

U13: Co-Simulation of Heavy Truck Tire Dynamics and Electronic Stability Control Systems (Phase A)

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16. Abstract

Electronic stability control (ESC) systems have been proven to be an effective means of preventing instability and loss of control on both passenger vehicles and heavy trucks. In addition, roll stability algorithms are an effective means of reducing the risk of rollover on heavy trucks with their relatively high centers of gravity. Stability control systems are so effective that the U.S. government has mandated their inclusion on all new passenger vehicles by September 2011, and similar legislation is anticipated soon for heavy trucks. The goal of this research project is to produce a software co-simulation of an articulated tractor-trailer model together with an ESC algorithm.

Such a simulation platform will enable the investigation of truck performance both with and without stability control and the sensitivity of vehicle performance to changes in vehicle parameters. The simulations will be used to conduct experiments to determine particular vehicle configurations and parameters that result in improved vehicle stability and dynamic performance. In addition the simulation platform will provide a means to investigate advanced stability control algorithms, such as algorithms that automatically adapt to changes in vehicle parameters such as trailer load configurations. The co-simulation may be a Hardware-In-the-Loop (HIL) simulation system utilizing a commercial ESC Electronic Control Unit (ECU) or pure software co-simulation with an algorithm representative of a commercial ESC system.

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Table of Contents

EXECUTIVE SUMMARY	1
BACKGROUND	1
BRIEF OVERVIEW	1
RESEARCH TEAM	2
ESC HIL SIMULATION INVESTIGATION	2
ESC SOFTWARE CO-SIMULATION RESULTS	4
TRACK TESTING PRELIMINARY RESULTS	6
FUTURE PROGRAM EFFORTS	9
CHAPTER 1 – GENERAL OVERVIEW	11
BACKGROUND	11
PROJECT TEAM	
MARC Roles and Responsibilities	
Clemson University Roles and Responsibilities	12
National Instruments Roles and Responsibilities	12
PROJECT DESCRIPTION AND OBJECTIVES	
LITERATURE REVIEW	
CHAPTER 2 – ESC HIL SIMULATION INVESTIGATION	15
HIL PROJECT TASKS	15
Simulation Model Development	15
ESC Control Algorithm Development	15
Integration of Michelin Tire Models	15
ESC Hardware Acquisition	
HIL Hardware Specification	16
HIL Hardware Acquisition	17
HIL ECU SENSOR AND ACTUATOR INTEGRATION	
INFORMATION REQUIRED OF ECU SUPPLIER	
ALTERNATIVE PROJECT PLAN: TRUCK/ESC CO-SIMULATION	
CHAPTER 3 – ESC SOFTWARE CO-SIMULATION	21
SOFTWARE CO-SIMULATION SYSTEM	21
YAW STABILITY CONTROL	21
Linear Model Development	21
Yaw Stability Control Algorithm	
CHAPTER 4 - TRACK TESTING	37
SLOWLY INCREASING RAMP STEER MANEUVER	
STEP STEER MANEUVER	
LANE CHANGE MANEUVERS	41
Design of Open-Loop Lane Change Maneuver	41
Dry Lane Change	
Wet Lane Change	45
Comparison of Vehicle States on Wet and Dry Maneuvers	

CHAPTER 5 – CONCLUSION	53
CHAPTER 6 - REFERENCES	55
APPENDIX A	A-1
LITERATURE REVIEW	A-1
ELECTRONIC STABILITY CONTROL ALGORITHMS	A-1
ESC for Passenger Vehicles	A-1
ESC for Articulated Heavy Trucks	A-3
ARTICULATED TRUCK MODELS FOR HANDLING SIMULATION	A-5
Analytical Truck Models	A-6
Commercial Truck Models	A-7

List of Figures

Figure 1. Diagram. HIL System Components.	3
Figure 2. Diagram. ESC Co-Simulation System.	4
Figure 3. Diagram. ESC Control Strategy.	5
Figure 4. Photograph. HIL Rack and Equipment.	18
Figure 5. Graphs. Step Steer Model Response, 100 kph (a).	23
Figure 6. Graphs. Step Steer Model Response, 100 kph (b)	24
Figure 7. Graphs. Simplified TruckSim [®] Model Investigation, 100 kph (a)	26
Figure 8. Graphs. Simplified TruckSim [®] Model Investigation, 100 kph (b)	27
Figure 9. Graphs. Modified Linear Tractor/Semi-Trailer Model, 100 kph (a).	31
Figure 10. Graphs. Modified Linear Tractor/Semi-Trailer Model, 100 kph (b)	32
Figure 11. Graphs. ESC System Co-Simulation on Low Friction Surface, 100 kph (a)	34
Figure 12. Graphs. ESC System Co-Simulation on Low Friction Surface, 100 kph (b)	35
Figure 13. Graph. Slowly Increasing Steer Test RSC Intervention.	38
Figure 14. Graph. Slowly Increasing Steer Test Wheel-End Brake Pressures.	39
Figure 15. Graph. Step Steer Test RSC Intervention.	40
Figure 16. Graph. Step Steer Test Tractor (Red) and Trailer (Blue) Lateral Acceleration	40
Figure 17. Graph. Step Steer Test Wheel-End Brake Pressures.	41
Figure 18. Graph. Double Lane Change Open-Loop Steering Profile.	42
Figure 19. Graph. Dry Double Lane Change Ground Track.	43
Figure 20. Graph. Dry Lane Change High CG Test RSC Intervention.	43
Figure 21. Graph. Dry Lane Change High CG Test Wheel-End Brake Pressures	44
Figure 22. Graph. Dry Lane Change Low CG Test RSC Intervention	45
Figure 23. Graph. Dry Lane Change Low CG Test Wheel-End Brake Pressures.	45
Figure 24. Graph. Wet Lane Change High CG Test RSC and YSC Intervention	46
Figure 25. Graph. Wet Lane Change Low CG Test Wheel-End Brake Pressures	46
Figure 26. Graph. Wet Lane Change High CG Test Tractor States and YSC Intervention	47
Figure 27. Graph. Dry and Wet Lane Change Steering Input Comparison.	48
Figure 28. Graph. Dry and Wet Lane Change Vehicle Speed Comparison.	49
Figure 29. Graph. Dry and Wet Lane Change Tractor Yaw Rate Comparison.	49
Figure 30. Graph. Dry and Wet Lane Change Tractor Lateral Acceleration Comparison	50
Figure 31. Graph. Dry and Wet Lane Change Trailer Articulation Angle Comparison	50
Figure 32. Graph. Dry and Wet Lane Change Trailer Lateral Acceleration Comparison	50
Figure 33. Graph. Dry and Wet Lane Change Trailer Lateral Velocity Comparison	51
Figure A- 1. Diagram. Basic ESC High-Level Control Structure.	. A-2
Figure A- 2. Illustration. Jackknife (Oversteer) Mitigation [23].	. A-4
Figure A- 3. Illustration. Rollover Condition [23]	. A-4
Figure A- 4. Illustration. Rollover Mitigation [23]	. A-5

List of Tables

Table 1. Specified National Instrum	ents Hardware and Software1	7
Table 2. Track Testing Maneuvers.		7

List of Equations

Equation 1. Tractor and Semi-Trailer Lateral Accelerations	22
Equation 2. Axle Linear Roll Steer Model.	28
Equation 3. Quasi-Steady State Roll Model.	29
Equation 4. Linear Lateral Force Compliance Steer Model.	29
Equation 5. Lateral Force Model.	30
Equation 6. Effective Cornering Stiffness Due to Lateral Force Compliance	30
Equation 7. Effective Aligning Stiffness Due to Lateral Force Compliance	30
Equation 8. Effective Cornering Stiffness Due to Aligning Moment Compliance.	30
Equation 9. Effective Aligning Stiffness Due to Lateral Force Compliance	30

Executive Summary

Background

The Co-Simulation of Heavy Truck Tire Dynamics and Electronic Stability Control System project is a research effort conducted by the National Transportation Research Center, Inc., University Transportation Center (NTRCI) in partnership with Clemson University International Center for Automotive Research (CU-ICAR), Michelin Americas Research Company (MARC), and National Instruments (NI). This research is one of the projects conducted by the NTRCI in its role as a University Transportation Center (UTC) for the Research and Innovative Technology Administration (RITA), an agency within the Department of Transportation (DOT).

Brief Overview

The overall objectives of this research were to:

- Determine performance of the truck ESC system and validate against measured test track data.
- Understand the fundamental operation of the ESC system with respect to vehicle dynamics
- Determine the robustness of the ESC system with respect to vehicle configurations and various vehicle parameters
- Study potential improvements to vehicle stability and/or dynamic performance with the ESC system active due to changes in vehicle parameters
- Investigate further improvements in stability control and system identification algorithm performance through closed-loop simulation with vehicle models

This project entailed the following activities:

- 1. A review of literature relevant to the operation of heavy truck ESC systems
- 2. Investigation of a Hardware-In-the-Loop (HIL) simulation system for truck/ESC system co-simulation
- 3. Investigation of an offline software co-simulation system
- 4. Track testing of a tractor/semi-trailer coordinated with the Heavy Truck Rollover Characterization project
- 5. Initial investigation of the track testing results for ESC co-simulation system validation

A review of relevant literature was conducted to understand the state-of-the-art of heavy truck electronic stability control algorithms. A review of passenger car ESC algorithms was conducted to understand the fundamental operation of these systems since adoption of ESC in the passenger car market preceded that of the heavy truck market. In addition, a review of the relevant literature on ESC and roll control systems for articulated heavy trucks was conducted.

Since a dynamic model of the articulated truck lateral handling was found to serve as the basis for determining desired vehicle behavior in ESC systems, a survey of low-order models suitable for processing on an ESC embedded controller was conducted. In addition, high-fidelity commercial tools capable of simulating tractor/semi-trailer systems such as TruckSim[®] and SIMPACK[®] were reviewed.

Research Team

Clemson was the overall lead on Phase A of this research, with significant engineering support provided by Michelin. Clemson led all tasks and deliverables, including the development of the co-simulation system, evaluation of track testing results, and all project reporting. Michelin served as the primary industry technical lead and provided significant engineering support, primarily in the development of the tractor model and flatbed trailer model used for initial ESC co-simulation development. National Instruments supplied real-time hardware for the HIL test system, as well as LabVIEW[™] software used for modeling the ESC control algorithm and co-simulating with the TruckSim[®] vehicle model. National Instruments also provided direct financial support to the project.

In addition to these tasks, Clemson and Michelin were both heavily involved in the planning and execution of the tractor/semi-trailer track testing and data analysis. This track testing was a combined effort of both this ESC co-simulation project Phase A (Project U13) and the Heavy Truck Rollover Characterization project (HTRC) Phase B (Project U19). Clemson helped define test maneuvers and instrumentation that yielded useful data for analysis in both projects. Michelin and other members of the HTRC project also contributed to all phases of planning, execution, and data analysis of the track testing.

ESC HIL Simulation Investigation

The requirements for a hardware-in-the-loop simulation system were investigated and deemed to be infeasible without direct support of the ESC electronic control unit (ECU) supplier. The envisioned system would use the ECU connected to a National Instruments Real-Time PXI system running a TruckSim[®] model of the tractor/semi-trailer combination vehicle. Such an HIL simulator could be used to simulate and investigate the performance of the truck equipped with ESC without the need for expensive and time-consuming track time. A conceptual view of the HIL simulation system is shown in Figure 1.



Figure 1. Diagram. HIL System Components.

Investigation of the available information on input/output signals to the ECU resulted in the determination that insufficient details of the ECU operation were available for integration into an HIL system. This included unpublished requirements of the sensors/actuators for diagnostics purposes and proprietary communication buses used in the system. The initial project proposal included engaging the ESC ECU supplier, Bendix Commercial Vehicle Systems, as a partner in the project. However, as Bendix involvement occurred very late in the project timeline, the project focus was shifted away from an HIL simulation system and instead focused on a software co-simulation system.

The following is a list of specific information that would be required from the ECU supplier for the development of an HIL system:

- Details of CAN bus communication with Yaw Angle (YAS) and Steer Angle (SAS) sensor packs
- Details of powertrain J1939 bus communication (which signals, and when?)
- Sensor/actuator inductance/resistance matching for diagnostics (e.g. valve "chuff" test)
- Specific hardware needs for HIL system
 - Valves absolutely required for HIL?
 - Additional sensors or hardware?
- Need to simulate Power Line Carrier (PLC) signal for trailer ECU?

Ultimately due to the timing of the discussions with Bendix and the lack of externally available information, the decision was made to put the HIL system development on hold.

ESC Software Co-Simulation Results

Due to the lack of available information required for the implementation of an HIL simulation system during Phase A of the project, a pure software co-simulation strategy was pursued. In this case the vehicle model is not run in real-time on dedicated hardware, but instead is simulated on a standard desktop PC. However, a second simulation environment is used to model the behavior of the ESC control algorithm. For this project, LabVIEWTM was used to implement an ESC algorithm that is representative of those found in the available literature. LabVIEWTM and TruckSim[®] are used together to "co-simulate" the complete vehicle with ESC system. Information must be exchanged at each simulation timestep between the two tools to ensure that the complete system is accurately simulated. A schematic view of the software co-simulation system is shown in Figure 2.



Figure 2. Diagram. ESC Co-Simulation System.

Heavy truck ESC systems found in the literature typically implement a state feedback control scheme with a linear vehicle model used to determine "desired" states of the vehicle. A schematic view of the control strategy used in the co-simulation system is provided in Figure 3. A critical component of the algorithm is the use of a linear dynamic model to determine the desired vehicle states based on vehicle speed and steering input. The model is the "articulated bicycle" handling model commonly found in the literature with the following states: tractor lateral velocity, tractor yaw rate, trailer relative articulation angle and articulation rate.



Figure 3. Diagram. ESC Control Strategy.

The yaw stability portion of the controller is shown in the figure. Here yaw rate and lateral acceleration are used as model outputs and are compared to the measured values from TruckSim[®] model outputs corresponding to actual sensors used on the real vehicle. A threshold is used to prevent intervention under normal driving conditions and feedback gains are applied to the state deviations. The resulting control output includes logic to apply differential braking pressures to appropriate axles of the vehicle to correct the instability. These pressures serve as setpoints to individual wheel Anti-Lock Brake Systems (ABS) controllers also implemented in LabVIEW[™], but not depicted in the simple schematic figure shown.

A key finding of the research is that the vehicle states of the classic "articulated bicycle model" did not match those of the complete nonlinear vehicle model in TruckSim[®], even for low severity maneuvers within the linear handling region of operation of the tires. As a result, this simple model would be difficult to use to predict desired behavior of the vehicle under normal conditions, since such state deviations may be interpreted as yaw instability and result in unnecessary intervention of the ESC system.

A thorough investigation into the differences between the linear model and TruckSim[®] model resulted in the identification of three main vehicle properties that were the root cause of the deviations:

- 1. Axle roll steer
- 2. Tire lateral force compliance steer
- 3. Tire aligning moment compliance steer (primarily on front axle)

Strategies were devised to incorporate each of these individual effects into the linear articulated bicycle model. The resulting model was found to have very good agreement with the complete nonlinear TruckSim[®] model when exercised with maneuvers in the linear regime.

A yaw stability control system was devised in which the yaw rate was used as a state feedback variable. Co-simulation tests were executed using a lane change maneuver on a low friction surface implemented in TruckSim[®]. Without ESC activation, the tractor and trailer develop significant amounts of sideslip and the combination vehicle experiences large yaw oscillations when exiting the maneuver. With the designed ESC system co-simulated with the TruckSim[®] model executing the same maneuver, the vehicle sideslip and trailer oscillations were significantly reduced throughout the maneuver.

In addition, roll stability control was realized with a second algorithm executing in parallel to the yaw stability control algorithm. This control strategy used simple thresholds for the tractor and trailer lateral acceleration based upon static rollover threshold values for typical tractors and loaded trailers found in the literature. When tractor lateral acceleration or estimated trailer lateral acceleration exceeded these thresholds, brakes were applied to all vehicle wheels to reduce vehicle speed and thus lateral acceleration. Simulations of step steer maneuvers with the co-simulation system proved the strategy to be an effective means of mitigating the threat of vehicle rollover.

Track Testing Preliminary Results

As described previously, track testing of a tractor and tanker semi-trailer was conducted to provide data to use for the validation of the ESC co-simulation project. This testing was conducted in conjunction with the HTRC Phase B project. Members of both project teams collaborated on all test planning phases, test execution, and post-test analysis of the data. While some test instrumentation and maneuvers were designed specifically for the ESC Co-Simulation project, the majority of instrumentation and test maneuvers were designed to yield data of use for both projects.

The vehicle tested was a Volvo VT830 tractor with an LBT tanker trailer. Four maneuvers were used for testing: slowly-increasing ramp steer, step steer, dry double lane change, and wet double lane change. The Volvo tractor was equipped with a Bendix full ESC system and the trailer was equipped with an independent Bendix roll stability control system. Tests for the various maneuvers were executed with various combinations of tractor and trailer stability control systems enabled/disabled. In addition, different vehicle configurations were tested. All maneuvers were repeated for vehicle configurations with standard dual tires with the tanker fully loaded in both a "high" center of gravity (CG) configuration and a "low" CG configuration. In addition, some maneuvers were also repeated with a configuration. A steering robot was used to conduct all tests. For the ESC Co-Simulation project, the objective was to produce data that

could be used to validate the ESC co-simulation system. A thorough validation of the ESC cosimulation system using the track test data is being deferred to the proposed subsequent Phase B of the project. After completion of the testing, an initial analysis of the track test data with tractor ESC systems enabled and disabled (but with trailer stability control disabled) was conducted. The purpose of this investigation was a qualitative analysis to determine whether the logic applied to the baseline ESC algorithm developed based on available literature matched the behavior of the commercial ESC system. The following is a summary of the basic analysis of tractor ESC performance for the four defined test maneuvers.

A slowly increasing ramp steer test was used to simulate quasi-steady state conditions. For this test the steering wheel angle was increased at a rate of 10 deg/sec while the driver attempted to maintain a constant speed of 30 mph (unless the ESC system intervened with throttle control). This test was used primarily by the HTRC project team to determine basic vehicle properties such as understeer characteristics. However, the maneuver can be used to evaluate the roll stability control (RSC) portion of the tractor ESC system as vehicle lateral acceleration increases linearly with steer input. Tractor ESC intervention was found to occur consistently at approximately 0.3 g of lateral acceleration. This value is consistent with a static rollover threshold for a fully loaded trailer as described in available literature. In the low CG case the RSC application may be at slightly higher lateral acceleration, but this is difficult to say conclusively. In all ramp steer maneuvers the ESC system intervened by disabling torque demand of the engine and by applying brakes to the trailer to reduce vehicle speed. Bendix literature on the system indicates that it will brake all tractor and trailer drive wheels under large lateral acceleration conditions. The reason for this discrepancy under these conditions is unclear.

A step steer test was conducted using a quick ramp of steering wheel angle to 170 degrees in 1 second at a speed of 33 mph. The test was conducted "dropped throttle" in the sense that the tractor was put in neutral and allowed to coast while the steering robot and data acquisition triggered when the target speed was reached. For the high CG configuration step steer with tractor ESC on and trailer ESC off, the lateral acceleration builds quickly as the steering wheel ramps. The tractor ESC system did not intervene in the step steer test until the tractor unit reached approximately 0.4 g of lateral acceleration. This number is considerably higher than the threshold in the steady state ramp steer test. This is likely a result of the fact that the tractor has a higher static rollover threshold than that of the trailer due to a lower CG height, and the fact that the trailer lateral acceleration lags that of the tractor. The exact trigger points for ESC system intervention cannot be determined conclusively without conducting a prohibitively large number of tests under different input conditions. When the ESC system intervened in the step steer tests, all axles of the tractor and trailer were braked as described in the Bendix system literature. Bias between drive axle left and right wheels and bias between the tractor steer and drive axles was observed in the system activation at times during the maneuver. The strategy of the system for introducing bias in the braking activation is unknown.

Clemson took the lead role in designing an open-loop steering profile for the steering robot that would approximate the path of the vehicle during an emergency double-lane change maneuver. The TruckSim[®] vehicle model was used with an updated preliminary model of the tanker to design the maneuver. First a simple driver steering model was used in TruckSim[®] to simulate a driver attempting to follow a profile that effected a double lane change maneuver. The resulting driver steering input profile was analyzed and a piecewise linear profile suitable for the steering robot was designed to approximate the steering profile input by the driver model. The amplitudes and times of points in the profile were varied and simulated again in TruckSim[®] to make sure that the designed open-loop profile resulted in a full double lane change with the vehicle following approximately the same track at the end of the maneuver as in the beginning of the maneuver. Subsequent track testing of the actual vehicle using the designed steering robot profile resulted in a maneuver that closely approximated a double lane change maneuver, but with far more repeatable results than can be realized by a human driver. For many of the tests conducted, the amplitude of the steering profile was increased in order to increase steering severity and induce wheel lift at the relatively low speeds that could be realized on the test track.

Open-loop double lane change maneuvers were conducted on the dry asphalt surface at speeds found to induce wheel lift for a particular vehicle configuration. In both the high and low CG configurations, two stability control events were observed when tractor ESC was enabled: one during the initial lane change to the left and another during the return lane change to the right. In both cases, ESC brake intervention was similar to that observed for the step steer tests. However, the second ESC intervention had much lower brake pressures, likely due to the fact that the vehicle speed was greatly reduced by the initial ESC brake activation.

The ESC system output on the vehicle Controller Area Network (CAN) bus indicates when the system activates for either a roll stability event or a yaw stability event. A roll stability event occurs when high lateral acceleration puts the vehicle in danger of imminent rollover. Yaw instability indicates that the vehicle is not following the intended heading of the driver resulting in either an understeer or oversteer condition. Due to the high coefficient of friction on the dry asphalt and the high center or gravity of the combination vehicle, testing on the dry test surface did not result in any yaw stability interventions by the ESC system. For this reason, tests were also conducted on a wet Jennite pad available at the Transportation Research Center (TRC) track test facility. This surface has a far lower coefficient of friction than that of the dry surface and could be used to induce some yaw instability. However, the surface did not have a peak coefficient of friction sufficiently low to activate yaw stability events during a maneuver without activating roll stability events.

In addition, ESC activation during maneuvers conducted on the wet pad were not as repeatable as the maneuvers conducted on the dry asphalt. However, in some test maneuvers, data clearly indicates that the ESC system sensed yaw instability and corrected for both understeer and oversteer conditions briefly during the maneuver. As expected on the initial understeer condition in the lane change maneuver conducted on the wet pad, the ESC system braked the inside drive

wheel to produce an additional moment to help the vehicle steer into the turn as desired. When the vehicle began the maneuver to return the vehicle to the original lane of travel, sufficient lateral force was generated to induce a roll stability control event and all wheels were braked as in the cases on dry pavement. However, immediately following the roll stability event a second yaw instability event occurred in which the vehicle was oversteering, possibly induced by the roll stability braking. In this case the system braked the front outside wheel to produce a restoring moment to straighten the vehicle and correct the oversteer condition. Note that Bendix literature refers to a tractor oversteer event as a jackknife condition and indicates that in this situation the system applies brakes to both the outside steer wheel and trailer wheels to straighten the vehicle. The reason that the trailer brakes were not activated by the system during the oversteer event is unclear.

In general the testing results indicated that the stability control system intervened qualitatively as expected based on ESC control strategies described in the available literature. However, a number of open questions remain regarding the specific strategy used by the Bendix commercial tractor ESC system. It is likely the case that the Bendix system has a number of "corrections" implemented in their algorithms to handle special conditions encountered when testing on actual vehicles. Exhaustive track testing would need to be employed in order to fully characterize the system. Such testing is obviously prohibitively expensive. As a result, the ESC algorithm developed for co-simulation may be validated in a future project phase for the specific maneuvers tested, however the simulated ESC algorithm may not accurately reproduce the behavior of the commercial system under all dynamic conditions that may be realized in simulation conditions.

Future Program Efforts

The proposed Phase B of the project entails the investigation of advanced heavy tractor trailer ESC system concepts. The baseline ESC algorithm developed in the Phase A co-simulation project will be validated against track test data obtained as part of that project. This ESC algorithm will serve as the baseline for comparison for any potential benefits of the advanced ESC system developed. The addition of new on-board system sensors will be investigated, while keeping an emphasis on cost effectiveness. In addition, a communication link between the vehicle tractor and trailer will be considered to enable the exchange of vehicle configuration information, sensor data, etc. Advanced ESC algorithms that will take advantage of the additional sensor and communication data will be developed to provide additional system robustness and performance. A design proposal for an advanced prototype ESC system for the HTRC "SafeTruck" vehicle that takes advantage of these improvements will be developed.

The research is to be conducted by Clemson University with assistance from Michelin Americas Research Company and Bendix Commercial Vehicle Systems. Michelin and Bendix will provide oversight and guidance on the development of advanced stability control system concepts. Michelin will also continue to provide support in the form of truck modeling as part of both the HTRC and ESC projects. In addition, National Instruments will provide support in the form of a LabVIEWTM software license to be used for ESC algorithm modeling and simulation.

The Phase B project is expected to last for one year, with the possibility of follow-on research in subsequent years. This second year of research will primarily consist of research on novel advanced ESC system concepts including sensors, vehicle communication, and algorithms.

Chapter 1 – General Overview

Background

A study by the Insurance Institute for Highway Safety on all types of road vehicles has found that Electronic Stability Control (ESC) systems "could prevent nearly one-third of all fatal crashes and reduce the risk of rolling over by as much as 80 percent." In light of these benefits, NHTSA has issued Federal Motor Vehicle Safety Standard 126, which mandates that all new light vehicles include ESC systems as standard equipment by September 2011. While the inclusion of ESC on heavy trucks is not yet mandated, increasingly the cost benefits of such systems are being emphasized by suppliers and OEMs and it is believed that legislation mandating ESC systems on heavy trucks is on the horizon.

It is well known that the interaction of vehicle dynamics and tire dynamics play a significant role in the overall vehicle handling and performance. This is true for heavy trucks, and significant effort is made to understand these interactions when designing new tires for a particular truck. One tool commonly used is numerical simulation on the computer. Mathematical models of the vehicle and tires of adequate fidelity can produce simulation results that are representative of the performance of the real vehicle. Increasingly passenger vehicles as well as heavy trucks are being outfitted with ESC systems that enhance vehicle stability at the limits of vehicle performance. While the vehicle and tire dynamics involved are well understood, the ESC algorithms are generally proprietary to the system suppliers and therefore their effect on the vehicle performance at the limits of handling cannot be readily simulated. A Hardware-In-the-Loop (HIL) simulation system that incorporates the ESC system electronic control unit hardware into the simulation will provide a means of using computer simulation to investigate these effects without having knowledge of the underlying control algorithms themselves.

State-of-the-art vehicle stability control systems operate by comparing vehicle dynamic behavior to a pre-determined non-linear time-invariant dynamic model of the vehicle. This generally results in acceptable behavior for passenger vehicles in which the load variations are minor due to passengers, luggage, etc. Key parameters such as sprung mass, moments of inertia and height of center of gravity only vary by a small percentage, making the model used in the ESC system adequate for minor variations. However, in a heavy truck there are substantial load variations due to the presence/absence of a trailer, trailer cargo mass, cargo distribution, height of cargo, etc. that could have a significant impact on the model. It is not clear that a stability control system tuned to one particular heavy truck configuration provides adequate performance over all vehicle configurations. In addition, it may be the case that the configuration of the ESC model provides the most conservative control action, and thus over-corrects for other load configurations. The HIL simulation system and models described above may be used to investigate the effect of load variations described above on a truck equipped with a commercial ESC system. In addition the system may serve as a simulation platform to investigate the effectiveness of adaptive ESC algorithms designed to overcome these limitations.

Project Team

This research was conducted by organizational participants from academia and private industry. Specifically, the partners in the project included:

- Michelin Americas Research Company (MARC)
- Clemson University (CU)
- National Instruments Corp.(NI)
- Other Heavy Truck Rollover Characterization (HTRC) project partners:
 - Oak Ridge National Laboratory
 - Battelle Memorial Institute
 - Western Michigan University

MARC Roles and Responsibilities

The primary roles and responsibilities of MARC in this project involved them as the primary industry technical lead. MARC's roles included simulation modeling, on-track testing, and data analysis. MARC's close technical relationship with Clemson University has facilitated a strong understanding of the role of tires in vehicle dynamics and stability.

Clemson University Roles and Responsibilities

Professors Thomas R. Kurfess and E. Harry Law of the Department of Mechanical Engineering led Clemson's effort on the research project. Doctoral candidate John Limroth provided the student labor on the project. CU had a leading role in all project tasks.

National Instruments Roles and Responsibilities

NI provided direct financial support as well as the core hardware and software components of the real-time HIL system. This included the PXI system for simulation of the vehicle and tire models, and the LabVIEW[™], LabVIEW[™] Real-Time, LabVIEW[™] Simulation and LabVIEW[™] FPGA software necessary for simulation development. NI also provided all I/O boards including data acquisition, Controller Area Network bus boards, and associated signal conditioning and cables.

Project Description and Objectives

The overall objectives of this research were to:

- Determine performance of the truck ESC system and validate against measured test track data.
- Understand the fundamental operation of the ESC system with respect to vehicle dynamics
- Determine the robustness of the ESC system with respect to vehicle configurations and various vehicle parameters

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- 1. A review of literature relevant to the operation of heavy truck ESC systems
- 2. Investigation of a Hardware-In-the-Loop (HIL) simulation system for truck/ESC system co-simulation
- 3. Investigation of an offline software co-simulation system
- 4. Track testing of a tractor/semi-trailer coordinated with Heavy Truck Rollover Characterization project
- 5. Initial investigation of the track testing results for ESC co-simulation system validation

Literature Review

A review of relevant literature was conducted to understand the state-of-the-art of heavy truck electronic stability control algorithms. A review of passenger car ESC algorithms was conducted to understand the fundamental operation of these systems since adoption of ESC in the passenger car market preceded that of the heavy truck market. In addition, a review of the relevant literature on ESC and roll control systems for articulated heavy truck was conducted.

In addition to ESC algorithms, the literature review included models of articulated trucks used for both control algorithms and simulation systems. Since a dynamic model of the articulated truck lateral handling was found to serve as the basis for determining desired vehicle behavior in ESC systems, a survey of low-order models suitable for processing on an ESC embedded controller was conducted. In addition, high-fidelity commercial tools capable of simulating tractor/semi-trailer systems such as TruckSim[®] and SIMPACK[®] were reviewed.

The literature review was provided previously in the project interim report, and is provided again in Appendix A– Literature Review for the sake of completeness.

Chapter 2 – ESC HIL Simulation Investigation

Investigation of the available information on input/output signals to the ECU resulted in the determination that insufficient details of the ECU operation were available for integration into an HIL system. This included unpublished requirements of the sensors/actuators for diagnostics purposes and proprietary communication buses used in the system. The initial project proposal included engaging the ESC ECU supplier, Bendix Commercial Vehicle Systems, as a partner in the project. However, as Bendix involvement occurred very late in the project timeline, the project focus was shifted away from an HIL simulation system and instead focused on a software co-simulation system.

HIL Project Tasks

Simulation Model Development

An initial TruckSim[®] model of the Volvo Test truck was developed by Michelin and provided to Clemson. Clemson acquired the TruckSim[®] license and a real-time license that would enable HIL simulation within LabVIEW^M Real-Time on the National Instruments hardware. The model provided by Michelin was verified on the host personal computer and basic handling maneuvers were simulated.

The first TruckSim[®] model of the Volvo Truck provided by Michelin has been tested in offline simulation on the host computer. While the model developed by Michelin incorporates the TruckSim[®] flexible frame option, this option is not included in the license obtained for this project and therefore the model feature has been disabled. This feature is not critical for the handling maneuvers necessary for ESC system testing in either offline or HIL simulation.

In addition to the TruckSim[®] model, a baseline 4 Degree of Freedom (DOF) simulation model was developed for the project. This model, based on first principles, provides an analytical model to which the TruckSim[®] model may be compared. The analytical model should provide insight into the ESC system operation.

ESC Control Algorithm Development

A basic state feedback controller for ESC on an articulated heavy truck has been developed. This controller is of a standard form as found in the literature described in Appendix A. The controller was initially tested against an arbitrary truck model before the Volvo TruckSim[®] model parameters were available. This standard controller is now being adapted to the specific Volvo test truck used in this project.

Integration of Michelin Tire Models

Models of the Michelin XZA3 and XDA tires were provided by Michelin with the Volvo VT830 TruckSim[®] model. An agreement was established between Clemson and Michelin for the

purpose of sharing this confidential tire information. These tire models will be used for initial model development and simulation.

ESC Hardware Acquisition

While quotes have been obtained for the ECU and peripheral ESC system hardware, this equipment has not yet been acquired. The reason for this is that the specific equipment required depends upon whether the actual system sensors and actuators are required. For example, the actuator valves may need to be present for the ECU to pass initial diagnostic tests on startup, and may also be necessary to ensure that the valve electrical characteristics detected by the ECU are within expected ranges. In addition, if Bendix is not willing to provide information regarding the local CAN communication bus between ECU, yaw angle sensors, and steer angle sensors, then these hardware components may need to be integrated into the system. The specific hardware will be acquired after the status of participation by Bendix has been established.

HIL Hardware Specification

The following hardware required for the HIL test system has been specified for the project:

- Middle Atlantic 21 Space Rolling Rack The rack will house all HIL hardware and includes two sliding shelves and a Tripp-Lite power strip.
- B&K Precision 13.8 VDC 4A Power Supply The 4 Amp power supply can be used to power auxiliary devices such as the yaw rate and steer angle sensor packs. A larger power supply capable of supplying 30 Amps for valve actuation is available in-house at Clemson and will be used for the project.

In addition, the National Instruments hardware required for the real-time simulator has been specified. This includes a PXI controller and chassis that will serve as the real-time computer for running TruckSim[®] simulation models. In addition, all I/O devices and signal conditioning hardware has been specified. The LabVIEW[™] software and associated modules required are also specified. The complete set of NI hardware is indicated in Table 1.

Controller and Options		
779886-33	NI PXI-8106 RT Controller	1
779302-1024	1.5 GB RAM	1
779660-01	USB English Keyboard and Optical Mouse	1
Modules and Options		
780340-01	NI PXI-7852R RIO Module	1
191667-01	SHC68-68-RDIO Cable (1m)	2
189588-01	SHC68-68-RMIO Cable (1m)	1
776844-01	SCB-68 Connector Block	1
Chassis and Options		
778636-02	NI PXI-1042Q Chassis	1
960597-08	PXI 8-Slot Factory Installation Service	1
763000-01	United States 120VAC	1
778643-01	PXI Rack Mount Kit (Front)	1
778643-02	PXI Rack Mount Kit (Rear)	1
0		
	NI 0151 D Series Expension Chassis	2
779008-01	NI 9151 K Series Expansion Chassis	2
779429-01	NI 9853 CAN Module	1
192017-02	NI CAN High-Speed Cable, 2m	1
779372-01	NI 9201 AI Module	1
779104-01	NI 9934 25pin D-Sub connector kit	2
779012-01	NI 9263 AO Module	1
779017-01	NI 9932 Backshell with T0-pos connector block	1
779006-01	NI 9401 DIO Module	1
779017-01	NI 9932 Backshell with 10-pos connector block	1
779097-01	NI 9904 Panel Mounting Kit	1
		· · ·
Software		
778694W-09	LabVIEW EPGA Module + 1 Yr SSP	1
777844W-09	NI JabVIEW Beal-Time Module + 1 vr SSP	1
776678-09	NI LabVIEW PDS + SSP	1
777136-01	LabVIEW Advanced Signal Processing Toolkit	1
778941-03	LabVIEW System Identification Toolkit	1
780050-09	LabVIEW Control Design and Simulation Module	1
779953-09	LabVIEW Statechart Module	1

Table 1. Specified National Instruments Hardware and Software.

HIL Hardware Acquisition

All hardware specified in the HIL Hardware Specification section above has been acquired. Figure 4 shows a picture of the rack and associated hardware for the HIL test system. The power strip and National Instruments Real-Time PXI controller and chassis are located at the top of the rack. The first sliding shelf contains the B&K power supply (right) and Bendix ESC ECU and Yaw Angle Sensor pack (left). The ECU and sensor pack were removed temporarily from the Volvo test truck for the purposes of system development. The actual components to be used in the HIL system have not yet been acquired.



Figure 4. Photograph. HIL Rack and Equipment.

HIL ECU Sensor and Actuator Integration

John Limroth met with several HIL simulation experts to discuss I/O aspects of the HIL system and the need for information from the ECU supplier. The HIL experts indicated a strong need for information from the supplier in the areas of diagnostic information, system initialization and I/O information.

Further investigation into the ECU requirements and meetings with Hardware-In-the-Loop experts have led to the determination that cooperation from the ECU manufacturer Bendix is critical. John met with several experts during National Instruments' NIWeek conference to discuss the project:

- Discussions with NI experts on CAN/J1939:
 - Sensor CAN-bus reverse-engineering possible, but will be difficult
 - J1939 bus standards are defined, but which signals are used (and when) by the ECU needs to be known
- Discussions with NI experienced HIL partners:
 - ECU switches power to actuator valves, therefore a high output (300W) power supply and actual valves are needed in the HIL system

- Valves are actuated via voltage, but current on the line is monitored by the ECU to determine valve position feedback. These electrical signals are difficult to simulate without actual valves.
- ECU diagnostics are critical for development and debugging. A diagnostic interface tool is needed, and the sensor/actuator impedances must be matched correctly.

Information Required of ECU Supplier

Michelin organized a meeting with Bendix to discuss participation in the project. While the initial indication from Bendix was positive, at that meeting no specific plan was established for sharing information regarding the ECU for this project. The partnering discussions are taking far longer than was anticipated when the original schedule was developed. If Bendix is involved and supportive, they could provide much needed aid in the development of the I/O for the HIL system. Further investigation into the HIL system and meetings with external experts has led to the conclusion that cooperation from Bendix is critical to the successful implementation of the HIL system. The critical nature of the information has been relayed to Michelin who is maintaining the relationship with Bendix. Michelin will continue to pursue Bendix and request the necessary information.

The following is a list of specific information needed from Bendix for the development of the HIL system:

- Details of CAN bus communication with Yaw Angle (YAS) and Steer Angle (SAS) sensor packs
- Details of powertrain J1939 bus communication (which signals, and when?)
- Sensor/actuator inductance/resistance matching for diagnostics (e.g. valve "chuff" test)
- Specific hardware needs for HIL system
 - Valves absolutely required for HIL?
 - Additional sensors or hardware?
- Need to simulate Power Line Carrier (PLC) signal for trailer ECU?

Alternative Project Plan: Truck/ESC Co-Simulation

An offline software simulation will be developed in which the TruckSim[®] model will be cosimulated with an ESC control algorithm implemented in LabVIEWTM. This will provide an alternative configuration that will allow closed loop testing of the vehicle model, tire models, and ESC algorithm, although without the actual ESC algorithm of the commercial ECU that would be present in the HIL test system. The system will also provide an interface to the ESC system that will be used in the real-time implementation of the HIL system should ECU information be forthcoming from Bendix.

Chapter 3 – ESC Software Co-Simulation

Software Co-Simulation System

Due to the lack of available information required for the implementation of a HIL simulation system during Phase A of the project, a pure software co-simulation strategy was pursued. In this case the vehicle model is not run in real-time on dedicated hardware, but instead is simulated on a standard desktop PC. However, a second simulation environment is used to model the behavior of the ESC control algorithm. For this project, LabVIEWTM was used to implement an ESC algorithm that is representative of those found in the available literature. LabVIEWTM and TruckSim[®] are used together to "co-simulate" the complete vehicle with an ESC system. Information must be exchanged at each simulation timestep between the two tools to ensure that the complete system is accurately simulated. A schematic view of the software co-simulation system is shown in Figure 2.

Yaw Stability Control

Heavy truck ESC systems found in the literature typically implement a state feedback control scheme with a linear vehicle model used to determine "desired" states of the vehicle. A schematic view of the control strategy used in the co-simulation system is provided in Figure 3. A critical component of the algorithm is the use of a linear dynamic model to determine the desired vehicle states based on vehicle speed and steering input. The model is the "articulated bicycle" handling model commonly found in the literature with the following states: tractor lateral velocity, tractor yaw rate, trailer relative articulation angle and articulation rate.

The yaw stability portion of the controller is shown in the figure. Here yaw rate and lateral acceleration are used as model outputs and are compared to the measured values from TruckSim[®] model outputs corresponding to actual sensors used on the real vehicle. A threshold is used to prevent intervention under normal driving conditions and feedback gains are applied to the state deviations. The resulting control output includes logic to apply differential braking pressures to appropriate axles of the vehicle to correct the instability. These pressures serve as setpoints to individual wheel ABS controllers also implemented in LabVIEW[™], but not depicted in the simple schematic figure shown.

Linear Model Development

As seen in Figure 3, a simple dynamic model of the vehicle yaw behavior is a critical component of the yaw stability control algorithm. For this reason great care was taken to ensure that the linear model used in the algorithm accurately reflects the true behavior of the vehicle. The linear model used is the classic 4th order "articulated bicycle" model found in Genta [1]. Longitudinal dynamics of the vehicle are not included in the model as the vehicle speed is assumed to be approximately constant. The input to the model is the steering wheel angle, which is sensed in

the actual vehicle ESC system. The model used for this research is based on the Genta model and includes these forces on the vehicle lateral motion:

- 1. Tire lateral force due to cornering stiffness
- 2. Tire aligning moment due to aligning stiffness
- 3. Tire longitudinal force
- 4. Longitudinal aerodynamic drag
- 5. Aerodynamic yaw moment.

The states used for this model are:

- 1. Lateral velocity, v_y
- 2. Yaw rate, $d\psi/dt$
- 3. Trailer articulation angle, θ
- 4. Trailer articulation angular rate, $d\theta/dt$.

The model outputs include all vehicle states, as well as the tractor and semi-trailer lateral accelerations. These lateral accelerations were added to the model since these are simple linear combinations of the vehicle states and their time derivatives as shown in Equation 1. It should be noted that only the vehicle yaw rate and lateral accelerations are actually sensed by the ESC system as no sensors are provisioned for measuring lateral velocity or trailer articulation.

$$A_{y} = \dot{v}_{y} + v_{x}\dot{\psi}$$
$$A_{y,s} = \dot{v}_{y} + v_{x}\dot{\psi} - c\ddot{\psi} - a_{s} \quad \ddot{\psi} - \ddot{\theta}$$

Equation 1. Tractor and Semi-Trailer Lateral Accelerations.

The linear model was developed using the geometrical and other properties of the Volvo VT830 tractor and Utility flatbed trailer used for testing in the Heavy Truck Rollover Characterization study. The model was implemented in LabVIEWTM and used in a co-simulation manner in order to directly compare the outputs of the linear model with those of the high-fidelity TruckSim[®] model. The steering input used in the TruckSim[®] simulations was output to LabVIEWTM and the linear model run in parallel to the TruckSim[®] model. The signals corresponding to the model outputs were also output from TruckSim[®] and plotted against the outputs of the linear model. An example for a step steer type maneuver is shown below in Figure 5 and Figure 6.





Figure 5. Graphs. Step Steer Model Response, 100 kph (a).



Figure 6. Graphs. Step Steer Model Response, 100 kph (b).

Note that the steering wheel input for the maneuver shown is about 20 degrees, which results in a road-wheel angle of ~1 deg. This is well within the linear range of the slip angles at the tires – the tire models used in TruckSim[®] exhibit linear lateral force response to slip angles up to about

5 degrees. In addition the resulting angles are well within the region in which small angles may be approximated with linear functions. Because of these reasons, the linear model should be expected to very closely approximate the actual behavior of the real vehicle under these circumstances. However as seen in Figure 5 and Figure 6, the model does not accurately reflect the vehicle states, especially during steady state cornering.

As a result, a key finding of the research is that the vehicle states of the classic "articulated bicycle model" did not match those of the complete nonlinear vehicle model in TruckSim[®], even for low severity maneuvers within the linear handling region of operation of the tires. Therefore this simple model would be difficult to use to predict desired behavior of the vehicle under normal conditions since such state deviations may be interpreted as yaw instability and result in unnecessary intervention of the ESC system. For this reason the linear model was further investigated to determine the source of these model deviations.







Figure 7. Graphs. Simplified TruckSim[®] Model Investigation, 100 kph (a).



Figure 8. Graphs. Simplified TruckSim[®] Model Investigation, 100 kph (b).

A modified TruckSim[®] model was created with greatly simplified suspension models for the tractor and trailer. While the original suspension models are fully nonlinear and incorporate all measured effects from kinematics and compliance testing, the simplified suspension models only incorporate linear spring force and damper force coefficients. A comparison of these models can be seen in the black and red traces of the plots shown in Figure 7 and Figure 8. The simplified TruckSim[®] model was found to be in very good agreement with the linear model computed using LabVIEWTM and shown in Figure 5 and Figure 6.

Individual suspension effects were incorporated into the simplified suspension models in TruckSim[®] in an effort to identify the key characteristics responsible for the deviations from the complete suspension model. After much trial and error, a modified linear suspension model with some of the measured kinematic and compliance effects incorporated was found to be in very good agreement with the original fully nonlinear model. This can be seen in the green traces of Figure 7 and Figure 8. The specific suspension properties found to be responsible for the majority of the model deviations are:

- 1. Axle roll steer
- 2. Tire lateral force compliance steer
- 3. Tire aligning moment compliance steer (primarily on front axle).

The "Simple Modified" model results shown in Figure 7 and Figure 8 are from a model with the linear spring and force components as well as linear coefficients corresponding to these three identified properties. In order to realize a linear model for ESC algorithm use, strategies were devised to incorporate these three effects were into the standard linear vehicle model.

Axle Roll Steer

Axle roll steer is a kinematic effect by which an axle causes a steer angle of the wheels as the vehicle sprung mass rolls in response to lateral forces on the vehicle. This effect is common in trailing arm suspensions such as those on the tractor drive axles and trailer axles. A simple linear coefficient can be used to represent axle wheel steer as a function of relative roll angle between an axle and the vehicle sprung mass as seen in Equation 2.

$$\delta_{rollsteer,i} = \varepsilon_i \varphi_{suspension,i}$$

Equation 2. Axle Linear Roll Steer Model.

In order to incorporate this effect into the model, a quasi-steady state model of the roll steer effect was employed since a roll degree of freedom is not included in the present vehicle handling model. The steady state model assumes that the vehicle roll angle is that which produces equilibrium between the roll moment generated from lateral acceleration and the roll moment generated from the roll stiffness of the suspension. The specific implementation in the model is shown in Equation 3. The lateral acceleration is used to determine the vehicle roll angle at equilibrium due to the equivalent suspension roll stiffness k_{ϕ} . This equivalent roll stiffness is

due to suspension and axle roll stiffness combined in series at each axle, with all axles assumed to be acting in parallel on the vehicle sprung mass. The axle roll relative to the vehicle roll angle can then be found from the relative suspension roll stiffness and axle roll stiffness due to the tires for each axle. The roll steer effect is then implemented using Equation 2. As a result the roll steer effect at each axle is simply a linear function of the yaw rate and is easily incorporated into the linear vehicle model.

$$\begin{aligned} A_{y} &= \dot{v}_{y} + v_{x}\dot{\psi} \approx v_{x}\dot{\psi} \\ \varphi &= k_{\varphi}A_{y} \\ \varphi_{susp,i} \approx \frac{k_{axle,i}}{k_{axle,i} + k_{susp,i}} \varphi \end{aligned}$$

Equation 3. Quasi-Steady State Roll Model.

Lateral Force and Aligning Moment Compliance Steer

Lateral force compliance steer is an effect in which the lateral force generated at the wheels induces a steer effect due to compliance of the suspension mechanism. Again, such an effect is present especially on trailing arm suspensions. Aligning moment compliance steer is a similar effect where the aligning moment generated at the wheels due to pneumatic and mechanical trail produces additional steer effect due to suspension compliance. This effect is especially present on the tractor steer axle due to compliance effects in the steering mechanism.

The lateral force compliance steer effect can be approximated for a small region of operation using a linear coefficient as seen in Equation 4.

$$\delta_{cF_y} = k_{cF_y}F_y$$

Equation 4. Linear Lateral Force Compliance Steer Model.

The net result is a slight reduction in the lateral force generated for a given slip angle of the wheel since the slip angle itself is slightly reduced due to the compliance steer effect. Assuming a nominal wheel slip angle α_0 without compliance steer, the lateral force generated at the wheel due to axle linear cornering stiffness C_{α} is shown in Equation 5. Thus the compliance steer effect can be modeled by changing the axle cornering stiffness as shown in Equation 6. Note that due to sign conventions the lateral force compliance steer coefficient for tractor and trailer axles all have negative values. Thus the net effect is a slight reduction in cornering stiffness.

$$F_{y} = -C_{\alpha} \alpha$$
$$= -C_{\alpha} \alpha_{0} - \delta_{cF_{y}}$$
$$= -C_{\alpha} \alpha_{0} - k_{cF_{y}} F_{y}$$
$$= -\frac{C_{\alpha}}{1 - k_{cF_{y}} C_{\alpha}} \alpha_{0}$$

Equation 5. Lateral Force Model.

$$C_{\alpha,i}^* = \frac{C_{\alpha,i}}{1 - k_{cF_{\nu},i}C_{\alpha,i}}$$

Equation 6. Effective Cornering Stiffness Due to Lateral Force Compliance.

The lateral force compliance steer also affects the tire aligning moment in a similar manner. This can be represented with an effective aligning stiffness as shown in Equation 7.

$$C_{M_{z},i}^{*} = \frac{C_{M_{z},i}}{1 - k_{cF_{y},i}C_{\alpha,i}}$$

Equation 7. Effective Aligning Stiffness Due to Lateral Force Compliance.

Note that the aligning moment compliance steer effect can be modeled in a manner completely analogous to the lateral force compliance steer effect. The net result is an effective change in cornering stiffness and aligning stiffness at each axle as shown in Equation 8 and Equation 9. Note the tractor and trailer axles have positive values for aligning moment compliance steer coefficient, and thus the net result is a reduction in both cornering stiffness and aligning stiffness.

$$C_{\alpha,i}^{**} = \frac{C_{\alpha,i}^{*}}{1 + k_{cM_{*},i}C_{M_{*},i}}$$

Equation 8. Effective Cornering Stiffness Due to Aligning Moment Compliance.

$$C_{M_z,i}^{**} = \frac{C_{M_z,i}^*}{1 + k_{cM_z,i}} C_{M_z,i}$$

Equation 9. Effective Aligning Stiffness Due to Lateral Force Compliance.

The result of the described modifications to the linear tractor/semi-trailer model can be seen in Figure 9 and Figure 10. Each of the linear models represents the addition of each additional effect of axle roll steer, lateral force compliance steer and aligning moment compliance steer. The linear model that includes all three effects can be seen to very closely match the original complete nonlinear TruckSim[®] model. During transients some deviation can be observed in the

states. This is most likely due to the steady state assumption of the axle roll steer model, or to other minor nonlinear effects not modeled.









Figure 9. Graphs. Modified Linear Tractor/Semi-Trailer Model, 100 kph (a).





Figure 10. Graphs. Modified Linear Tractor/Semi-Trailer Model, 100 kph (b).

Yaw Stability Control Algorithm

A yaw stability control system was devised in which the yaw rate was used as a state feedback variable. The structure of the controller used is shown in Figure 3. The linear model described above was generated for vehicle speeds from 10 to 100 kph in increments of 10 kph. During execution of the algorithm, the linear model coefficients were linearly interpolated based on the current vehicle speed in order to yield an accurate model of the vehicle lateral dynamics. The interpolation was found to yield good accuracy without requiring excessive computation of model coefficients or excessive memory requirements for additional models at smaller speed increments.

A threshold of 2 deg/sec for error between measured yaw rate and desired yaw rate (from the linear model) was selected. This threshold was found to be sufficient to avoid unnecessary ESC system intervention, yet still provide quick actuation of the system when significant error in yaw rate was detected. A yaw rate feedback gain of 0.05 brake demand % / (deg/sec) was determined by hand tuning the model to yield sufficient control. The resulting brake demand % was distributed to the inside drive wheels in the case of understeer and to the outside steer wheel and trailer wheels in the case of oversteer. An ABS control system was also modeled in LabVIEWTM to avoid wheel lock when braking during ESC events. The ESC system determines a command brake pressure for each wheel ABS control system, which then modulates individual brake pressures to ensure that appropriate wheel slip is maintained. Note that the tractor has four brake actuation valves: two to control right and left steer wheels, one to modulate both right drive wheels and one to modulate both left drive wheels. The trailer system has a single valve to control all four trailer wheels.

Co-simulation tests were executed using a lane change maneuver on a low friction surface ($\mu = 0.2$) at 100 kph implemented in TruckSim[®]. The results of the yaw stability control system can be seen in Figure 11 and Figure 12. Without ESC activation, the tractor and trailer develop significant amounts of sideslip and the combination vehicle experiences large yaw oscillations when exiting the maneuver. With the designed ESC system co-simulated with the TruckSim[®] model executing the same maneuver, the vehicle sideslip and trailer oscillations were significantly reduced throughout the maneuver. In addition, the driver steering effort is significantly reduced when the ESC system is activated, as seen in the Road Wheel Angle (RWA) plot of Figure 11.



Figure 11. Graphs. ESC System Co-Simulation on Low Friction Surface, 100 kph (a).



Figure 12. Graphs. ESC System Co-Simulation on Low Friction Surface, 100 kph (b).

Chapter 4 - Track Testing

Track testing of a tractor and tanker semi-trailer was conducted to provide data to use for the validation of the ESC co-simulation project. This testing was conducted in conjunction with the HTRC Phase B project. Members of both project teams collaborated on all test planning phases, test execution, and post-test analysis of the data. While some test instrumentation and maneuvers were designed specifically for the ESC Co-Simulation project, the majority of instrumentation and test maneuvers were designed to yield data of use for both projects. The actual track testing was realized using funds from both projects. A summary of the tests executed is shown in Table 2.

		Runs (Low	Runs (High	ESC (tractor	Vehicle	Time per	Total
Request	Maneuver	speed)	speed)	-trailer)	Config	pass	Time
HTRC	Ramp steer	2	10	Off-off	3	10	360
HTRC	Ramp steer	0	5	On-on	3	10	150
HTRC	Ramp steer	0	5	On-off	2	10	100
HTRC	Ramp steer	0	5	Off-on	2	10	100
HTRC	Step-steer	6	15	Off-off	3	5	315
HTRC	Step-steer	0	5	On-off	3	5	75
HTRC	Step-steer	0	5	Off-on	2	5	50
CS	Step-steer	0	5	Off-off	2	5	50
CS	Step-steer	0	5	On-off	2	5	50
HTRC	Dry Lane Change	6	15	Off-off	3	5	315
HTRC	Dry Lane Change	0	5	Off-on	2	5	50
HTRC	Dry Lane Change	0	5	On-off	3	5	75
CS	Wet Lane Change	0	5	Off-off	2	5	50
HTRC	Wet Lane Change	0	5	On-on	2	5	50
HTRC	Wet Lane Change	0	5	Off-on	2	5	50
CS	Wet Lane Change	0	5	On-off	2	5	50

Table 2. Track Testing Maneuvers.

The vehicle tested was a Volvo VT830 tractor with an LBT tanker trailer. Four maneuvers were used for testing: slowly-increasing ramp steer, step steer, dry double lane change and wet double lane change. The first column of Table 2 shows which project is the primary end user of the test results, the Heavy Truck Rollover Characterization project (HTRC) or the ESC co-simulation project (CS). The Volvo tractor was equipped with a Bendix full ESC system and the trailer was equipped with an independent Bendix roll stability control system. Tests for the various maneuvers were executed with various combinations of tractor and trailer stability control systems enabled/disabled as indicated. In addition, different vehicle configurations were tested. All maneuvers were repeated for vehicle configurations with standard dual tires with the tanker fully loaded in both a "high" center of gravity (CG) configuration and a "low" CG configuration. In addition, some maneuvers were also repeated with a configuration of new generation single-wide tires and the "high" CG configuration. The Vehicle Config column in Table 2 indicates whether all three configurations were tested or only the two configurations with dual tires. A steering robot was used to conduct all tests in a manner that would yield data for multiple runs

that was far more repeatable than could be realized by a human driver following a desired steering profile or vehicle path.

For the ESC Co-Simulation project, the objective was to produce data that could be used to validate the ESC co-simulation system. However, due to scheduling delays for the track testing, the testing was conducted at the end of the time period defined for Phase A of the ESC Co-Simulation project. For this reason, a thorough validation of the ESC co-simulation system using the track test data is deferred to the proposed subsequent Phase B of the project. After completion of the testing, an initial analysis of the track test data with tractor ESC systems enabled and disabled (but with trailer stability control disabled) was conducted. The purpose of this investigation was a qualitative analysis to determine whether the logic applied to the baseline ESC algorithm developed based on available literature matched the behavior of the commercial ESC system.

This section provides an overview of test results of tractor ESC intervention on all maneuvers performed during track testing of the Volvo tractor and LBT tanker semitrailer during May 2009. Observations are made on each intervention, and aspects of the ESC system behavior that are not fully understood are indicated.

Slowly Increasing Ramp Steer Maneuver

A slowly increasing ramp steer test is used to simulate quasi-steady state conditions. For this test the steering wheel angle is increased at a rate of 10 deg/sec while the driver attempts to maintain a constant speed of 30 mph (unless the ESC system intervenes with throttle control). Figure 13 shows the lateral acceleration and tractor ESC system intervention for the ramp steer test with the high CG configuration. Note that the trailer ESC system is disabled in this "on-off" configuration. This plot shows that the roll stability control (RSC) in the tractor ESC system disables the engine torque output approximately a half second before the brakes are actuated.



Figure 13. Graph. Slowly Increasing Steer Test RSC Intervention.

All of the slowly increasing ramp steer tests consistently indicate a lateral acceleration of ~0.3 g at tractor RSC intervention. The level of lateral acceleration before intervention is approximately the same for both high and low CG cases. In the low CG case the RSC application may be at slightly higher lateral acceleration, but this is difficult to say conclusively.

Figure 14 shows braking application at each wheel during ramp steer test. Here "LAX1PRES" is the brake pressure at the left wheel of axle 1 (steer axle). Axle 3 is the tractor rear drive axle and is not shown in the plot since both drive axles share common valves for the left and right sides and therefore have the same pressures. Similarly LAX4PRES (brake pressure of left wheel of 4th axle) is representative of the brake pressure at all four trailer wheels since only one valve actuates all wheels. In the RSC intervention for the slowly increasing steer test the tractor brakes are not applied. The reason for this is unknown, but one possible explanation is that the rollover threshold for the fully loaded trailer is assumed to be much lower than that of the tractor. Applying brakes to the trailer only would ensure that the tractor is not braked unnecessarily when there is no trailer load (i.e. bobtail). However, since the trailer load is sensed by the controller it is unclear why it would not brake the tractor as well in this situation. Note that the trailer brakes appear to be pulsed at a frequency <1 Hz. It is not clear if this is normal ABS operation in response to wheel slip or some other operation of the tractor ESC and/or ABS systems.



Figure 14. Graph. Slowly Increasing Steer Test Wheel-End Brake Pressures.

Step Steer Maneuver

The step steer test is a quick ramp in 1 second to 170 deg at a speed of 33 mph. The test is conducted "dropped throttle" in the sense that the tractor is put in neutral and allowed to coast while the steering robot and data acquisition trigger when the target speed is reached. Figure 15 shows high CG configuration step steer with tractor ESC on and trailer ESC off. Note that lateral acceleration builds quickly as the steering wheel ramps. In this case the tractor RSC system activates braking, but does not intervene in the engine. Presumably this is because the

test was conducted with "dropped throttle" and thus no torque was demanded of the engine anyway.



Figure 15. Graph. Step Steer Test RSC Intervention.

Note that the intervention occurs at much higher lateral acceleration than in the slowly increasing ramp steer case. There are two possible explanations for this. First, there may simply be a dwell time of ~0.3 seconds after the $0.3 \text{ g} (3 \text{ m/s}^2)$ threshold is reached before the system intervenes. This would be noticeable in the step steer case, but not in the slowly increasing steer case since it is effectively a steady state test. An alternative possibility is that the system assumes a higher rollover threshold for the tractor than that of the trailer. In the dynamic step steer test, the lateral acceleration in the trailer lags that of the tractor significantly, as can be observed in Figure 16. Although this lateral acceleration data is noisy and the sign is reversed from the signal returned from the ECU, the lateral acceleration of the trailer clearly lags that of the tractor by about 0.5 seconds. At ~1.6 seconds when tractor ESC intervention begins as seen in Figure 15, Figure 16 shows that the trailer lateral acceleration is only at ~2 m/s^2.



Figure 16. Graph. Step Steer Test Tractor (Red) and Trailer (Blue) Lateral Acceleration.

The braking shown in Figure 17 indicates that the tractor ESC system implements a complex actuation of brakes on the various wheels. Note that at first all 6 wheels of the tractor are braked ~ 0.2 seconds before the trailer brakes are applied at all. This is in contrast to the slowly increasing steer case where the trailer brakes were applied but the tractor brakes were not applied at any time during the maneuver. One possible explanation for this would be the dynamic versus steady state explanation given above. Since the system detects high lateral acceleration of the tractor before that of the trailer, the brakes may be applied only to the unit that has exceeded its lateral acceleration threshold.



Figure 17. Graph. Step Steer Test Wheel-End Brake Pressures.

Also note initially all tractor brake pressures are applied equally. At about 0.1 seconds into the braking intervention, the brake pressures of the left wheels of the tractor drive axles seem to be capped at approximately 25 psi. They are then held at this level for the remainder of the intervention. This may be due to the fact that the step steer maneuver is performed to the left and therefore these wheels are unloaded due to lateral load transfer on the drive axles. Further brake actuation might induce wheel slip on these wheels. At about 0.25 seconds into the braking actuation, a bias in brake pressure appears between the front axle and right drive axle brake pressures. The brake pressure of the front axle is maintained at ~20 psi below that of the right drive wheels for the remainder of the intervention. The reason for this bias is unclear.

Lane Change Maneuvers

Design of Open-Loop Lane Change Maneuver

Clemson took the lead role in designing an open-loop steering profile for the steering robot that would approximate the path of the vehicle during an emergency double-lane change maneuver. The TruckSim[®] vehicle model was used with an updated preliminary model of the tanker to design the maneuver. First, a simple driver steering model was used in TruckSim[®] to simulate a driver attempting to follow a profile that effected a double lane change maneuver. The resulting driver steering input profile was analyzed and a piecewise linear profile suitable for the steering

robot was designed to approximate the steering profile input by the driver model. The amplitudes and times of points in the profile were varied and simulated again in TruckSim[®] to make sure that the designed open-loop profile resulted in a full double lane change with the vehicle following approximately the same track at the end of the maneuver as in the beginning of the maneuver. Subsequent track testing of the actual vehicle using the designed steering robot profile resulted in a maneuver that closely approximated a double lane change maneuver, but with far more repeatable results than can be realized by a human driver. For many of the tests conducted, the amplitude of the steering profile was increased in order to increase steering severity and induce wheel lift at the relatively low speeds that could be realized on the test track.

Dry Lane Change

The lane change maneuver is an open-loop maneuver implemented by the steering robot intended to simulate a double lane change maneuver. The designed steering profile is shown in Figure 18. Note that speed is maintained at constant speed by the driver (unless tractor ESC system intervenes with throttle control). The speed was determined from the speed required for wheel lift on the trailer for each loading condition. The speed for wheel lift in the high CG condition was determined to be 32 mph, while it was 38 mph for the low CG configuration.



Figure 18. Graph. Double Lane Change Open-Loop Steering Profile.

A ground trace of a run with ESC disabled is shown in Figure 19. Note that the vehicle experiences a lateral displacement of ~ 5 m, which is slightly more than a typical lane change of 3.5 m. This amplitude was selected to achieve wheel lift on the trailer without requiring extremely high rates of speed as would be required for a lateral displacement of one lane. Note also that the vehicle returns to roughly the original trajectory. A slight difference in heading can be observed due to a slightly different vehicle speed from that used to design the maneuver and differences between the simulation model and actual vehicle.



Figure 19. Graph. Dry Double Lane Change Ground Track.

Dry Lane Change High CG Configuration

The activation of the tractor ESC (RSC) system during the dry lane change maneuver is shown in Figure 20 and the wheel-end brake pressures are shown in Figure 21. The Yaw Stability Control (YSC) algorithm of the tractor ESC system did not intervene in any of the tests on the dry surface. During the first intervention, the behavior of the system is very similar to that of the step steer maneuver, except that the engine and brake intervention are initiated simultaneously. This can be attributed to the fact that the lane change maneuvers were implemented with throttle input to maintain speed while the step steer test was conducted dropped throttle. The results shown in these figures were very consistent over five repeat runs of this same lane change test. During the first intervention the system performs as in the step steer: tractor brakes before trailer brakes, cap on brake pressure to inside drive wheels and steer/drive axle brake bias at high pressures. Notice that as the lateral acceleration reverses direction while ESC is still intervening, the cap on the left drive wheel brake pressures is relaxed.



Figure 20. Graph. Dry Lane Change High CG Test RSC Intervention.

As the vehicle begins to initiate the return to the original lane, a second ESC intervention is observed. In this case the engine intervention occurs before brake intervention as with the step steer maneuver. During this second intervention, the tractor braking leads the trailer braking as before, however at no time are the steer axle wheels braked. The reason for the lack of braking of the steering wheels at this point in the maneuver is unknown. The drive axles are braked with a slight bias to the left (now outside) wheels.



Figure 21. Graph. Dry Lane Change High CG Test Wheel-End Brake Pressures.

Dry Lane Change Low CG Configuration

Figure 22 and Figure 23 show similar plots for the dry lane change maneuver for the low CG configuration. While the brake pressures for the first tractor ESC intervention are similar to those of the high CG configuration, the brake actuation during the second intervention is significantly different in nature. This is likely primarily due to the fact that the driver attempts to maintain 38 mph instead of 32 mph, however the reasons for this different brake actuation are unclear. In the low CG case, the steer axle and left (outside) drive axle wheel brakes actuate with a bias to the steer axle. In this case the right (inside) drive axle wheels are not braked at all. Note that these observed differences in the second ESC intervention were consistent among repeated tests of both the high and low CG configurations. Obviously there is some complex logic used to decide the levels of brake activation to individual wheels that is not immediately understood without access to the proprietary data of the ESC supplier. The differences may be due to the understeer/oversteer conditions of the vehicle during the second ESC intervention.



Figure 22. Graph. Dry Lane Change Low CG Test RSC Intervention.



Figure 23. Graph. Dry Lane Change Low CG Test Wheel-End Brake Pressures.

Wet Lane Change

The wet lane change maneuver is similar to the dry lane change maneuver except that it is executed on a wet Jennite surface that has a reported coefficient of friction of ~0.3. The objective of this test is to activate the yaw stability control (YSC) part of the ESC system. Again, an open-loop maneuver implemented by the steering robot is used to simulate a double lane change maneuver. The designed steering profile is shown in Figure 26. The wet tests were conducted as "dropped throttle" tests to increase consistency. The profile amplitude was increased significantly compared to the dry tests in order to induce understeer/oversteer effects at relatively low speeds. This was primarily due to physical size constraints of the wet Jennite pad used for testing.

Unfortunately, the wet Jennite pad did not have a low enough coefficient of friction to consistently induce yaw instability without excessive lateral acceleration. As a result, the roll stability control was almost always activated during test runs in which yaw stability control was

activated, although not necessarily at the same time. In addition, the yaw stability control was not activated consistently. For example, during five repeat test runs at 28 mph for the high CG case, YSC braking was only activated on three of the five runs. The analysis of data in Figure 24 and Figure 25 below is taken from a test run executed at 27 mph, which was the only test run that clearly indicated both oversteer and understeer correction.

Figure 24 shows the tractor lateral acceleration and tractor RSC and YSC system activations during a wet double lane change maneuver at 27 mph for the high CG configuration. Note that RSC engine activation is included simply for comparison to previous plots – the engine was not deactivated since the test was conducted dropped throttle. YSC events occurred 2 seconds into the maneuver and again at ~6 seconds while an RSC event occurred at ~5 seconds.



Figure 24. Graph. Wet Lane Change High CG Test RSC and YSC Intervention.

Figure 25 shows the individual wheel brake pressures during the wet test. During the first YSC event at ~2 seconds, the left drive wheels were briefly braked indicating an understeer correction. As expected, the steer and trailer wheels were not braked while correcting for understeer.



Figure 25. Graph. Wet Lane Change Low CG Test Wheel-End Brake Pressures.

Figure 26 shows the ESC system yaw rate sensor signal and measured lateral velocity during the maneuver. (Note that the yaw rate sensor signal was incorrectly labeled simply "Yaw" in the data acquisition system.) A significant lag in the buildup of yaw rate seems to trigger the understeer correction.



Figure 26. Graph. Wet Lane Change High CG Test Tractor States and YSC Intervention.

A sustained lateral acceleration of -3 m/s seems to have triggered the RSC event. The braking actuation during this event is very similar to that of the second intervention on the high CG configuration/dry lane change maneuver. The steer axle brakes are not applied, but the drive axles are braked followed by the trailer axles. There is a slight bias towards the left (outside) drive wheel, but this bias changes to the right drive wheel momentarily at times during the intervention as the braking levels increase or decrease.

At ~5.8 seconds into the maneuver a second YSC intervention occurs during which the left steer wheel is braked, indicating an oversteer correction. In Figure 26 two factors seem to contribute to the oversteer condition. First, the yaw rate to the right continues to increase even as the steering wheel angle is decreased starting at 5.2 seconds. Second, there is a significant increase in lateral velocity of the tractor prior to the YSC intervention. If the ESC controller estimates lateral velocity and/or vehicle sideslip, this may be a condition that triggers the oversteer correction. Note that in this particular maneuver the braking of the drive axles during RSC intervention may actually induce the vehicle sideslip and thus the subsequent oversteer correction, the ESC system does not apply the trailer brakes. Bendix advertises that during jackknife prevention

both the outside steer wheel and trailer wheels are braked. However, it is possible that in this case the system detected an oversteer condition of the tractor but not an impending jackknife situation in which the trailer is also understeering.

Comparison of Vehicle States on Wet and Dry Maneuvers

Most electronic stability control system algorithms described in the literature operate on deviations of vehicle states from expected states as determined from a simple vehicle model. To evaluate the effectiveness of these state comparisons, relevant vehicle states are compared in this section for the same lane change maneuver on both wet and dry surfaces. These tests were conducted with tractor and trailer ESC systems disabled with a "dropped throttle" maneuver triggered at 27 mph. Figure 27 and Figure 28 show the steer input and vehicle speed during both maneuvers, which can be observed to be very consistent between the tests. Note that although in these figures the file names seem to indicate different initial speeds, both runs shown were executed with an initial speed of 27 mph. Also, in the file names the ESC naming convention "off-off" indicates that both tractor ESC and trailer ESC were disabled. The file name for wet maneuvers has a designation "xxx-off" indicating that the file contains some runs with tractor ESC disabled.



Figure 27. Graph. Dry and Wet Lane Change Steering Input Comparison.



Figure 28. Graph. Dry and Wet Lane Change Vehicle Speed Comparison.

Figure 29 through Figure 33 show various vehicle states as measured by both the tractor ESC system sensors and external data acquisition systems during the manuevers. Note that a comparison of tractor lateral velocity is not included in the plots provided. Unfortunately, there was an error in the tractor inertial/GPS sensor unit during these wet tests that resulted in corrupted data. However, the trailer inertial/GPS sensor unit did work properly and a comparison of trailer lateral velocity is included.



Figure 29. Graph. Dry and Wet Lane Change Tractor Yaw Rate Comparison.



Figure 30. Graph. Dry and Wet Lane Change Tractor Lateral Acceleration Comparison.





Figure 31. Graph. Dry and Wet Lane Change Trailer Articulation Angle Comparison.

Figure 32. Graph. Dry and Wet Lane Change Trailer Lateral Acceleration Comparison.



Figure 33. Graph. Dry and Wet Lane Change Trailer Lateral Velocity Comparison.

Chapter 5 – Conclusion

This report summarizes the findings of the initial Phase A of NTRCI project U13: Co-Simulation of Heavy Truck Tire Dynamics and Electronic Stability Control System. This project was led by Clemson University with significant engineering support from Michelin Americas Research Company and funding and equipment support from National Instruments. A review of existing literature on the state of the art of ESC systems for articulated heavy trucks was conducted. The requirements for a Hardware-In-the-Loop (HIL) system for heavy truck ESC systems were investigated. A co-simulation platform was developed using LabVIEW[™] and TruckSim[®] for the investigation of ESC control algorithms. An ESC algorithm was developed and simulated using the co-simulation system described. In addition, track testing of an ESC-equipped heavy tractor and tanker semi-trailer was conducted in conjunction with Heavy Truck Rollover Characterization project participants. Data from the track testing was analyzed to characterize the commercial ESC system equipped on the Volvo test tractor.

The review of the literature on heavy truck ESC systems showed a fairly common basis of approach to stability control strategies. However, various approaches described used different vehicle states for feedback control. Many of these states are not measured on state-of-the-art commercial systems, such as trailer articulation angle and angular rate. While the literature shows how some unmeasured states such as lateral velocity may be estimated, little consideration is given to other states such as articulation and roll angles. Therefore the algorithms presented in the available literature are of limited use in developing an algorithm that is truly representative of a commercial system. Many assumptions must be made about the implementation in the commercial systems in order to simulate their behavior.

The investigation of the HIL simulation system revealed that successful implementation requires information about the ESC system and controller that is not publicly available. In general this implies that successful HIL simulation can only be realized with support from the ESC system supplier. The primary areas of information required are in the requirements of the diagnostic systems within the controller and the CAN-bus protocols used for communication between system sensors and the electronic control unit.

A key finding in the development of an algorithm for ESC system co-simulation is the need for an accurate linear reference model of vehicle dynamics. The standard linear "articulated bicycle" handling model found in the literature was found to be insufficient to accurately reflect the vehicle states during handling maneuvers, even in steady state conditions. Further investigation yielded three factors that contributed significantly to the vehicle handling dynamics: axle roll steer, lateral force compliance steer, and aligning moment compliance steer. Models of each of these effects were developed and successfully incorporated into an enhanced linear handling model. The resulting model was found to track vehicle states of the complete nonlinear TruckSim[®] model accurately both under transient and steady state conditions. A state feedback controller for yaw stability based on this model was developed and implemented. The system was found to qualitatively represent the dynamics of the commercial ESC system. Further simulation, analysis and calibration of the system would be required to quantitatively represent the behavior of the commercial system with the co-simulation system. Support from the commercial ESC system supplier would also likely be required to successfully emulate the behavior of the system.

Analysis of track test data resulted in identification of a number of nuances of the commercial system that are not well understood. In general the system performed as expected in keeping the vehicle under control during roll or yaw instability events. However, the strategy, timing and actuation of individual wheel activations could not be discerned completely from the tests conducted. In addition, the interaction of ESC and ABS systems could not be fully separated from the measured data. In order to fully characterize the system, exhaustive track testing would need to be conducted. It is recommended that should the HIL or co-simulation systems be pursued further, the support of the system supplier Bendix should be sought to provide the missing information needed.

As a continuation of this project, a proposed Phase B entails the investigation of advanced ESC system approaches. Such research would utilize the ESC co-simulation system developed in Phase A of the project to further study advancements in sensors, tractor-trailer communication, and algorithms for ESC. This proposed research would support the efforts of the Heavy Truck Rollover Characterization project, which aims to include the design of a demonstrator truck for technologies that enhance vehicle stability with regard to rollover. The ESC system is an integral part of the stability equation, and enhancements designed in Phase B of this project could ultimately be integrated into the HTRC demonstrator truck.

Chapter 6 - References

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Appendix A

Literature Review

This section provides an overview of the current literature relevant to this project. This fulfills the deliverable task outlined as Task 2 in the project proposal. First, an overview of ESC systems for passenger vehicles and heavy trucks is provided. Then the truck models used for simulation of handling dynamics are investigated.

Electronic Stability Control Algorithms

Electronic stability control is currently implemented in many production passenger vehicles [2, 3]. In passenger vehicles the electronic stability control system is used to prevent spin-out and to match the vehicle yaw rate response to the intent of the driver. The value of peak friction coefficient, μ , at which un-tripped rollover may occur in passenger vehicles is 1.1 to 1.7, whereas this value is in the range 0.4 to 0.8 for heavy and medium trucks [1]. Thus while passenger cars will tend to spin or plow out (limit oversteer or understeer) at the limits of stability, heavy trucks are much more likely to roll over. Rollover accidents tend to be significantly more fatal and costly than other types of accidents, especially in the case of heavy trucks. For this reason electronic stability control in heavy trucks provides the additional safety function of rollover prevention.

ESC for Passenger Vehicles

The fundamental concept of current ESC systems is the use of differential braking to apply a yaw moment to the vehicle in order to ensure the vehicle follows the path indicated by the driver steering input. Actuation is accomplished by the use of hydraulic valves in the braking system that are also used for Anti-lock Braking System (ABS) functionality [4-8]. Sensors used by these systems typically use a steering wheel angle sensor, individual wheel speed sensors, a lateral accelerometer and a yaw rate sensor[8].

It should be noted that ESC affects both vehicle handling stability and responsiveness, and often the design of the system involves a trade-off between the two[9]. One objective of this research is to match the model used for determining driver intent to the actual physical system in order to reduce the compromise in vehicle responsiveness due to the ESC system.

The general form of a typical ESC control scheme is shown in Figure A-1. The current state vector \mathbf{x} of the vehicle is determined from the measurements of the set of sensors described above. Some parameters such as the sideslip cannot be measured directly, and instead must be estimated from the various sensor values. One challenge to estimating sideslip is that the road bank angle causes a bias in the lateral accelerometer measurement. One algorithm for estimating road bank angle and compensating the lateral acceleration measurement is proposed in Tseng [3].

The desired states are typically determined from the measured steering wheel angle either by a linear state space dynamic model or a steady state model of the vehicle. Lookup tables are typically used to vary the model parameters with vehicle speed. The desired states are then compared to the measured and estimated states. Typically, a deadband function is employed to ensure that activation of the system only occurs when there is significant deviation between the desired and the measured state values. Some form of transfer function may then be applied to the error signal to determine the demanded moment to the lower-level system that implements differential braking. For example, in the case of full-state feedback control, this transfer function is simply a set of gains applied to the error signal [10, 11]. The output is generally differential braking pressures applied per axle or individual brake torques applied at each wheel [2, 11].



Figure A- 1. Diagram. Basic ESC High-Level Control Structure.

The most common form of feedback control for ESC is the full-state feedback Linear Quadratic Regulator (LQR) [10, 11]. Such a design automatically places the poles of the closed loop system such that a cost function with weighted Q and R matrices to be applied to the state errors and control outputs respectively is minimized. Alternatively, if only a single variable is used for feedback such as yaw rate, a simple PD controller may be used to place the closed loop poles at a desired location[2].

One approach applied by Anwar [12] is a model-predictive controller for yaw control. Another optimization-based approach is described by Eslamian [13]. This approach uses an optimization to design a non-linear controller for side-slip regulation. Sliding mode control is yet another approach that has been used to address the stability control problem [14-16].

It should be noted that differential braking has been employed on passenger vehicles for functions other than yaw rate and sideslip tracking. Wielenga [17] has proposed an "anti-rollover braking" scheme which uses differential braking to avoid rollover in vehicles with a relatively high center of gravity. In high sideslip conditions, such a vehicle is prone to rollover instead of spinning or plowing out as would a normal passenger car. Braking applied to the front outside wheel in such a condition will slow the vehicle and provide a moment to reduce the vehicle sideslip. It should be noted, however, that while such a system might mitigate the risk of rollover, the vehicle will not necessarily track the direction intended by the driver and may still leave the roadway and result in an accident. However, rollover is a serious concern for heavy trucks, and as will be seen, such a scheme may be used to augment the normal ESC operation in applications for heavy trucks.

ESC for Articulated Heavy Trucks

Increasingly, new heavy trucks are being equipped with electronic driver aids such as ESC systems to augment driver input and ensure vehicle stability in extreme maneuvers. Several approaches similar to the implementation of ESC on passenger vehicles have been proposed for heavy trucks in the literature [18-21].

The handling dynamics of a vehicle with an articulated trailer differ significantly from the dynamics of a single rigid-body vehicle. The following are some of the unique characteristics of the articulated vehicle-trailer combination [22]:

- Smaller stability region in yaw and roll modes at high speed
- Jackknifing and trailer swing phenomena
- Lateral trailer oscillation, possibly self-excited
- Phase lag between vehicle and trailer motions
- Rearward amplification: highest peak lateral acceleration at the rearmost trailer
- Unstable reverse motion

The smaller stability region in yaw and roll modes corresponds to the addition of the trailer dynamics as compared to a single rigid vehicle. The addition of the trailer adds a pole pair to the dynamics with significantly lower damping ratio and natural frequency than that of the vehicle dynamics alone [22].

Similar to the pole pair corresponding to the vehicle yaw dynamics, the trailer pole pair moves closer to the imaginary axis with increasing vehicle speed. Thus instability in the trailer mode is reached at lower speed than the instability in the vehicle mode.

Under large lateral accelerations, heavy trucks are more prone to rollover while passenger cars are more likely to spin or plow out. This is due to the relatively high center of gravity relative to the vehicle track width [1]. The stability control systems for passenger cars are focused on correcting oversteer or understeer conditions. However, the objective of stability control systems

for articulated heavy trucks is often augmented to address other concerns such as jackknifing or rollover.

The operating concept of a commercial tractor-trailer stability control system is shown in Figures 2-4 [23]. A jackknife condition may occur when the tractor unit yaw angle is greater than that desired by the driver, resulting in excessive tractor sideslip and trailer articulation angle. Such a condition may lead to instability and loss of control of the vehicle. As shown in Figure A- 2, commercial stability control systems intervene in the case of jackknifing by braking the outside front wheel of the tractor and by braking the trailer.



Figure A- 2. Illustration. Jackknife (Oversteer) Mitigation [23].

Rollover conditions may occur when the lateral acceleration of the vehicle is high enough to produce sufficient roll to reduce the normal force on the inside tires to zero as shown in Figure A-3. Such a condition is mitigated by the stability control system by braking all wheels of the tractor and trailer if possible. The maximum braking force is used to reduce vehicle forward velocity and thus lateral acceleration as much as possible to prevent rollover [23]. This is shown in Figure A-4.



Figure A- 3. Illustration. Rollover Condition [23].



Figure A- 4. Illustration. Rollover Mitigation [23].

A number of applications of ESC to combination vehicles are found in the literature. Some are applied to passenger cars that may be used to tow a trailer [22, 24, 25]. In this case the controller usually attempts to ensure that the vehicle states track the model of a nominal vehicle without a trailer. Other systems have been designed specifically for articulated tractor-trailer combinations [18-21].

The application of the Bosch vehicle dynamics control system to articulated commercial vehicles was presented in Hecker [18]. This system uses a steady state model based on the articulated bicycle model to determine desired states for yaw rate, sideslip and trailer angle for a given steer input. The trailer articulation angle is then used as a feedback variable in the ESC scheme; however, no mention is made of how this angle is sensed by the system. One significant point made in Hecker [18] is that the model used for desired states must be adjusted for the variable loading conditions unique to heavy trucks. While mention is made that the system identifies the vehicle mass and adapts the model accordingly, no explanation is provided regarding the method of estimation used. Such an adaptation is the primary focus of the current proposed research project.

Another approach to ESC control of an articulated heavy truck is presented in Eisele and Peng [19]. This system adds roll control to the normal oversteer and understeer objectives of the controller. In addition to normal yaw rate and sideslip feedback control, roll control is provided by feedback terms applied to roll angle, roll rate and lateral acceleration. Feedback control of the trailer angle is not considered in this ESC algorithm. Simulations of fishhook maneuvers demonstrate the capability of the system to avoid rollover.

Articulated Truck Models for Handling Simulation

A number of models have been developed for the purposes of simulation of handling maneuvers of articulated trucks, with model fidelity ranging from simple to very complex. The simplest of these is the three degree-of-freedom (3 DOF) linearized articulated bicycle or "tricycle" model originally developed by Jindra [26]. This model is commonly used for eigenvalue analysis, control design and determination of the desired states in the ESC controller. Non-linear formulations of between 4 and 9 DOF may be used to provide more accurate simulations of

actual vehicle performance. In addition, commercial tools such as TruckSim[®] or SIMPACK[®] provide very high fidelity models of the complete combination vehicle non-linear dynamics, and provide extensions for application in real-time Hardware-In-the-Loop (HIL) application.

Analytical Truck Models

A general non-linear 4-DOF model of the articulated truck can be readily derived for the basic lateral dynamics of the vehicle. In Genta [1], such a model is derived using a Lagrange formulation of the equations of motion. The tractor and trailer are treated as rigid bodies, and the four degrees of freedom are:

- Forward motion
- Lateral motion
- Yaw angle
- Trailer articulation angle

A more accurate model of the vehicle dynamics may be realized by adding a roll degree of freedom. There is significant interaction between the roll and yaw dynamics; therefore, the addition of the roll degree of freedom allows higher fidelity simulation of the vehicle lateral handling. Such a model is proposed in Chen and Tomizuka [27]. The model presented is a 5-DOF model where the trailer and tractor share a common roll axis. It should be noted that the common roll axis constraint is not physically realizable in the case where there is an articulation angle between the tractor and trailer. As explained in Gäfvert and Lindgärde [28], in this case either the tractor or trailer must be allowed to rotate in a pitch motion to realize the fifth-wheel hitch constraint. However the common roll angle approximation provides a simple model that reasonably represents the physical system under normal conditions.

A more complete model of the vehicle is realized in Gäfvert and Lindgärde [28] by also considering the pitch and heave motions of the tractor and trailer. A common roll angle between the tractor and trailer is still assumed, therefore the complete system is a 9-DOF model. The large number of terms in the non-linear model equations are managed by assembling the model with the aid of a Maple program. The Maple program handles all of the algebraic manipulations in the derivations of the equations of motion from the constitutive equations, and is capable of directly generating C code of the final equations. In this manner the model C code may be called from a simulation tool such as Simulink for the purpose of simulating vehicle motion.

In Gäfvert and Lindgärde [28], the claim is made that the pitch and heave degrees of freedom are necessary for the simulation of the articulated vehicle for handling maneuvers such as testing ESC algorithms. To this end a comparison is made between simulation output of the 9-DOF model and a linearized 3-DOF model. However, it is unclear that such a 9-DOF model has significantly higher accuracy than a 4 or 5-DOF nonlinear model, as these models are not included in the comparison. Therefore the increased model complexity of the 9-DOF model may not be warranted for the simulation of handling maneuvers.

It should be noted that the fifth wheel connection introduces a significant nonlinearity between the tractor and the trailer. The common roll angle between tractor and trailer assumed in Chen and Tomizuka [27] and Gäfvert and Lindgärde [28] is valid only up to a certain applied moment between the two bodies. At this point the trailer may roll relative to the tractor until the lash in the fifth wheel connection is taken up. This effect is modeled by Law [29] by means of a nonlinear spring with high torsional stiffness before and after the lash moment, and effectively zero stiffness at the point of lash. This fifth wheel/kingpin lashing phenomenon may have a significant effect on the roll dynamics, and may need to be included in the vehicle models to accurately simulate the vehicle motion.

Commercial Truck Models

TruckSim[®] is a commercial heavy truck simulation tool provided by Mechanical Simulation Corporation [30]. The product provides a very high fidelity non-linear model of different truck configurations, including articulated tractor/trailer combinations. Each sprung mass is modeled with 6 degrees of freedom, each solid axle has an additional 2 degrees of freedom and each wheel has a spin degree of freedom. In addition, each solid axle suspension has 6 compliance degrees of freedom. Each tire has 2 dynamic degrees of freedom: one for lagged lateral slip and one for lagged longitudinal slip. Other degrees of freedom are used in the modeling of powertrain and hydraulic systems and for various friction effects. TruckSim RT[®] provides an option for running a TruckSim[®] model on third-party HIL hardware for real-time simulation.

SIMPACK[®] is a high-end multi-body simulation tool provided by INTEC GmbH [31]. The multibody nature of the tool allows virtually any desired level of fidelity of vehicle model. The SIMPACK[®] Automotive add-on provides example multi-body models of standard articulated truck configurations that may be adapted to a specific application. SIMPACK[®] also includes interfaces and specific solver technology that enables it to be used on HIL hardware for real-time simulation.