.
- 23 1970, "Off-tracking Characteristics of Trucks and Truck Combinations," Western Highway Institute, San Francisco, CA.
- 24 Chen, Y., Zheng, X., Peterson, A., and Ahmadian, M., 2020, "Simulation Evaluation on the Rollover Propensity of Multi-Trailer Trucks at Roundabouts," *SAE Technical Paper*, 2020-01-5005, DOI:10.4271/2020-01-5005.
- 25 Grislis, A., 2010, "Longer Combination Vehicles and Road Safety," *Transport*, 25:3, pp. 336-343, DOI: 10.3846/transport.2010.41.
- 26 Bickel, D.R., 2003, "Smoothing Before Estimating Uncertainty, Scaling, and Intermittency: Application to Short Heart Rate Signals," *Fractals*, 11:3, pp. 245-252.
- 27 Pecchini, D., and Giuliani, F., 2013, "Experimental Test of An Articulated Lorry Swept Path," *Journal of Transportation Engineering*, 139(12), pp. 1174-1183, DOI: 10.1061/(ASCE)TE.1943-5436.0000601.
- 28 Balderas, L., Fancher, P.S., 1988, "Representation of Truck Tire Properties in Braking and Handling Studies: The Influence of Vertical Load on Side Force Characteristics," Final Report, University of Michigan Transportation Research Institute, UMTRI-88-29.
- 29 Jazar, R. N., 2013, *Vehicle Dynamics: Theory and Application 2nd Edition*, Springer Science & Business Media, Riverdale, NY. DOI 10.1007/978-1-4614-8544-5.
- 30 Zaho, Y.Q., Li, H.Q., and Lin, F., 2017, "Estimation of Road Friction Coefficient in Different Road Conditions Based on Vehicle Braking Dynamics," *Chinese Journal of Mechanical Engineering*, vol. 30, pp. 982-990.
- 31 Gillespie, T.D., 1992, *Fundamentals of Vehicle Dynamics*, Society of Automotive Engineers, Inc., Warrendale, PA.
- 32 Fancher, P., and Winkler, C., 2007, "Directional Performance Issues in Evaluation and Design of Articulated Heavy Vehicles," *Vehicle System Dynamics*, 45(7-8), pp. 607-647.
- 33 International Organization for Standardization, 2011, "Road vehicles - Vehicle dynamics and road-holding ability - Vocabulary," No. ISO 8855:2011(E).
- 34 Hou, Y., 2017, "Roll and Yaw Stability Evaluation of Class 8 Trucks with Single and Dual Trailers in Low- and High-Speed Driving Conditions," Ph.D. dissertation, Virginia Tech Department of Mechanical Engineering.
- 35 Kim, A.E., 2017, "Experimental Evaluation of the Dynamic Performance Benefits of Roll Stability Control Systems on A-train Doubles," M.S. thesis, Virginia Tech Department of Mechanical Engineering.
- 36 Schramm, D., Hiller, M., and Bardini, R., 2018, *Vehicle Dynamics Modeling and Simulation 2nd Edition*, Springer-Verlag GmbH Deutschland, Berlin, Germany.
- 37 2005, "Ontario Regulation 32/94: Vehicle Configurations," <https://www.ontario.ca/laws/regulation/940032>.

- 38 Dahlberg, E., 2000, "A Method Determining the Dynamic Rollover Threshold of Commercial Vehicles," SAE Technical Paper Series.
- 39 Walz, M., 2005, "Trends in the Static Stability Factor of Passenger Cars, Light Trucks, and Vans," No. DOT HS 809 868, National Highway Traffic Safety Administration, Office of Regulatory Analysis and Evaluation Planning, Evaluation and Budget.
- 40 Winkler, C.B., and Ervin, R., 1999, "Rollover of Heavy Commercial Vehicles," No. UMTRI-99-19, The University of Michigan Transportation Research Institute, Ann Arbor, MI.
- 41 Winkler, C.B., and Fancher, P.S., 1988, "Directional Dynamics of Multi-Articulated Heavy Trucks Employing Controlled Steering of Dolly Wheels." *Vehicle System Dynamics*, 17:sup1, pp. 525-540, DOI: 10.1080/00423118808969293.
- 42 McGee, H., Strickland, R., 1994, "An Automatic Warning System to Prevent Truck Rollover on Curved Ramps," *Public Roads* Vol. 57, No. 4, Federal Highway Administration Research and Technology.
- 43 Strickland, R., and McGee., H., 1998, "Evaluation of Prototype Automatic Truck Rollover Warning Systems," Final Report, Federal Highway Administration, FHWA-RD-97-124.
- 44 Ervin, R., Winkler, C., Fancher, P., Hagan, M., Krishnaswami, V., Zhang, H., and Bogard, S., 1998, "Two Active Systems for Enhancing Dynamic Stability in Heavy Truck Operations," UMTRI-98-39 Final Report DTNH22-95-H-07002.
- 45 Lin, R.C., Cebon, D., and Cole, D.J., 1994, "An Investigation of Active Roll Control of Heavy Road Vehicles," *Vehicle System Dynamics*, 23:sup1, pp. 308-321, DOI: 10.1080/00423119308969523.
- 46 Sampson, D.J.M., and Cebon, D., 1998, "An Investigation of Roll Control System Design for Articulated Heavy Vehicles", *Proc. 4th International Symposium on Advanced Vehicle Control*, Nagoya, Japan.
- 47 Sampson, D.J.M., McKeivitt, G., and Cebon, D., 1999, "The Development of an Active Roll Control System for Heavy Vehicles", *Proc. 16th IAVSD Symposium on the Dynamics of Vehicles on Roads and Tracks*, Pretoria, South Africa.
- 48 Dunwoody, B., and Froese, S., 1993, "Active Roll Control of a Semi-Trailer," *SAE Transactions*, Vol. 102, Section 2: *Journal of Commercial Vehicles*, pp. 999-1004.
- 49 Sampson, D.J.M., and Cebon, D., 2003, "Achievable Roll Stability of Heavy Road Vehicles", *IMechE J. Automobile Engineering*, 217(4), pp. 269-287.
- 50 Miede, A., and Cebon., D., 2005, "Active Roll Control of an Experimental Articulated Vehicle," *Proc. IMechE Vol. 219 Part D: J. Automobile Engineering*, DOI: 10.1243/095440705X28385.
- 51 Huang, H.H., Yedavalli R.K., and Guenther, D.A., 2012, "Active Roll Control for Rollover Prevention of Heavy Articulated Vehicles with Multiple-Rollover-Index Minimization," *Vehicle System Dynamics*, 50:3, pp. 471-493, DOI: 10.1080/00423114.2011.597863.
- 52 Lin, R.C., Cebon, D., and Cole, D.J., 1996, "Optimal Roll Control of a Single-Unit Lorry", *IMechE J. Automobile Engineering*, 210(1) pp. 45-55.
- 53 Chen, Y., Ahmadian, M., and Peterson, A., 2015, "Pneumatically Balanced Heavy Truck Air Suspensions for Improved Roll Stability," *SAE Technical Paper* No. 2015-01-2749.
- 54 Richardson, S., Sandvik, A., Jones, C., Josevski, N., Pok, W., and Orton, T., 2013, "The Comparative Testing of Single and Double Ride Height Control Valve Suspension Control Systems," *23rd International Technical Conference on the Enhanced Safety of Vehicles*, No. 13-0292.

- 55 Hou, Y., Chen, Y., and Ahmadian, M., 2016, "A Simulation-Based Study on the Improvement of Semi-Truck Roll Stability in Roundabouts," SAE Technical Paper 2016-01-8038.
- 56 White, D.L., 2000, "Parametric Study of Leveling System Characteristics on Roll Stability of Trailing Arm Air Suspension for Heavy Trucks," SAE Technical Paper 2000-01-3480.
- 57 Yu, H., Güvenç, L., and Özgüner, Ü., 2008, "Heavy Duty Vehicle Rollover Detection and Active Roll Control," *Vehicle System Dynamics*, 46:6, pp. 451-470, DOI: 10.1080/00423110701477529.
- 58 Miede, A., and Cebon, D., 2005, "Optimal Roll Control of an Articulated Vehicles: Theory and Model Validation," *Vehicle System Dynamics*, 43:12, pp. 867-884, DOI: 10.1080/00423110500217167.
- 59 Rakheja, S., and Piche, A., 1990, "Development of Directional Stability Criteria for an Early Warning Safety Device," SAE Transactions, Vol. 99, Section 2: Journal of Commercial Vehicles, pp. 877-889.
- 60 Fancher, P., Winkler, C., Ervin, R., and Zhang, H., 1998, "Using Braking to Control the Lateral Motions of Full-Trailers," *Vehicle System Dynamics*, 29:S1, pp. 462-478, DOI: 10.1080/00423119808969579.
- 61 Svenson, A., Salaani, and M., Heydinger, G., 2009, "Implementation of Stability Control for Tractor-Trailers Using the National Advanced Driving Simulator." NHTSA – ESV, No. 09-0190.
- 62 Rao, S.Y., Jeong, J., Ashby, R.M., Heydinger, G.J., and Guenther, D.A., 2014, "Modeling and Validation of ABS and RSC Control Algorithms for a 6x54 Tractor and Trailer Models Using SIL Simulation," SAE Technical Paper 2014-01-0135, DOI: 10.4271/2014-01-0135.
- 63 Rao, S.Y., 2013, "Development of a Heavy Truck Vehicle Dynamics Model Using TruckSim and Model Based Design of ABS and ESC Controllers in Simulink," Master Thesis, The Ohio State University, Columbus, OH.
- 64 National Highway Traffic Safety Administration, 2015, "FMVSS No. 136, Electronic Stability Control Systems for Heavy Vehicles," Docket No. NHTSA-2015-0056, Department of Transportation, Washington, DC.
- 65 Saskatchewan Government Insurance, 2019, "Simple Tractor-Trailer System," Air Brake Manual, accessed November 2021, https://www.sgi.sk.ca/air-brake/-/knowledge_base/air-brake/simple-tractor-trailer-system.
- 66 Saskatchewan Government Insurance, 2019, "Air Brake Manual.," https://www.sgi.sk.ca/documents/625510/626984/airbrake_manual.pdf/1601c666-44bf-4078-a1ae-97d436440885.
- 67 National Highway Traffic & Safety Administration, 2009, "Federal Motor Vehicle Safety Standards; Air Brake Systems," Docket No. 49 CFR Part 571, NHTSA-2009-0083.
- 68 National Highway Traffic & Safety Administration, 2011, "Tractor Semi-Trailer Stability Objective Performance Test Research – Roll Stability," Accessed: January 2020. <https://www.nhtsa.gov/sites/nhtsa.gov/files/811467.pdf>
- 69 Zhang, Z., Sun, N., Chen, Y., and Ahmadian, M., 2021, "Detailed Modeling of Pneumatic Braking in Long Combination Vehicles," *SAE International Journal of Commercial Vehicles*, 14(3):2021, doi:10.4271/0214-03-0020.
- 70 Gohring, E., and Glasner, E.C., 1990, "Performance Comparison of Drum and Disc Brakes for Heavy Duty Commercial Vehicles," Truck and Bus Meeting and Exposition, 1990.

- 71 Dunn, A.L., Hoover, R.L., and Zagorski, S.B., 2004, "The Effects of Foundation Brake Configuration on Class-8 Tractor Dry Stopping Performance," Transportation Research Center, Inc., SAE Technical Paper Series, 2004-01-2701.
- 72 Hoover, R.L., and Zagorski, S.B., 2005, "Comparison of Heavy Truck Foundation Brake Performance Measured with an Inertia Brake Dynamometer and Analyses of Brake Output Response to Dynamic Pressure Inputs," Transportation Research Center, Inc., New Concepts in Commercial Vehicle Suspensions, Advancements in Braking, Steering, and Stability and Handling Simulation, SAE Technical Paper Series, 2005-01-3611.
- 73 Zagorski, S.B. and Dunn, A.L., 2005, "Braking of Commercial Vehicles Equipped with Air-Disc Brakes from High Speed Effects on Stopping Distance," SAE Transactions, 2005, Vol. 114, Section 2: Journal of Commercial Vehicles, pp. 1-11.
- 74 Hoover, R.L., Zagorski, S.B., and Bushkirk, T.V., 2005, "Air Disc and S-Cam Brake Stopping Performance Comparison for Class-8 Tractor and Trailer Combinations," Transportation Research Center, Inc., New Concepts in Commercial Vehicle Suspensions, Advancements in Braking, Steering, and Stability and Handling Simulation, SAE Technical Paper Series, 2005-01-3614.
- 75 "The Compelling Case for Air Disc Brakes in Heavy Truck Braking: A White Paper," Bendix Spicer Foundation Brake LLC, Elyria, OH.
- 76 Lascrain, M.B., 2021, "Variability Analysis of FMVSS-121 Air Brake Systems: 60-mi/hr Service Brake System Performance Data for Truck Tractors," Oak Ridge National Laboratory, Oak Ridge, Tennessee, FMCSA-RRT-19-005.
- 77 Salaani, M.K., Heydinger, G., Grygier, P., and Garrott, W.R., 2008 "Simulator Study of Heavy Truck Air Disc Brake Effectiveness During Emergency Braking," SAE World Congress, 2008.
- 78 Wang, X., Liang, Y., Huang, C., and Ying, G., 2009 "Experimental Research on Heavy-Duty Tractor Heat Performance of Brake," SAE Brake Colloquium & Exhibition, 2009.
- 79 Parmigiani, J., and Ovaert, T., 2000, "The Transient Temperature Distribution in a Heavy Duty Brake System During Fatigue Crack Testing." SAE World Congress, 2000.
- 80 Shome, B., Kumar, V., Chacko, S., Paluskar, V., and Shridhare, M., 2006, "Numerical Simulation of Drum Brake Cooling for Heavy Trucks," SAE Brake Colloquium & Exhibition, 2006.
- 81 Guo, Y., and Wang, C., 2003, "Temperature Rise Model of Drum Brake," Applied Mechanics and Materials, 419, pp. 701-705.
- 82 Afazal, A., and Mujeebu, M., 2019, "Thermo-Mechanical and Structural Performances of Automobile Disc Brakes: A Review of Numerical and Experimental Studies." Computation Methods in Engineering, 26, pp. 1489-1513.
- 83 Menghua Yan, and Jinliang Xu, 2008, "Prediction Model for Brake-Drum Temperature of Large Trucks on Consecutive Mountain Downgrade Routes Based on Energy Conservation Law", Mathematical Problems in Engineering, vol. 2018, Article ID 4587673, <https://doi.org/10.1155/2018/4587673>.
- 84 Venkatesh, S., and Murugapoopathiraja, K., 2019, "Scoping Review of Brake Friction Material for Automotive," International Conference on Advances in Materials, Manufacturing, and Applied Sciences. Materials Today: Proceedings 16 (2019) pp. 927-933.

- 85 Dunn, A.L., Heydinger, G.J., Rizzoni, G., and Guenther, D.A., 2003, "Empirical Models of Commercial Brake Torque from Experimental Data," SAE Transactions 112 (2003), pp. 1455-1466.
- 86 Subramanian, S.C., Darbha, S., and Rajagopal, K.R., 2004, "Modeling the Pneumatic Subsystem of an S-Cam Air Brake System," Journal of Dynamic Systems, Measurement, and Control, 126, no. 1 (2004): 36-46.
- 87 Green, M., 2010, "How Long Does It Take to Stop? Methodological Analysis of Driver Perception-Brake Times," Transportation Human Factors, vol. 2(3), pp. 195-216.
- 88 Dunn, A.L., Heydinger, G.J., Rizzoni, G., and Guenther, D., 2003, "In-Depth Analysis of the Influence of High Torque Brakes on the Jackknife Stability of Heavy Trucks," SAE Transactions 112 (2003), pp. 255-272.
- 89 Mithun, S., Mariappa, S., and Gayakwad, S., 2014, "Modeling and Simulation of Pneumatic Brake System Used in Heavy Commercial Vehicle," IOSR Journal of Mechanical and Civil Engineering 11, no. 1 (2014): 1-9.
- 90 Subramanian, S.C., Darbha, S., and Rajagopal, K., 2006, "A Diagnostic System for Air Brakes in Commercial Vehicles," IEEE Transactions on Intelligent Transportation Systems 7, no. 3 (2006): 360-376.
- 91 Acarman, T., Ozguner, U., Hatipoglu, C., and Igusky, A.-M., "Pneumatic Brake System Modeling for Systems Analysis," SAE Technical Paper 2000-01-3414, 2000, <https://doi.org/10.4271/2000-01-3414>.
- 92 Bu, F. and Tan, H.-S., "Pneumatic Brake Control for Precision Stopping of Heavy-Duty Vehicles," IEEE Transactions on Control Systems Technology 15, no. 1 (2006): 53-64.
- 93 Wu, J., Zhang, H., Zhang, Y., and Chen, L., "Robust Design of a Pneumatic Brake System in Commercial Vehicles," SAE Int. J. Commer. Veh. 2, no. 1 (2009): 17-28, <https://doi.org/10.4271/2009-01-0408>.
- 94 Palanivelu, S., Patil, J., and Jindal, A.K., "Modeling and Optimization of Pneumatic Brake System for Commercial Vehicles by Model Based Design Approach," SAE Technical Paper 2017-01-2493, 2017, <https://doi.org/10.4271/2017-01-2493>.
- 95 Chen, C., Wagner, G., Pace, M., and Rogness, T., "Prediction of Heavy Truck Stopping Distance and Vehicle Behavior Using Real-Time HIL and SIL Simulation," SAE Technical Paper 2008-01-2604, 2008, <https://doi.org/10.4271/2008-01-2604>.
- 96 Suh, M.-W., Park, Y.-K., Kwon, S.-J., Yang, S.-H. et al., "A Simulation Program for the Braking Characteristics of Tractor-Semitrailer Vehicle," SAE Transactions 109 (2000): 540-550.
- 97 Kipling, R.R., 2017, "Twin 33 Foot Truck Trailers: Making U.S. Freight Transport Safer and More Efficient," 2017, <https://americansformoderntransportation.org/wp-content/uploads/Ron-Knipling-Twin-33-Study-vF.pdf>, accessed 18 Jan. 2022.
- 98 A. Snyder, B. Jones, P. Grygier, and R. Garrott, 2005, "NHTSA Light Vehicle ABS Performance Test Development," Transportation Research Center Inc NHTSA/NVS, vol. 312, 2005.
- 99 Dabrowski, T., Kurzynski, D., Lagowski, P., and Warianek, M., 2018, "The Analysis of Burnishing Process Influence on Disc Brake Friction Characteristics and Its Influence on Vehicle Safety – By Inertia Brake Dynamometer Test," Autobusy – Technika, Eksploatacja, Systemy Transportowe, Vol. 19, No. 6 (2018).
- 100 Kero, M., 2013, "Procedure Improvements for Brake Burnish and Wear Testing," SAE Technical Paper 2013-01-2350.

- 101 Tuegel, R., 1968, "Truck and Bus Brake System Performance Requirements (SAE J992)," SAE Technical Paper 680175.
- 102 Wilson, M., Aylor, D., Zuby, D., and Nolan, J., 2015, "Brake Burnishing Effect on AEB Performance," SAE Journal of Transportation Safety, Vol. 3, No. 1, pp. 39-46.

Appendix A

MATLAB Script for Sayers' Numerical Tractrix Algorithm

```
function [x,y,th,p] = ComputeSayersTractrix(s0,p0,setup_orientation,L,H)
% -----
% Determines axle paths for tractor-trailer with given dimensions as determined by Michael W.
% Sayers' tractrix numerical algorithm.
% ref: Sayers, M., 1986, "Off-tracking Models," Transportation Research Board,1052-011,53-62.
%
% inputs:
%   - s0 = steer axle path arc lengths [m]
%   - p0 = steer axle path curvature [1/m]
%   - setup_orientation = vehicle's initial orientation at beginning of maneuver [deg]
%   - L = vector of vehicle unit wheelbases [m]
%   - H = vector of vehicle unit hitch lengths [m]
%
% outputs:
%   - Ax = 2D matrix of each vehicle unit's axle centerline X coordinate path [m]
%   - Ay = 2D matrix of each vehicle unit's axle centerline Y coordinate path [m]
%   - phi = 2D matrix of each vehicle unit's orientation [deg]
%   - p = 2D matrix of vehicle unit path curvature [1/m]
%
% Note: drive tandem axles on tractor are grouped as one, first column in matrices are
% for steer axle which is represented as its own vehicle unit here.
%
% (c) algorithm: Michael Sayers 1986
%   MATLAB installation with modifications to algorithm: Campbell Neighborgall 2021
% -----

HO = L-H; % hitch offset is negative if behind rear axle
setup_orientation = setup_orientation*pi/180; % [deg]-> [rad]
th0 = cumsum(s0.*p0) +setup_orientation; % steer axle (wheel) heading [rad]

ns = length(s0); % number of path points
nv = length(L); % number of vehicle units

% initialize matrices
z = zeros(ns,nv);
p = z; s = z; th = z;
x = z; dx = z; Dx = z;
y = z; dy = z; Dy = z;

% steer path input and initial position
p(:,1) = p0;
s(:,1) = s0;
s(1,:) = s0(1); % all units move same distance (zero art angle, straight-line)
th(:,1) = th0;

% setup initial conditions for axle positions [m]
for j = 2:nv
    x(1,j) = -(L(j) + sum(H(1:j-1)))*cos(setup_orientation);
    y(1,j) = -(L(j) + sum(H(1:j-1)))*sin(setup_orientation);
end

% -- compute path curvature of rear axle of unit j --
for d = 2:ns

    % compute new coordinates of input path (eqn 7&8) - GOOD
    if abs(p0(d)) > 0
        x(d,1) = x(d-1,1) + (1/p0(d))*(sin(th0(d)) - sin(th0(d-1)));
        y(d,1) = y(d-1,1) - (1/p0(d))*(cos(th0(d)) - cos(th0(d-1)));
    else
        x(d,1) = x(d-1,1) + s0(d)*cos(th0(d));
        y(d,1) = y(d-1,1) + s0(d)*sin(th0(d));
    end

    for j = 2:nv
        % compute path curvature of rear axle of each unit "j" for step "d" (eqn 13)
        p(d,j) = (tan(th(d-1,j-1) - th(d-1,j)) + HO(j-1)*p(d,j-1)) ./ ...
            (L(j)*(1 - (HO(j-1)*p(d-1,j-1)*tan(th(d-1,j-1) - th(d-1,j)))));

        % compute incremental arc length (eqn 21)
        if abs(p(d,j)) > 0
```

```

dx(d,j) = (sin(th(d-1,j) + s(d-1,j)*p(d,j)) - sin(th(d-1,j)))/(p(d,j)*s(d-1,j));
dy(d,j) = -(cos(th(d-1,j) + s(d-1,j)*p(d,j)) - cos(th(d-1,j)))/(p(d,j)*s(d-1,j));
else
p(d,j) = 0;
dx(d,j) = cos(th(d-1,j));
dy(d,j) = sin(th(d-1,j));
end
Dx(d,j) = -x(d-1,j) + x(d,j-1) + HO(j-1)*cos(th(d,j-1));
Dy(d,j) = -y(d-1,j) + y(d,j-1) + HO(j-1)*sin(th(d,j-1));

s(d,j) = (Dx(d,j)*dx(d,j) + Dy(d,j)*dy(d,j) - sqrt((L(j)^2 - Dx(d,j)^2 - Dy(d,j)^2)* ...
(dx(d,j)^2 + dy(d,j)^2) + (Dx(d,j)*dx(d,j) + Dy(d,j)*dy(d,j))^2)) ./ ...
(dx(d,j)^2 + dy(d,j)^2);

% compute new heading angle (eqn 6)
th(d,j) = th(d-1,j) + s(d,j)*p(d,j);

% compute new coordinates for rear wheel (eqn 7&8)
if abs(p(d,j)) > 0
x(d,j) = x(d-1,j) + (1/p(d,j))*(sin(th(d,j)) - sin(th(d-1,j)));
y(d,j) = y(d-1,j) - (1/p(d,j))*(cos(th(d,j)) - cos(th(d-1,j)));
else
x(d,j) = x(d-1,j) + s(d,j)*cos(th(d-1,j));
y(d,j) = y(d-1,j) + s(d,j)*sin(th(d-1,j));
end
end
end
th = th*180/pi;
end

```

Appendix B

MATLAB Script for SGM-A

```
function [Ax,Ay,phi] = ComputeSGM_A(xyphi,setup_orientation,L,H)
% -----
% MATLAB function computes each trailing vehicle units' trajectory given the tractor's
% steer axle path, maneuver setup orientation, and vehicle unit dimensions. This model is
% purely geometric and assumes no wheel slip just after the current step.
% It is assumed that the initial positions of the trailers are directly in line with the
% tractor - i.e.: all articulation angles are zero at time t=0.
%
% inputs:
% - xyphi = tractor steer axle position and orientation [m,m,deg]
% - setup_orientation = tractor's initial orientation at beginning of maneuver [deg]
% - L = vector of wheelbases for all vehicle units [m]
%   example: [0,6.325,7.010,1.820,7.010] - first index is steer axle
% - H = vector of lengths from hitch to hitch for all vehicle units [m]
%   example: [0,6.560,7.772,1.820,7.772] - first index is steer axle
%
% output:
% - Ax = 2D matrix of each vehicle unit's axle's centerline X coordinate path [m]
% - Ay = 2D matrix of each vehicle unit's axle's centerline Y coordinate path [m]
% - phi = 2D matrix of each vehicle unit's orientation [deg]
%
% (c) Campbell Neighborgall 2020
% -----
% number of axles in model (5 for D28 & D33, 3 for S53)
N = length(L);

% initialize matrices
z = zeros(size(xyphi,1),N);
Ax = z;    Ay = z;
dx = z;    dy = z;
dist = z;  th = z;
beta = z;  phi = z;
Ax(:,1) = xyphi(:,1);
Ay(:,1) = xyphi(:,2);
Px = Ax;  Py = Ay;
phi = z + setup_orientation*pi/180;

% setup initial conditions for axle and hitch positions [m]
for i = 2:N
    Ax(1,i) = -(L(i) + sum(H(1:i-1)))*cos(setup_orientation*pi/180);
    Ay(1,i) = -(L(i) + sum(H(1:i-1)))*sin(setup_orientation*pi/180);
    Px(1,i) = -sum(H(2:i))*cos(setup_orientation*pi/180);
    Py(1,i) = -sum(H(2:i))*sin(setup_orientation*pi/180);
end

% steer axle path
dx(:,1) = [0; diff(Ax(:,1))];
dy(:,1) = [0; diff(Ay(:,1))];
dist(:,1) = sqrt(dx(:,1).^2 + dy(:,1).^2);

% iterate through each steer axle position and calculate path of each axle and hitch
for t = 2:size(Ax,1) % time (position) step
    % steer axle heading direction from position (t-1) to position (t)
    th(t,1) = atan2(dy(t,1),dx(t,1));
    if (th(t-1,1) > 3.14) && (th(t,1) < 0)
        th(t,1) = 2*pi + th(t,1);
    end

    for i = 2:N % axle
        % determine unit orientation at position (t)
        beta(t,i) = asin(dist(t,i-1)*sin(pi + th(t,i-1) - phi(t-1,i))/L(i));
        phi(t,i) = phi(t-1,i) - beta(t,i);

        % update axle & hitch position
        Ax(t,i) = Px(t,i-1) - L(i)*cos(phi(t,i));
        Ay(t,i) = Py(t,i-1) - L(i)*sin(phi(t,i));
        Px(t,i) = Px(t,i-1) - H(i)*cos(phi(t,i));
        Py(t,i) = Py(t,i-1) - H(i)*sin(phi(t,i));

        % setup for next iteration
```

```

dx(t,i) = Px(t,i) - Px(t-1,i);
dy(t,i) = Py(t,i) - Py(t-1,i);
dist(t,i) = sqrt(dx(t,i).^2 + dy(t,i).^2);

th(t,i) = atan2(dy(t,i),dx(t,i));
if (th(t-1,i) > 3.14) && (th(t,i) < 0)
    th(t,i) = 2*pi + th(t,i);
end
end
end
phi = phi*180/pi;
end

```

Appendix C

MATLAB Script for SGM-B

```
function [Ax,Ay,phi] = ComputeSGM_B(xyphi,setup_orientation,L,H)
% -----
% MATLAB function computes each trailing vehicle units' trajectory given the tractor's
% steer axle path, maneuver setup orientation, and vehicle unit dimensions. This model is
% purely geometric and assumes no wheel slip just before the current step.
% It is assumed that the initial positions of the trailers are directly in line with the
% tractor - i.e.: all articulation angles are zero at time t=0.
%
% inputs:
% - xyphi = tractor steer axle position and orientation [m,m,deg]
% - setup_orientation = tractor's initial orientation at beginning of maneuver [deg]
% - L = vector of wheelbases for all vehicle units [m]
%   example: [0,6.325,7.010,1.820,7.010] - first index is steer axle
% - H = vector of lengths from hitch to hitch for all vehicle units [m]
%   example: [0,6.560,7.772,1.820,7.772] - first index is steer axle
%
% output:
% - Ax = 2D matrix of each vehicle unit's axle's centerline X coordinate path [m]
% - Ay = 2D matrix of each vehicle unit's axle's centerline Y coordinate path [m]
% - phi = 2D matrix of each vehicle unit's orientation [deg]
%
% (c) Campbell Neighborgall 2020
% -----
% number of axles in model (5 for D28 & D33, 3 for S53)
N = length(L);

% initialize matrices
z = zeros(size(xyphi,1),N);
Ax = z;    Ay = z;
dx = z;    dy = z;
dist = z;   direction = z;
Ax(:,1) = xyphi(:,1);
Ay(:,1) = xyphi(:,2);
Px = Ax;   Py = Ay;

% setup initial conditions for axle and hitch positions [m]
for i = 2:N
    Ax(1,i) = -(L(i) + sum(H(1:i-1)))*cos(setup_orientation*pi/180);
    Ay(1,i) = -(L(i) + sum(H(1:i-1)))*sin(setup_orientation*pi/180);
    Px(1,i) = -sum(H(2:i))*cos(setup_orientation*pi/180);
    Py(1,i) = -sum(H(2:i))*sin(setup_orientation*pi/180);
end

% iterate through each steer axle position and calculate path of each axle and hitch
for t = 2:size(Ax,1) % time (position) step

    for i = 2:N % axle

        % from axle i to hitch i-1 (dt)
        dx(t,i) = Px(t,i-1) - Ax(t-1,i);
        dy(t,i) = Py(t,i-1) - Ay(t-1,i);
        dist(t,i) = sqrt(dx(t,i).^2 + dy(t,i).^2);
        direction(t,i) = atan2(dy(t,i),dx(t,i));
        if (direction(t-1,i) > 3.14) && (direction(t,i) < 0)
            direction(t,i) = 2*pi + direction(t,i);
        end

        % from axle i to axle i (dt)
        Ax(t,i) = Ax(t-1,i) + (dist(t,i) - L(i))*cos(direction(t,i));
        Ay(t,i) = Ay(t-1,i) + (dist(t,i) - L(i))*sin(direction(t,i));

        % from hitch i to hitch i (dt)
        Px(t,i) = Ax(t-1,i) + (dist(t,i) - H(i))*cos(direction(t,i));
        Py(t,i) = Ay(t-1,i) + (dist(t,i) - H(i))*sin(direction(t,i));

    end
end
phi = direction*180/pi;
```

Appendix D

Analog Sensor Calibration Curves

(a) String potentiometers

Table D-1. Calibration curve-fits for suspension travel string potentiometers.

UniMeasure Model	S/N	Curve-fit y = inches, x = voltage	R-squared
VP-510 (drop-frame)	31050012	$y = 0.9989x + 0.1679$	1
	30050468	$y = 0.9714x + 0.1924$	1
	31050010	$y = 0.9719x + 0.1981$	1
	30030372	$y = 0.9819x + 0.1878$	0.9999
	30050467	$y = 0.9916x + 0.4518$	1
JX-P510 (straight-rail)	50030319	$y = 0.9955x + 0.1969$	1
	50030317	$y = 0.9984x + 0.1766$	1
	50030318	$y = 0.9985x + 0.1772$	1
	50030321	$y = 0.9986x + 0.1996$	1
	50030320	$y = 1.0013x + 0.1824$	1

(b) Accelerometers

Table D-2. Accelerometer calibration data provided by the manufacturer, PCB Piezotronics®.

PCB Model	S/N	Axis	Sensitivity (mV/g)	Offset (mV)
3713F1110G Triaxial	LW9827	x	133.0	7.1
		y	132.9	7.1
		z	133.0	9.5
	LW9759	x	132.6	10.5
		y	132.8	7.5
		z	132.9	10.0
	LW9547	x	132.5	8.7
		y	132.5	8.0
		z	133.6	9.9
	LW9830	x	132.9	10.5
		y	133.3	8.8
		z	133.0	9.8
	LW9828	x	132.9	10.4
		y	132.9	6.4
		z	132.9	9.2
3741F1210G Single-axis	14409	SA	270.0	0.2
	14410	SA	270.0	2.0

(c) Pressure transducers – brake chamber, tank, suspension, and braking robot air pressures

The Honeywell pressure transducers are ratiometric sensors with pressure ranges of 0-250psi and acceptable excitation (input) voltage range of 4.75-5.25V. The equation below converts the output and input voltage to pressure. This pressure is then put through the linear curve-fit to match the common reference pressure transducer measurement.

$$\text{Honeywell: } P = \frac{V_{meas} - 0.1V_{in}}{0.8V_{in}} (250\text{psi}) \quad (\text{D-1})$$

The OMEGA pressure transducer is not ratiometric, but instead has a voltage regulator so that its output is in proportion to the measured pressure. Like the Honeywell's, this pressure (equation shown below) is then converted to the reference pressure with the curve-fit provided in Table D-1.

$$\text{OMEGA: } P = \frac{200\text{psi}}{5\text{V}} (V_{meas}) \quad (\text{D-2})$$

The reference pressure transducer's curve fit was determined from a series of tightly controlled, static pressures.

Table D-3. Curve-fits for all pressure transducers, operational and spare.

Brand/Model	#	S/N	Curve-fit y = equivalent reference pressure x = measured pressure	R-squared
Honeywell PX2EN1XX250PSAAX	reference	1932410027	y = 0.9946 x + 0.3914	1
	1	1225700097	y = 0.9987 x + 0.4150	1
	2	1424710213	y = 0.9794 x - 0.8805	1
	3	1424710201	y = 0.9871 x + 0.1930	1
	4	1225700100	y = 0.9962 x + 0.9042	1
	5	1932410100	y = 0.9929 x + 0.6217	1
	6	1932400077	y = 0.9910 x + 0.9261	1
	7	1424710198	y = 0.9969 x + 0.2017	1
	8	1424710202	y = 0.9978 x + 0.2363	1
	9	1424710194	y = 0.9962 x + 0.6249	1
	10	1424710203	y = 0.9965 x + 0.2396	1
	11	1225700030	y = 0.9969 x + 0.6637	1
	12	1424710188	y = 0.9971 x + 0.2607	1
	13	1424710187	y = 0.9956 x + 0.5589	1
	14	1932510230	y = 0.9924 x + 0.5137	1
	15	1932410083	y = 0.9953 x + 0.4655	1
	16	1932900052	y = 0.9976 x + 0.7787	1
	17	1932310116	y = 0.9966 x + 0.7023	1

	18	1932310126	$y = 0.9952 x + 0.7882$	1
	19	1932310148	$y = 0.9856 x + 0.6637$	1
	20	1932300158	$y = 0.9934 x + 1.5438$	1
	21	1932300157	$y = 0.9967 x + 0.7902$	1
	22	091413D194	$y = 0.9973 x - 0.6394$	1
OMEGA PX329-200G5V	23	042614D073	$y = 0.9941 x - 0.6974$	1
	24	091413D177	$y = 0.9957 x - 0.3317$	1
	25	111111D111	$y = 0.9932 x - 0.9410$	1
	26	092913D230	$y = 1.0337 x - 5.0371$	1

(d) Load Cells

The load cells installed in the outriggers were custom ordered from Forsentek. These have a maximum rated tensile load of 100 tons (200,000 lbf). Load cells are ratiometric sensors, so their output voltage V_{in} must be compared to the input voltage V_{in} , maximum rated load F_{max} , and sensitivity S to determine the measured force. This is presented in the equation D-3. The sensitivity for each load cell is provided in Table D-4.

$$F_{meas} = \frac{F_{max} V_{out}}{V_{in} S} \quad (D-3)$$

Table D-4. Sensitivity analysis of the load cells that are used in the outrigger to detect the outrigger contact force.

S/N	Sensitivity [mV/V]
F90522131	1.5668
F90522132	1.5384
F90522133	1.5491
F90522134	1.5134
F90522135	1.5437
F90522136	1.5533
F90522137	1.5673
F90603102	1.5549

Appendix E

Pin Assignments for DB-37 Cables and NI cRIO

Module 1 (front trailer and tractor)

MODULE #1 (front trailer and tractor)												
Pin # (breakout board)	Analogue Input (NI 9205)	TDMS (channel)	LabVIEW (scope)	Sensor	Label	Signal Type	Paired With (differential)	NI 9205 Voltage	Sensor Make	Sensor Model	Supply Voltage	Wire Color
1	A10	1	1-1	load cell (driver)	LC-1-D	DIFF	pin #20	+/- 0.2 V	Forsentek	FLP-100t	10	G
2	A11	2	1-2	load cell (passenger)	LC-1-P	DIFF	pin #21	+/- 0.2 V	Forsentek	FLP-100t	10	G
3	A12	3	1-3	tractor brake pressure (driver)	PT-B-D-D	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
4	A13	4	1-4	triaxial accelerometer (x)	A3-1-X	NRSE	-	+/- 1 V	PCB	3713F-1110G	13	G
5	A14	5	1-5	triaxial accelerometer (y)	A3-1-Y	NRSE	-	+/- 1 V				O
6	A15	6	1-6	triaxial accelerometer (z)	A3-1-Z	NRSE	-	+/- 1 V				B
7	A16	7	1-7	string pot - suspension (driver)	SP-1-D	NRSE	-	+/- 10 V	UniMeasure	JX-P510 / VP510	13	G
8	A17	8	1-8	string pot - suspension (passenger)	SP-1-P	NRSE	-	+/- 10 V	UniMeasure	JX-P510 / VP510	13	G
9	DO0											
10	COM											
11	A16	9	1-9	suspension pressure	PT-S-1	NRSE	-	+/- 5 V	Omega	PX329-200G5V	13	G
12	A17	10	1-10	brake pressure (driver)	PT-B-D-D	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
13	A18	11	1-11	brake pressure (passenger)	PT-B-P-P	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
14	A19	12	1-12	airtank pressure	PT-A-1	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
15	A20	13	1-13	ABS relay	ABS-2	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
16	A21	14	1-14	RSC relay 1	RSC1-1	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
17	A22	15	1-15	RSC relay 2	RSC2-1	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
18	A23	16	1-16	load cell 10V regulator	10V-1	NRSE	-	+/- 10 V	Texas Instruments	UA7810CKCS	13	P
19	AISENSE											
20	A18	(paired with 1)	-	load cell (driver)	LC-1-D	DIFF	pin #1	-	-	-	-	O
21	A19	(paired with 2)	-	load cell (passenger)	LC-1-P	DIFF	pin #2	-	-	-	-	O
22	A10	17	1-17	tractor brake pressure (passenger)	PT-B-D-P	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
23	A11	18	1-18	PTs & relays 5V regulator	5V-1	NRSE	-	+/- 10 V	Texas Instruments	UA7805CKCS	13	P
24	A12	19	1-19	string pot - steer angle	SP-0	NRSE	-	+/- 10 V	UniMeasure	UX-P510-15-NJC-N6-13M	13	G
25	A13	20	1-20	triaxial accelerometer (x) - tractor	A3-O-X	NRSE	-	+/- 1 V	PCB	3713F-1110G	13	G
26	A14	21	1-21	triaxial accelerometer (y) - tractor	A3-O-Y	NRSE	-	+/- 1 V				O
27	A15	22	1-22	triaxial accelerometer (z) - tractor	A3-O-Z	NRSE	-	+/- 1 V				B
28	PFI0											
29	COM											
30	A124	23	1-23	Steering Robot Actuation Flag	FLAG-S	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
31	A125	24	1-24	Braking Robot Actuation Flag	FLAG-B	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	O
32	A126	25	1-25	ESC relay	ESC-0	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
33	A127	26	1-26	ABS relay	ABS-0	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
34	A128	27	1-27	tractor tank pressure	PT-A-0	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
35	A129	28	1-28	auxiliary air tank pressure	PT-AA	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
36	A130	29	1-29	brake robot actuator air pressure	PT-BA	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250PSAAX	5	G
37	A131	30	1-30	PTs & relays 5V regulator	5V-0	NRSE	-	+/- 10 V	Texas Instruments	UA7805CKCS	13	P

Module 2 (rear trailer)

MODULE #2 (rear trailer)												
Pin # (breakout board)	Analogue Input (NI 9205)	TDM5 (channel)	LabVIEW (scope)	Sensor	Label	Singal Type	Paired With (differential)	NI 9205 Voltage Range	Sensor Make	Sensor Model	Supply Voltage	Wire Color
1	AI0	31	2-1	load cell (driver)	LC-2-D	DIFF	pin #20	+/- 0.2 V	Forsestek	FLP-100t	10	G
2	AI1	32	2-2	load cell (passenger)	LC-2-P	DIFF	pin #22	+/- 0.2 V	Forsestek	FLP-100t	10	G
3	AI2	33	2-3	single-axis acceleration	A1-2	DIFF	pin #20	+/- 1 V	PCB	3741F1210G	13	G
4	AI3	34	2-4	triaxial accelerometer (x)	A3-2-X	NRSE	-	+/- 1 V	PCB	3713F1110G	13	G
5	AI4	35	2-5	triaxial accelerometer (y)	A3-2-Y	NRSE	-	+/- 1 V			O	
6	AI5	36	2-6	triaxial accelerometer (z)	A3-2-Z	NRSE	-	+/- 1 V			B	
7	AI6	37	2-7	string pot - suspension (driver)	SP-2-D	NRSE	-	+/- 10 V	UniMeasure	JX-P510 / VP510	13	G
8	AI7	38	2-8	string pot - suspension (passenger)	SP-2-P	NRSE	-	+/- 10 V	UniMeasure	JX-P510 / VP510	13	G
9	DO0											
10	COM											
11	AI16	39	2-9	suspension pressure	PT-S-2	NRSE	-	+/- 5 V	Omega	PX329-200G5V	13	G
12	AI17	40	2-10	brake pressure (driver)*	PT-B-2-D	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250P-SAAAX	5	G
13	AI18	41	2-11	brake pressure (passenger)*	PT-B-2-P	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250P-SAAAX	5	G
14	AI19	42	2-12	airtank pressure	PT-A-2	NRSE	-	+/- 5 V	Honeywell	PX2EN1XX250P-SAAAX	5	G
15	AI20	43	2-13	ABS relay	ABS-2	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
16	AI21	44	2-14	RSC relay 1	RSC1-2	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
17	AI22	45	2-15	RSC relay 2	RSC2-2	NRSE	-	+/- 10 V	TE Connectivity	KUP-11D55-12	5	G
18	AI23	46	2-16	load cell 10V regulator	LC-2-V	NRSE	-	+/- 10 V	Texas Instrument	UA7810CKCS	13	P
19	AISENSE											
20	AI8	(paired with 31)	-	load cell (driver)	LC-2-D	DIFF	pin #1	-	-	-	-	O
21	AI9	(paired with 32)	-	load cell (passenger)	LC-2-P	DIFF	pin #2	-	-	-	-	O
22	AI10	(paired with 33)	-	single-axis acceleration	A1-2	DIFF	pin #3	-	-	-	-	O
23	AI11	47	2-17	PTs & relays 5V regulator	5V-2	NRSE	-	+/- 10 V	Texas Instrument	UA7805CKCS	13	P
24	AI12											
25	AI13											
26	AI14											
27	AI15											
28	PFI0											
29	COM											
30	AI24											
31	AI25											
32	AI26											
33	AI27											
34	AI28											
35	AI29											
36	AI30											
37	AI31											

LEAVE BLANK FOR TRACTOR JB

LEAVE BLANK FOR TRACTOR JB

LEAVE BLANK FOR TRACTOR JB

LEAVE BLANK FOR TRACTOR JB