DEVELOPMENT OF A HARDWARE IN THE LOOP SIMULATION SYSTEM FOR HEAVY TRUCK ESC EVALUATION AND TRAILER PARAMETER AND STATE ESTIMATION

DISSERTATION

Presented in Partial Fulfillment of the Requirements for the Degree Doctor of Philosophy in the Graduate School of the Ohio State University

By

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ABSTRACT

According to NHTSA’s 2011 Traffic Safety Facts, large-truck occupant fatalities increased from 530 in 2010 to 635 in 2011, which is a 20% increase. This was a second consecutive year in which large truck fatalities have increased (9% increase from 2009 to 2010). There was also a 15% increase in large truck occupant injuries from 2010. Moreover, the fatal crashes involving large trucks increased by 1.9%, in contrast to other-vehicle-occupant fatalities that declined by 3.6% from 2010.

Given the high accident involvement rate of heavy trucks, the research presented in this dissertation focused on methods of improving and testing heavy truck ESC performance. The first part of the research, aimed at estimating trailer parameters and states using sensors on the tractor to enhance the capabilities of the tractor based ESC unit. The mass of the vehicle and road grade are first estimated using recursive least square estimation. The trailer CG position is then estimated using the load on the tractor drive axels. This is followed by a planar model of an articulated vehicle to calculate the lateral acceleration, longitudinal acceleration and yaw rate of the trailer CG. Finally a Dual Extended Kalman Filter is developed to estimate trailer roll angle and roll parameters.

The second phase of the research involved the development of a state of the art Hardware in the Loop simulation setup to test heavy truck ESC systems. The design of the HIL system is briefly discussed followed by the modeling of the vehicles in TruckSim. This is followed by a rigorous validation of the vehicle models and the
HIL setup. Finally some of the applications of the validated HIL setup is discussed. This includes an indepth study of the Sine with Dwell maneuver and effects of vehicle speed, surface friction and CG height on the vehicle stability. This is followed by the design of a steering controller which is used to study the advantages afforded by the ESC system in an actual crash scenario.
Dedicated to my Parents, who have loved me unconditionally and ensured that I always had all I needed and more
ACKNOWLEDGMENTS

I would first like to thank my committee Drs. Kahraman, Wang, Heydinger and Guenther. They have been inspirational teachers and contributed immensely toward my education. I would like to especially thank Denny, for being more than just an adviser. I can speak for all his students when I say he goes over and beyond for his students and we are definitely lucky to have him as our adviser.

I would like to thank The National Highway Traffic Safety Administration (NHTSA) for sponsoring my research. The engineers and staff at NHTSA and the Transportation Research Center have been very kind and helpful and made me feel at home. I would like to especially thank Dr. Kamel Salaani for helping me with my dissertation.

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Finally, my friends and roommates over the years who have made this whole experience a time I will always remember fondly. Sriram, Karthik, Vijay, Vivek, Venu,
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1.1 Motivation

According to the 2011 Traffic Safety Facts [1]; large-truck occupant fatalities increased from 530 in 2010 to 635 in 2011, which is a 20% increase. This was a second consecutive year in which large truck fatalities have increased (9% increase from 2009 to 2010). There was also a 15% increase in large truck occupant injuries from 2010. Moreover, the fatal crashes involving large trucks increased by 1.9%, in contrast to other-vehicle-occupant fatalities that declined by 3.6% from 2010.

In 2010, large trucks accounted for 8 percent of the vehicles in fatal crashes and these crashes accounted for 8 percent of all fatalities. But large trucks only accounted for 2 percent of the vehicles involved in injury crashes and 3 percent of the vehicles involved in property-damage-only crashes. This indicates that when a large truck is involved in an accident, very often it leads to fatalities [1].

Of the 3,484 large trucks involved in fatal crashes, 73 percent were combination trucks. The 2010 statistics reveal that large trucks have a fatal accident involvement rate of 1.22 vehicles per 100 million vehicle miles traveled compared to 1.53 for light trucks and 1.18 for passenger cars. This translates to a fatal accident involvement rate of 32.35 vehicles per 100,000 registered large trucks compared to 17.02 for light trucks and 13.09 for passenger cars. These statistics indicate that large trucks account
for a disproportionately large number of fatal crashes compared to any other type of vehicle (excluding motorcycles) even though they account for only a small fraction of registered vehicles.

These statistics indicate that there is potential to save a lot of lives with improvements in heavy truck safety. Improving heavy truck safety for occupants and other road users is a high priority for the National Highway Traffic Safety Administration (NHTSA). In this pursuit, NHTSA conducts extensive tests on heavy trucks to determine if the vehicle is safe and to make enforceable regulations for the manufacturers to follow with an aim to improve overall safety. However, testing tractor trailer combinations on the test track is a cumbersome, time consuming and expensive process, not to mention, the risk of injury to the driver and damage to the test surface thus prohibiting testing at higher speeds. These limitations to actual testing create the need for alternate means to test extreme maneuvers and one solution is to use Hardware in the Loop Simulation (HIL).

Hardware in the Loop (HIL) Simulation refers to the methodology of testing a subsystem where the actual subsystem is hooked up to a simulation environment where the rest of the system is simulated. This methodology is especially useful when it is not possible to model the subsystem in question. Testing of this type eliminates the hazards involved in actual testing and the system response to extreme procedures can be safely tested. HIL testing is hence being increasingly adopted in the industry and in research for developing and fine tuning new technology.

Research conducted by NHTSA indicates that there are significant safety benefits to using ESC systems [2]. In spite of this, estimates say that less than 20% of heavy trucks on the road today are equipped with an ESC system. Cost of incorporating such a system is the main reason for the slow adoption of this technology, especially in tractor trailers where just a tractor ESC improves performance, but an additional
trailer ESC may be desirable to further improve stability. The flip side of this is the increased cost, hence there is considerable advantages to be gained by integrating tractor and trailer ESC systems in terms of cost, technology adoption and consequently road safety. Thus the intent of this research, in addition to developing a HIL system, is to explore methods to estimate trailer parameters and states using sensors on the tractor, in an attempt to improve tractor ESC functionality, reduce costs and consequently increase technology adoption.

1.2 HIL Setup

To fill the need for further and more extreme testing, Transportation Research Center (TRC) and The Ohio State University under contract from NHTSA, has designed and built a HIL setup for tractor and trailer systems with up to five axles shown in Fig.1.1. The HIL rig consists of 10 brake chambers corresponding to the five axles. The brake chambers and hose lengths are sized according to measurements from a 2006 Volvo 6x4 VNL 64T630 tractor and a 53 foot Fruehauf Box trailer. The Electronic Stability Control (ESC) system is connected to the brake chambers just like on the actual truck. The ESC gets its vehicle state inputs from a TruckSim simulation model in real time and actuates the brakes as needed. The brake pressures from each of the chambers are measured and fed back into the TruckSim simulation to complete the loop, as shown in Figure 1.2.
Figure 1.1: HIL Setup

Figure 1.2: HIL Schematic
1.3 Research Objectives

As elaborated in the motivation section, the broad goals of this research are to build and validate a HIL simulation system to test heavy truck ESC systems and to estimate trailer parameters and states. The various steps that were followed are:

1. Build HIL System
   - Complete hardware assembly and wiring
   - Create TruckSim vehicle model and validate it
   - Setup real time interface between hardware and software

2. Showcase the applications of HIL simulation technology
   - Scenario reconstruction using the HIL to highlight the increase in stability afforded by ESC systems.
   - Simulation study of hard to test cases like high center of gravity (CG), low friction and high speed.

3. Trailer parameters and states estimation
   - Trailer mass and road grade estimation
   - Trailer CG estimation
   - Trailer yaw rate and lateral acceleration calculation
   - Trailer roll angle estimation

1.4 Dissertation Outline

The following chapters explain how these objectives are met. Chapter 2 includes a literature review of related research performed in the past which include research
Chapter 3 details the fundamentals of recursive least squared estimation methodology used to estimate vehicle mass and road grade, followed by the trailer CG position. The use of a planar model kinematics to calculate trailer yaw rate and lateral acceleration is then detailed. Finally, a dual Kalman filter estimation is set up to simultaneously estimate trailer roll angle and overall trailer suspension roll stiffness.

Chapter 4 describes the hardware in the loop setup designed to test heavy truck ESC systems. Chapter 5 describes the TruckSim modeling of a 2006 Volvo tractor, a Fruehauf box trailer and a Great Dane flatbed trailer including various loading conditions. Chapter 6 deals with the validation of the models and the HIL simulation setup.

Chapter 7 shows some of the applications of HIL simulation setup including simulation results of hard to test cases like high CG, high speed and low coefficient of friction conditions. This is followed by an actual scenario reconstruction case study implemented using two different path following algorithms.
Chapter References


CHAPTER 2
LITERATURE REVIEW

2.1 Introduction

This chapter includes a literature review of relevant work done in the past. The review is split into the following three categories:

- Hardware in the Loop Systems
- Vehicle Modeling in Trucksim
- Trailer Parameters and States Estimation

2.2 Hardware In The Loop Systems

Hardware in the loop technology is a relatively new methodology to develop and test products. In spite of this, the HIL methodology has gained wide acceptance in a wide variety of fields and especially in automotive applications. In the automotive field, HIL systems have been used to test and develop ABS systems [1], engine control systems [2], [3] and steer-by-wire systems [4] to name a few. HIL technology is also increasingly being adopted in testing conformance to regulations. In Europe, the UNECE Regulation No. 13 [5] allows homologation by test-supported simulation, where for a vehicle that has been physically tested, the compliance of versions or variants of the same vehicle type may be demonstrated by computer simulations.
using validated models. Hahn et al. [6] describe an example of such a homologation process.

There are two different methods for testing ESC functionality using computer simulation. The first one is to integrate the real ESC hardware with the vehicle simulation model in a HIL configuration. The second is to implement the ESC functions within a purely math environment called Software-in-the-loop. HIL simulations are the preferred method of testing and developing ESC systems, since within this environment the electronic unit can be treated as a black box and tested as it is actually applied in the real world. Bendix, which is one of the major ESC systems manufacturer for heavy trucks, uses HIL technology to test and develop their ESC systems. As discussed in [7] and [8], Bendix uses TruckSim to simulate the heavy truck dynamics along with dSPACE real time system to interface the hardware and software.

Clemson University International Center for Automotive Research under sponsorship from U.S.DOT Research and Innovative Technology Administration (RITA) undertook a project to build a HIL system to test heavy truck ESC systems [9],[10]. According to the initial report [9], due to complications in interfacing off-the-shelf ESC systems and limited cooperation from ESC manufacturers, the project was modified to be a software in the loop simulation exercise. The final report [10] shows results and validation of a software in the loop simulation of a heavy truck ESC where mathematical models for brake hardware and pneumatics were used.

The University of Michigan Transportation Research Institute (UMTRI), in a collaboration with NHTSA, has also built a HIL system to test heavy truck ESC effectiveness [11],[12],[13]. The UMTRI system was built to test Wabco ESC system and also used TruckSim to simulate heavy truck dynamics. RT-Lab system was used to interface the hardware and software. Inspite of having a working HIL system,
the UMTRI report [12] declares no data regarding vehicle modeling except for some basic geometric measurements of the vehicle. The report also explicitly states that no validation of the HIL simulation system was performed. The validation work reported in the later report [13] shows very poor correlation between simulation and experimental test for the ESC-off case. No validation by comparison with experimental data was reported for ESC-on case. Though general trends were studied using this HIL system, one to one correlation between the test vehicles and HIL simulations was not achieved by the UMTRI system.

Virginia Tech Transportation Institute (VTTI), in collaboration with NHTSA built a HIL system to test Bendix ESC systems for heavy trucks [14], [15]. This system also used TruckSim to simulate the heavy truck dynamics, but used a LabView real time system to interface the hardware and software. The ESC system used was a modified system supplied by Bendix for easy incorporation into the HIL system. K. Donoughe [14] discusses the validation process of the HIL system in her thesis. She reports very accurate predictions of speed at which wheel lift occurs in ramp steer maneuver, but the transient response of the simulation vehicle does not show very good correlation with the test vehicle. Conversations with subject matter experts also revealed that the VTTI HIL system had latency issues and could not run complex simulations in real time.

The UMTRI and VTTI HIL system designs were considered and their results studied to overcome their limitations. This helped in making design decisions for the current HIL system.
2.3 Vehicle Modeling

The validity of the results obtained from the HIL system depends largely on the accuracy of the vehicle model used. Hence an accurate vehicle model is critical for this study.

The vehicle models developed for use in the HIL simulation are based on models developed by Patrick McNaull [16], [17]. A 2006 Volvo 6x4 VNL tractor, a 28 foot Great Dane flat bed trailer and a 53 foot Fruehauf Box trailer were modeled for use with the HIL. The data used in the modeling have been sourced from various published documents and internal reports cited in this literature review.

The mass and inertia properties used in the tractor model were measured at the U.S. Army TARDEC facility [18]. The tractor steering, suspension bounce and roll tests were conducted by SEA Ltd. [19]. These results were used to model the tractor suspension and steering. The report also contained suspension test results for the Fruehauf box trailer. The suspension test results for the Great Dane trailer were sourced from a paper published by D. Mikesell et al. [20]. The mass and inertia properties of the axle are from a 1991 Volvo 6x4 tractor measured by UMTRI [21]. Since the 1991 Volvo tractor suspension links and geometry are similar to the 2006 Volvo tractor, these parameters are assumed for the current model.

The tire data used in this model was originally collected in a Society of Automotive Engineers (SAE) Cooperative Research Program investigation and used in the National Advanced Driving Simulator (NADS) tire model. This data along with the NADS tire model is discussed in detail by C. Derian [22]. Though the tractor and trailers modeled have different tires than the ones discussed in [22], we are interested in general physics of the system, and not to replicate the exact physics of the modeled tractor trailer. Hence it is acceptable to use a representative tire model.
2.4 Trailer Parameters and States Estimation

The aim of this research is to extend the capabilities of the tractor ESC by estimating trailer parameters and states. To this end, it is assumed that all the tractor states and parameters are known. The parameters that need to be estimated for the trailer are trailer mass and CG position while the states that need to be estimated are trailer yaw rate, lateral acceleration of trailer CG and trailer roll. These parameters and states were chosen since knowledge about these enables the ESC to make intelligent decisions about stability. Various techniques exist to estimate parameters and states, here the appropriate technique needs to be chosen depending on the parameter or state being estimated and the available measurements.

Koto et al. [23] use vehicle acceleration and torque differentials during gear shifts to estimate payload mass. Lingman et al. [24] describe a method using extended Kalman filters to estimate mass and road grade while Huh et al. [25] describe an integrated method using vehicle longitudinal, lateral and vertical dynamics coupled with least squares estimation, Kalman filter and dual least squares estimation respectively to extract the mass of the vehicle. Solmaz et al.[26] describe a Multiple Model Switching and Tuning (MMST) methodology to estimate vehicle CG position for a simple linear vehicle model. They use the standard sensors present on a passenger vehicle, but since the criterion here is to estimate trailer parameters using sensors on the tractor this methodology cannot be used. Allen et al.[27] presented an analysis of the NHTSA Inertia Database and gave regression equations that approximate moments of inertia and CG height given basic vehicle properties including weight, width, length and height. Such data would be very helpful, but this data does not exist for tractors and trailers, moreover for a trailer, these properties change drastically depending on the load it is carrying. Vahidi et al. suggest a recursive linear least squares estimator with multiple forgetting factors in [28] for
simultaneous estimation of road grade and vehicle mass in real time. Since vehicle parameters like mass or center of gravity, do not change rapidly a recursive linear least squares approach would be sufficient to estimate these parameters.

Significant amount of research has been conducted into estimating vehicle states using various measurements and estimation techniques. Chrstos [29] describes a yaw/roll plane model to study race car dynamics and estimate aerodynamic forces on the vehicle. The work presented by Dunn [30] in his PhD dissertation involves a detailed tractor trailer model to estimate jackknife instability using extended Kalman filters. Though these research projects achieve excellent results, the requirement of detailed modeling of the vehicle do not make these approaches ideal for a generalized on board estimation of states and parameters. Ryu, in his PhD dissertation [31] discusses the use of a two antenna GPS system combined with vehicle inertial sensors for online estimation of vehicle parameters and states, using recursive least square estimation and Kalman filters. Here again, this technique requires a dual antenna GPS system which increases cost. There is still a need for a low cost alternative which can provide many of the advantages of more complex systems.

2.5 Conclusion

The literature review performed in this chapter gives an idea about the work performed in the past on HIL simulation and testing of heavy trucks and on techniques used to estimate vehicle states and parameters. The review demonstrated the need for a reliable HIL simulation setup to test heavy truck ESC systems and that very few robust and validated systems exist in the research realm. Though there has been a lot of research done regarding parameter and state estimation of vehicles, the literature review also indicates that there is a need for cost effective estimation of trailer parameters to improve tractor ESC performance.
Chapter References


CHAPTER 3

TRAILER PARAMETERS AND STATES ESTIMATION

3.1 Introduction

This chapter details the techniques used to estimate various vehicle parameters and states. First, the fundamentals of recursive least square estimation are discussed and implemented to estimate vehicle mass, road grade followed by trailer CG location. This is followed by a description of planar model kinematics used to calculate trailer yaw rate and lateral acceleration. Finally a dual Kalman Filter is designed to simultaneously estimate trailer roll and overall trailer roll stiffness. It is assumed that all the tractor states and parameters are known.

3.2 Linear Recursive Least Square Estimation

Least squares is an estimation procedure that was developed independently by Gauss (1795), Legendre (1805) and Adrian (1808) and published in the first decade of the nineteenth century. The idea behind this methodology consists of adjusting the parameters of a model function to best fit a data set. The name "least squares" indicates that the overall solution minimizes the sum of the squares of the residuals. This is a very versatile technique that can be used to estimate parameters of any model which is linear in the parameters. Consider an observation/measurement $y$
that is modeled as a linear function of parameters $\Phi_i$ and inputs $x_i$ as shown in equation 3.2.1

$$y = \Phi_1 x_1 + \Phi_2 x_2 + \ldots + \Phi_p x_p$$  

(3.2.1)

Suppose, we have $n$ measurements of $y$, we can express equation 3.2.1 in matrix form as shown in equation 3.2.2;

$$\vec{Y} = \vec{\Phi} = \begin{bmatrix} y_1 \\ y_2 \\ \vdots \\ y_n \end{bmatrix} = \begin{bmatrix} \Phi_1 \\ \Phi_2 \\ \vdots \\ \Phi_p \end{bmatrix}, \quad X = \begin{bmatrix} x_{11} & x_{12} & \ldots & x_{1p} \\ x_{21} & x_{22} & \ldots & x_{2p} \\ \vdots & \vdots & \ddots \vdots \\ x_{n1} & x_{n2} & \ldots & x_{np} \end{bmatrix}$$

$$\vec{Y} = X\vec{\Phi}$$  

(3.2.2)

Now, the residual vector $\vec{\epsilon}$, defined as the difference between the observations and model predictions, is given by;

$$\vec{\epsilon} = \vec{Y} - X\vec{\Phi}$$

(3.2.3)

The sum of squares of the residuals is given by;

$$S = \vec{\epsilon}(\vec{\Phi}).\vec{\epsilon}(\vec{\Phi})$$

(3.2.4)

The least squares solution $\hat{\Phi}$, i.e. the parameters $\hat{\Phi}$ for which the sum of the squares of the residuals is minimum is given by:

$$\nabla_{\vec{\Phi}}[\vec{\epsilon}(\vec{\Phi}).\vec{\epsilon}(\vec{\Phi})] = 0$$

(3.2.5)

Equivalently

$$\nabla_{\vec{\Phi}}[(\vec{Y} - X\vec{\Phi})^T(\vec{Y} - X\vec{\Phi})] = 0$$

(3.2.6)

$$\nabla_{\vec{\Phi}}[\vec{Y}^T\vec{Y} - \vec{Y}^T X\vec{\Phi} - \vec{\Phi}^T X^T \vec{Y} + \vec{\Phi}^T X^T X \vec{\Phi}] = 0$$

(3.2.7)
Since $\bar{Y}^T X \bar{\Phi} = \bar{\Phi}^T X^T \bar{Y} = (X^T \bar{Y})^T \bar{\Phi}$

Equation 3.2.7 becomes;

$$-2X^T \bar{Y} + 2X^T X \bar{\Phi} = 0$$ (3.2.8)

which is to say

$$\hat{\Phi} = (X^T X)^{-1}X^T \bar{Y}$$ (3.2.9)

Provided $(X^T X)^{-1}$ exists, equation 3.2.9 gives the closed form solution. Now, as the number of observations $n$ increases, it becomes computationally cumbersome to compute the solution, thus for real time estimation of parameters, it would be computationally more efficient if the estimates in equation 3.2.9 are updated recursively as new data is recorded. The recursive form is given by:

$$\hat{\Phi}(k) = \hat{\Phi}(k-1) + L(k)(y(k) - X^T(k)\hat{\Phi}(k-1))$$ (3.2.10)

where

$$L(k) = P(k)X(k) = P(k-1)X(k)(1 + X^T(k)P(k-1)X(k))^{-1}$$ (3.2.11)

and

$$P(k) = (I - L(k)X^T(k))P(k-1)$$ (3.2.12)

$P(k)$ is normally referred to as the covariance matrix. The reader is encouraged to refer to parameter estimation books like [1] for a more detailed derivation. Equation 3.2.10 updates the estimates at each time step based on the error between the model output and the predicted output. The recursive scheme can be considered to be a filter that averages the data to come up with optimal estimates. Averaging is a good strategy if the parameters of the model are constant in nature. However, if the parameters change over time, a better approach would be to weight the observations so that the newer observations have a higher weight than the older observations. Such an approach is described in the following section.
3.3 Recursive Least Square Estimation with Forgetting

If the parameters being estimated change gradually over time, it is beneficial to gradually forget the older data in favor of more recent information. The concept of forgetting, in which older data is gradually discarded, is implemented by gradually reducing the weights of older information. This is done by introducing a weighting matrix $\Lambda$ into the sum of squares of residuals $S$ calculation from equation 3.2.4.

The new sum of squares of residual is given by:

$$S = \bar{c}(\tilde{\Phi}).\Lambda.\bar{c}(\tilde{\Phi}) \quad (3.3.1)$$

Progressing through the derivation of the least square solution as shown in equations 3.2.5 to 3.2.10, we get the closed form, weighted least square solution to be:

$$\hat{\Phi} = (X^T \Lambda X)^{-1} X^T \Lambda \bar{Y} \quad (3.3.2)$$

If $\Lambda$ is chosen to be of the form:

$$\Lambda = \begin{bmatrix}
\lambda^{n-1} & 0 & 0 & \ldots & 0 & 0 \\
0 & \lambda^{n-2} & 0 & \ldots & 0 & 0 \\
0 & 0 & \lambda^{n-3} & \ldots & 0 & 0 \\
\ldots \\
0 & 0 & 0 & \ldots & \lambda & 0 \\
0 & 0 & 0 & \ldots & 0 & 1
\end{bmatrix} \quad (3.3.3)$$

where; $0 < \lambda \leq 1$

Then it is clear that the older information have exponentially diminishing weight. This scheme is called least squares with exponential forgetting and $\lambda$ is called the forgetting factor. The recursive version of this scheme is given by:
\[
\hat{\Phi}(k) = \hat{\Phi}(k-1) + L(k)(y(k) - X^T(k)\hat{\Phi}(k-1))
\]  \hspace{1cm} (3.3.4)

where

\[
L(k) = P(k-1)X(k)(\lambda + X^T(k)P(k-1)X(k))^{-1}
\]  \hspace{1cm} (3.3.5)

and

\[
P(k) = (I - L(k)X^T(k))P(k-1)\frac{1}{\lambda}
\]  \hspace{1cm} (3.3.6)

This scheme is very similar to the Recursive Least Squares (RLS) scheme shown in equations 3.2.10 - 3.2.12. Here a single forgetting factor is applied to all the estimated parameters.

### 3.4 Recursive Least Square Estimation with Multiple Forgetting Factors

In some cases, it may be necessary to apply different forgetting factors to the estimated parameters, since it may be known that some system parameters change faster and hence need to be forgotten faster. For such applications, an RLS scheme with multiple forgetting may be necessary. Vahidi et al. [2] describe in detail the background and derivation of one such recursive scheme, where two parameters are estimated with different forgetting factors. The equations for this scheme are as follows:

\[
\hat{\Phi}(k) = \hat{\Phi}(k-1) + L_{new}(k)(y(k) - X^T(k)\hat{\Phi}(k-1))
\]  \hspace{1cm} (3.4.1)

Where;

\[
L_{new} = \frac{1}{1 + \frac{P_1(k-1)X_1(k)^2}{\lambda_1} + \frac{P_2(k-1)X_2(k)^2}{\lambda_2}} \left[ \begin{array}{c} P_1(k-1)X_1(k) \\ \frac{\lambda_1}{P_2(k-1)X_2(k)} \end{array} \right]
\]  \hspace{1cm} (3.4.2)
\[ P_1(k) = (I - l_1(k)X_1^T(k))P_1(k - 1) \frac{1}{\lambda_1} \]  
(3.4.3)

\[ P_2(k) = (I - l_2(k)X_2^T(k))P_2(k - 1) \frac{1}{\lambda_2} \]  
(3.4.4)

Where;

- \( X_1, X_2 \) - The two inputs to the linear system model
- \( \Phi_1, \Phi_2 \) - The two parameters being estimated
- \( \lambda_1, \lambda_2 \) - The two forgetting factors for the two parameters respectively
- \( P_1, P_2 \) - The two covariances corresponding to the two parameters
- \( l_1, l_2 \) - The two RLS gains which are the two elements of the \( l_{new} \) vector

### 3.5 Straight Drive Parameter Estimation

Certain vehicle parameters are estimated when the vehicle is driving in a straight line. These parameters include vehicle mass, road grade and the trailer CG position. Since total axle load, which is primarily a function of vehicle longitudinal dynamics, is used to estimate these parameters, it makes the estimation simpler and more robust if lateral acceleration effects can be ignored, hence a straight line drive scenario is preferred. There is no loss of utility of the estimation scheme due to this assumption since vehicles typically travel in straight lines for extended periods in daily use scenarios.

The vehicle mass and road grade estimation using RLS with multiple forgetting is described in the section below followed by the trailer CG estimation using RLS with exponential forgetting.

#### 3.5.1 Vehicle Mass and Road Grade Estimation

Vehicle parameters need to be estimated first since they are required for the vehicle states estimation. This section deals with the vehicle longitudinal dynamics where
vehicle longitudinal acceleration is measured and vehicle mass is estimated as a function of acceleration and engine torque, shown in eq.3.5.1 \[2\]

\[
M \ddot{u} = \frac{T_e - J_e \dot{\omega}}{r_g} - F_{fb} - \frac{1}{2} \rho C_d A_f u^2 - Mg(\mu \cos \theta + \sin \theta) \tag{3.5.1}
\]

Where;

\(M\) - Mass of Vehicle (kg)

\(u\) - Vehicle longitudinal velocity \((m/s^2)\)

\(T_e\) - Engine Output Torque \((Nm)\)

\(J_e\) - Powertrain inertia \((kg \ m^2)\)

\(F_{fb}\) - Braking Force \((N)\)

\(\omega\) - Crank shaft angular velocity \((rad/s)\)

\(C_d\) - Drag Coefficient

\(\rho\) - Air density \((kg/m^3)\)

\(A_f\) - Frontal Area \((m^2)\)

\(\mu\) - Coefficient of rolling resistance of road surface

\(\theta\) - Road grade \((rad)\)

Equation 3.5.1 is rearranged as shown in eq. 3.5.2.

\[
\dot{u} = \left(\frac{T_e - J_e \dot{\omega}}{r_g} - F_{fb} - \frac{1}{2} \rho C_d A_f u^2\right) \frac{1}{M} - \frac{g}{\cos(\theta_\mu)} \sin(\theta + \theta_\mu) \tag{3.5.2}
\]

Where;

\(\tan(\theta_\mu) = \mu\)

The equation in this form is used to estimate vehicle mass and road grade using recursive least square estimation with multiple forgetting factors. Equation 3.5.2 can be written in the following linear parametric form,

\[
y = X \Phi, \quad \Phi = [\Phi_1, \Phi_2]^T, \quad X = [X_1, X_2] \tag{3.5.3}
\]
Where
\[ \Phi = [\Phi_1, \Phi_2]^T = \left[ \frac{1}{M}, \sin(\theta + \theta_\mu) \right]^T \]
are the unknown parameters being estimated and
\[ y = \dot{u}, \quad X_1 = \left( \frac{T_e - J_e \dot{\omega}}{r_g} - F_{fb} - \frac{1}{2} \rho C_d A_f u^2 \right), \quad X_2 = -\frac{g}{\cos(\theta_\mu)} \]
can be calculated based on measured signals and known variables. Since the parameters being estimated are mass and road grade, and since it is expected that road grade changes at a higher rate while mass stays relatively constant during operation, it is necessary to use the RLS with multiple forgetting factors discussed in section 3.4.

### 3.5.1.1 Simulation Results for Mass and Road Grade Estimation

The RLS estimator was applied to a TruckSim model of a tractor trailer to estimate the mass and road grade while the model drove in a straight line while accelerating. The forgetting factors \( \lambda_1 \) and \( \lambda_2 \) are used to tune the RLS estimator. Here the values used are \( \lambda_1 = 0.9994 \) and \( \lambda_2 = 0.4 \).

Figure 3.1 shows the mass and road grade estimation plots for the Volvo tractor with the 28’ long Great Dane trailer model. Figure 3.2 shows the mass and road grade estimation for the Volvo tractor with the 53’ long Fruehauf box trailer model. As the plots show, the parameters are estimated very accurately by the RLS estimator with multiple forgetting factors. The spikes in the road grade estimation coincide with the gear shifts in the vehicle. During the gear shifts, the engine speeds spikes instantaneously and this disturbance causes the spikes in the road grade estimation.
Figure 3.1: Volvo Tractor with Great Dane Trailer Mass and Road Grade Estimation

Figure 3.2: Volvo Tractor with Fruehauf Box Trailer Mass and Road Grade Estimation

3.5.2 Trailer Mass and CG Location

Assuming that tractor mass is known, trailer mass can be estimated by direct subtraction (Equation 3.5.4). This assumption does not compromise generality since ESC systems are programmed according to the tractors they are being installed in and the tractor mass does not change considerably throughout its service history. The unsprung mass of the trailer on the other hand, is an input that is required from
the driver.

\[ m_2 = M - m_1 \]  \hspace{1cm} (3.5.4)

\[ m_{2sp} = m_2 - m_{2usp} \]  \hspace{1cm} (3.5.5)

Where;

\( m_1 \) - Mass of tractor (kg)

\( m_2 \) - Mass of trailer (kg)

\( m_{2sp} \) - Sprung mass of trailer (kg)

\( m_{2usp} \) - Unsprung mass of trailer (kg)

The tractors already come equipped with an airbag pressure sensor on the drive axles, which is used to adjust ride height. This airbag pressure is a measure of the load carried by the axle. The load on the drive axles is a function of the tractor and trailer CG location, longitudinal acceleration and road grade as shown in Equation 3.5.6. The effects of pitch angle are neglected since the wheel base of both the tractor and trailer are large and hence weight transfer due to pitch effects is negligible. This also simplifies the equation.

![Figure 3.3: Trailer CG Location Estimation](image)
\[
\begin{align*}
L_2 &= \left[ R_2 - \frac{m_1 g \cdot \cos(\theta)a_1 + (m_1 \alpha_x + m_1 g \cdot \sin(\theta))h_1}{L_1} \right] \\
&= [m_{2sp} g \cdot \cos(\theta)]b_2 - [m_{2sp} g \cdot \sin(\theta) + m_{2sp} \alpha_x]h_2
\end{align*}
\] (3.5.6)

Where

- \( \alpha_x \) - Longitudinal Acceleration (\( m/s^2 \))
- \( R_2 \) - Load on the drive axles of tractor (\( N \))
- \( a_1 \) - Tractor CG longitudinal distance behind the steer axle (\( m \))
- \( L_1 \) - Average wheel base of tractor (\( m \))
- \( L_2 \) - Average wheel base of trailer, from hitch to rear axles (\( m \))
- \( b_2 \) - Trailer CG longitudinal distance in front of the rear axles (\( m \))
- \( h_1 \) - Tractor CG height (\( m \))
- \( h_2 \) - Trailer CG height (\( m \))

Equation 3.5.6 is used to estimate the parameters \( b_2 \) and \( h_2 \) which correspond to the longitudinal position and height of the trailer CG. Figure 3.3 details the various parameters used in Equation 3.5.6. Here again the assumption is that the tractor parameters are known and the trailer length (\( L_2 \)) is provided as an input by the driver. These assumptions do not affect generality of the system, since the ESC is programmed for a particular tractor and the trailer length is readily known to the driver. Recursive least square estimation with single forgetting factor, described in section 3.3, can be used in this case since both the parameters being estimated do not change dynamically during vehicle operation.

Equation 3.5.2 can be written in the following linear parametric form,

\[
y = X\Phi, \quad X = [X_1, X_2], \quad \Phi = [\Phi_1, \Phi_2]^T
\] (3.5.7)
Where

\[ \Phi = [\Phi_1, \Phi_2]^T = [b_2, h_2]^T \]

are the unknown parameters being estimated and

\[
y = \left[ R_2 - \frac{m_1 g \cdot \cos(\theta) a_1 + (m_1 \alpha_x + m_1 g \cdot \sin(\theta)) h_1}{L_1} \right] L_2, \]

\[ X_1 = (m_2 g \cdot \cos(\theta)) \]

\[ X_2 = -(m_2 g \cdot \sin(\theta) + m_2 \alpha_x) \]

These are implemented in the RLS estimation with forgetting equations 3.3.4-3.3.6 to estimate the CG longitudinal position and height of the trailer.

### 3.5.2.1 Simulation Results for Trailer CG Estimation

To estimate the CG longitudinal position and height, the RLS estimation with single forgetting factor is used. Here again the forgetting factor \( \lambda \) is used to tune the filter. Since both these parameters do not change during operation of the vehicle, a value very close to 1 is chosen for \( \lambda \). Here, a value of \( \lambda = 0.995 \) is used.

Figure 3.4 shows the CG position estimation results for the Volvo tractor with the 28’ Great Dane trailer while Figure 3.5 shows the estimation results for the Volvo tractor with the Fruehauf box trailer. The estimator is able to converge on a solution fairly quickly and accurately. It was noted however, that the convergence of the CG height estimate was not as robust as that of the CG longitudinal position estimate. This is because the effect of CG height on axle load is less compared to CG longitudinal position and this phenomenon is exaggerated when the wheelbase is longer. Hence the estimator is not being supplied with enough energy to estimate CG height, making it less robust.
3.6 Tractor Trailer Planar Model for Trailer Yaw Rate and Lateral Acceleration Calculation

Figure 3.6 shows a planar model of a tractor trailer system. The kinematics of this model are used to calculate the yaw rate and lateral acceleration of the trailer CG. To be able to calculate these states, the tractor trailer system needs an additional sensor which senses the trailer articulation angle. Articulation angle is chosen since the sensor for this can be mounted on the tractor and hence the trailer needs no...
modification. Such a setup ensures that the system is independent of the trailer attached.

![Tractor Trailer Planar Model](image)

Figure 3.6: Tractor Trailer Planar Model

The various symbols used in this section are:

- $b_1$ - Tractor CG position in front of the hitch (m)
- $a_2$ - Trailer CG position behind the hitch (m)
- $r_1$ - Tractor yaw rate (deg/s)
- $r_2$ - Trailer yaw rate (deg/s)
- $\psi$ - Articulation angle (deg)
- $u_1$ - Tractor longitudinal velocity (m/s)
- $v_1$ - Lateral velocity of tractor CG (m/s)
- $u_2$ - Trailer longitudinal velocity (m/s)
- $v_2$ - Trailer lateral velocity (m/s)
- $v_p$ - Velocity of the hitch (m/s)
- $v_{CG_2}$ - Velocity of the trailer CG (m/s)
- $a_{x_2}$ - Longitudinal acceleration of trailer (m/s$^2$)
$a_{y_2}$ - Lateral acceleration of trailer CG ($m/s^2$)

The time derivatives of the above listed quantities are denoted by the standard notation, with dots on top of the symbol, where the number of dots represents the order of the derivative.

The figure 3.6 has two reference bases, that are body fixed to the tractor and semitrailer;

$E_i$ - basis fixed to the tractor center of gravity, $CG_1$, $E_1$ always points in direction of travel

e_i - basis fixed to the trailer center of gravity, $CG_2$, $e_1$ is always parallel to semitrailer longitudinal axis

The transformation matrix $Q_\psi$ such that;

$$e_i = Q_\psi E_i$$ (3.6.1)

where:

$$Q_\psi = \begin{bmatrix}
\cos\psi & -\sin\psi & 0 \\
\sin\psi & \cos\psi & 0 \\
0 & 0 & 1
\end{bmatrix}$$

The derivation for the trailer lateral acceleration is shown below; Considering the velocity of point $P$, $v_p$

$$\vec{v}_p = u_1 \hat{E}_1 + v_1 \hat{E}_2 + r_1 \hat{E}_3 \times -b_1 \hat{E}_1$$

$$= u_1 \hat{E}_1 + v_1 \hat{E}_2 - r_1 b_1 \hat{E}_2$$

$$\vec{v}_p = u_1 \hat{E}_1 + (v_1 - r_1 b_1) \hat{E}_2$$ (3.6.2)
Transforming \( \vec{v}_p \) from \( \hat{E}_i \) to \( \hat{e}_i \)

\[
\vec{v}^i_p = Q_\psi \vec{v}^E_p \\
= \begin{bmatrix}
\cos\psi & -\sin\psi & 0 \\
\sin\psi & \cos\psi & 0 \\
0 & 0 & 1
\end{bmatrix}
\begin{bmatrix}
u \\
v_1 - r_1 b_1 \\
0
\end{bmatrix}
\]

\[
= \begin{bmatrix}
u_1 \cos\psi + (r_1 b_1 - v_1) \sin\psi \\
u_1 \sin\psi + (v_1 - r_1 b_1) \cos\psi \\
0
\end{bmatrix}
\]

(3.6.3)

Now, expressing velocity of point P, with respect to CG2,

\[
\vec{v}_p = u_2 \hat{e}_1 + v_2 \hat{e}_2 + r_2 \hat{e}_3 \times a_2 \hat{e}_1 \\
= u_2 \hat{e}_1 + (v_2 + r_2 a_2) \hat{e}_2
\]

(3.6.4)

From equations 3.6.3 and 3.6.4,

\[
u_2 = u_1 \cos\psi + (r_1 b_1 - v_1) \sin\psi
\]

(3.6.5)

\[
v_2 + r_2 a_2 = u_1 \sin\psi + (v_1 - r_1 b_1) \cos\psi
\]

(3.6.6)

The yaw rates of the tractor and trailer are related by the equation;

\[
r_2 = r_1 - \dot{\psi}
\]

(3.6.7)

From equations 3.6.6 and 3.6.7, we get the lateral and longitudinal velocities of the trailer;

\[
v_2 = u_1 \sin\psi + (v_1 - r_1 b_1) \cos\psi - (r_1 - \dot{\psi}) a_2
\]

(3.6.8)

Differentiating equation 3.6.8, we get the expression for \( \dot{v}_2 \);

\[
\dot{v}_2 = \ddot{u}_1 \sin\psi + u_1 \dot{\psi} \cos\psi + (\dot{v}_1 - r_1 b_1) \cos\psi - (v_1 - r_1 b_1) \dot{\psi} \sin\psi + (\ddot{\psi} - \dot{r}_1) a_2
\]

(3.6.9)
Considering the velocity of the trailer CG, $v_{cg}$:

$$\vec{v}_{cg} = u_2 \hat{e}_1 + v_2 \hat{e}_2$$  \hspace{1cm} (3.6.10)

Differentiating the above expression,

$$\dot{\vec{v}}_{cg} = \dot{u}_2 \hat{e}_1 + \dot{v}_2 \hat{e}_2 + u_2 \dot{\hat{e}}_1 + v_2 \dot{\hat{e}}_2$$

$$= \dot{u}_2 \hat{e}_1 + \dot{v}_2 \hat{e}_2 + r_2 \dot{\hat{e}}_3 \times (u_2 \hat{e}_1 + v_2 \hat{e}_2)$$

$$= (\dot{u}_2 - r_2 v_2) \hat{e}_1 + (\dot{v}_2 + r_2 u_2) \hat{e}_2$$  \hspace{1cm} (3.6.11)

From Eq. 3.6.11, we get the expressions for trailer longitudinal and lateral acceleration;

$$a_{x2} = \dot{u}_2 - r_2 v_2$$  \hspace{1cm} (3.6.12)

$$a_{y2} = \dot{v}_2 + r_2 u_2$$  \hspace{1cm} (3.6.13)

Substituting for $\dot{u}_2, \dot{v}_2, u_2, v_2,$ in the above equation using Equations 3.6.5, 3.6.8 and 3.6.9 we get,

$$a_{x2} = \dot{u}_1 \cos\psi - u_1 \dot{\psi} \sin\psi + (\dot{r}_1 b_1 - \dot{v}_1) \sin\psi + (r_1 b_1 - v_1) \dot{\psi} \cos\psi - ...$$

$$+ (r_1 - \dot{\psi}) [u_1 \sin\psi + (v_1 - r_1 b_1) \cos\psi - (r_1 - \dot{\psi}) a_2]$$  \hspace{1cm} (3.6.14)

$$a_{y2} = \dot{u}_1 \sin\psi + u_1 \dot{\psi} \cos\psi + (\dot{v}_1 - \dot{r}_1 b_1) \cos\psi - (v_1 - r_1 b_1) \dot{\psi} \sin\psi + ...$$

$$+ (\dot{\psi} - \dot{r}_1) a_2 + (r_1 - \dot{\psi}) [u_1 \cos\psi + (r_1 b_1 - v_1) \sin\psi]$$  \hspace{1cm} (3.6.15)

Equations 3.6.14 and 3.6.15 give the expressions for the trailer longitudinal acceleration and the lateral acceleration at the trailer CG.

### 3.6.1 Simulation Results for Trailer Lateral Dynamics Calculation

The tractor states including the tractor longitudinal velocity, lateral velocity, longitudinal acceleration, lateral acceleration, and yaw rate are measured. In addition
to these, considering that the articulation angle is also measured, equations 3.6.5, 3.6.7, 3.6.8, 3.6.14 and 3.6.15 can be used to calculate the trailer yaw rate, longitudinal velocity, lateral velocity, longitudinal acceleration and lateral acceleration at the trailer CG. It is to be noted that the equations can be used to find the lateral and longitudinal velocity and acceleration of any point on the trailer by changing the value of $a_2$.

The tractor semitrailer combination is run through a left turn maneuver and the trailer states are calculated using the equations derived above. The results are plotted against the simulation output from TruckSim as shown in Figures 3.7 - 3.11. The plots show very good correlation between the TruckSim output and the values calculated using planar kinematics.

![Figure 3.7: Trailer Yaw Rate Comparison](image)

Figure 3.7: Trailer Yaw Rate Comparison
Figure 3.8: Trailer Longitudinal Velocity Comparison

Figure 3.9: Trailer CG Lateral Velocity Comparison
Figure 3.10: Trailer Longitudinal Acceleration Comparison

Figure 3.11: Trailer CG Lateral Acceleration Comparison
3.7 Roll Angle Estimation

The parameters and states estimated in the previous sections are necessary to estimate the trailer roll plane dynamics. Here again, the axle loads are used to estimate the roll angle of the trailer. Two methods are explored, one where only the drive axle loads are used to estimate trailer roll angle and the second where both the tractor drive axle and the trailer axle loads are used along with a dual Kalman Filter to simultaneously estimate roll angle and the overall roll stiffness of the trailer.

3.7.1 Roll Angle Calculation using Axle Loads

The tractor drive axle corner load is expressed as a function of tractor and trailer roll angles, lateral accelerations and CG positions as shown in Eq.3.7.1.

\[
R_{L2} = \frac{m_{1sp}}{g} \left[ g.a_1 \cos(\theta) + (u_1 + g \sin(\theta)h_1) \frac{g(t_1^2 - (h_1 - h_{1rc}) \frac{\phi_1}{180}) + h_1(\dot{v}_1 + u_1.\phi_1)}{t_1} \right] + \\
\frac{m_{2sp}}{g} \left[ g.b_2 \cos(\theta) - (u_2 + g \sin(\theta)h_2) \frac{g(t_1^2 - (h_2 - h_{2rc}) \frac{\phi_2}{180}) + h_2(\dot{v}_2 + u_2.\phi_2)}{t_1} \right] + \\
\frac{m_{1usp2}}{2} g
\]  

(3.7.1)

Where:

- \(m_{1sp}\) - Tractor sprung mass (kg)
- \(m_{1usp2}\) - Unsprung mass of tractor drive axles (kg)
- \(m_{2sp}\) - Sprung mass of trailer (kg)
- \(R_{L2}\) - Tractor left side drive axle load (N)
- \(t_1\) - Tractor track width (m)
- \(h_1\) - Tractor CG height (m)
- \(h_{1rc}\) - Tractor roll center height (m)
- \(h_2\) - Trailer CG height (m)
- \(h_{2rc}\) - Trailer CG height (m)
\( \phi_1 \) - Tractor roll angle (deg)
\( \phi_2 \) - Trailer roll angle (deg)

Equation 3.7.1 is a linear function of trailer roll angle \( \phi_2 \) and considering that all the other parameters are known, can be used to solve for \( \phi_2 \). The equation also requires knowledge of roll center height for the tractor and trailer. Fancher et al. from UMTRI [3] report extensive data on tractor trailer suspensions. They note that the roll center is generally located on the vehicle center line at a height above the ground where lateral forces are passed from the suspension to the chassis. According to their report [3], roll center heights for air suspension vehicles range from 24in to 29.5in, hence the mean value of 26.75in (0.68m) is chosen as the roll center height, which is also the roll center height for the model. The equation does not however, take into account the relative roll stiffnesses of the tractor and trailer axles, the lash in the trailer hitch or the effects of articulation angle. Though these effects can be modeled into the equation to give a better roll angle calculation, it would make the model suitable only with this particular tractor trailer model and lose generality of application. However these effects introduce sufficient errors in the roll angle calculation such that this method was not suitable for calculating trailer roll.

The relative roll stiffnesses of the axles was found to contribute the most to the error. Hence to eliminate the effects of the relative roll stiffnesses, the trailer axle corner loads along with the tractor axle corner loads were incorporated into Eq.3.7.1. The new equation is shown in Eq. 3.7.2

\[
R_{L1} + R_{L2} + R_{L3} = \frac{(m_{1usp} + m_{2usp})}{2} g + m_{1sp} \left[ g \left( \frac{t_1}{2} - (h_1 - h_{1rc}) \frac{\phi_1 \pi}{180} \right) + h_1 (\dot{v}_1 + u_1.\tau_1) \right] + \\
+ m_{2sp} \left[ g \left( \frac{t_2}{2} - (h_2 - h_{2rc}) \frac{\phi_2 \pi}{180} \right) + h_2 (\dot{v}_2 + u_2.\tau_2) \right] \tag{3.7.2}
\]

Where;
$R_{L1}$ - Tractor steer axle left side load (N)

$R_{L3}$ - Trailer axles left side load (N)

$m_{1_{usp}}$ - Tractor unsprung mass (kg)

$m_{2_{usp}}$ - Trailer unsprung mass (kg)

Eq. 3.7.2 is a linear expression in trailer roll angle $\phi_2$ and can be used to solve for the trailer roll angle. However, it is to be noted that the equation does not take into account the dynamics of the system, instead calculates the roll angle under steady state assumption. This implies that the equation will be inaccurate during highly dynamic maneuvers. To better predict roll angles during dynamic maneuvers, a dual Kalman Filter is designed and implemented to simultaneously filter the measured roll angle and estimate trailer overall roll stiffness. The dual Kalman Filter is discussed in the section below.

### 3.7.2 Dual Kalman Filter

A linear state observer, in short, uses measurements and a state space model of the system to estimate states of the same system, by expressing the unmeasurable states as a linear combination of the measured states, system inputs, and model estimated states. The Kalman Filter, developed by R.E Kalman in 1960 [4], is a type of linear observer that has a recursive scheme and provides unbiased and minimum variance estimates of the states. The Kalman Filter has been used extensively in guidance and control of vehicles in the aerospace and automotive industries. For example, Sidhu [5] discusses the implementation of a Kalman Filter to estimate tire lateral forces and predict vehicle stability and Ryu [6] implements the Kalman Filter for online estimation of vehicle states. The equations of a Kalman Filter for a discrete time linear system (Eq. 3.7.3) are shown in Equations 3.7.5-3.7.11.
Considering a linear state-space model of the form:

\[
x_{k+1} = Ax_k + Bu_k + v_k, \quad (3.7.3)
\]
\[
y_k = Hx_k + \eta_k \quad (3.7.4)
\]

Where \( x_k \) is the state vector, \( y_k \) is the observation vector. \( v_k \) and \( \eta_k \) are independent, zero-mean, Gaussian, state and measurement noise processes with covariance matrices \( Q \) and \( R \), respectively.
Kalman Filter

Initialization: For $k = 0$,

$$\hat{x}_0 = E[x_0],$$  \hspace{1cm} (3.7.5)

$$P_0 = E[(x_0 - E[x_0])(x_0 - E[x_0])^T].$$  \hspace{1cm} (3.7.6)

Computation: For $k \in \{1, 2, \ldots, \infty\}$,

State estimate propagation

$$\hat{x}_k^- = A\hat{x}_{k-1} + Bu_{k-1};$$  \hspace{1cm} (3.7.7)

Error covariance propagation

$$P_k^- = AP_{k-1}A^T + Q;$$  \hspace{1cm} (3.7.8)

Kalman gain matrix

$$K_k = P_k^- H^T[H P_k^- H^T + R]^{-1};$$  \hspace{1cm} (3.7.9)

State estimate update

$$\hat{x}_k = \hat{x}_k^- + K_k(y_k - H\hat{x}_k^-);$$  \hspace{1cm} (3.7.10)

Error covariance update

$$P_k = (I - K_k H)P_k^-.$$  \hspace{1cm} (3.7.11)

Extended Kalman Filter (EKF), as the name suggests, is an extension of the linear Kalman Filter that allows for the dynamics of the system being analyzed to have non-linear behavior. The EKF provides an efficient method for generating approximate maximum-likelihood estimates of the states of a discrete-time nonlinear...
dynamical system. The EKF has been employed in various vehicle dynamics application. Zagorski [7] uses the EKF to estimate the speed of a vehicle roll simulator. Chrstos [8] used the EKF to estimate states of tire lateral slip and body side-slip for a race car. Dunn [9] applied the EKF to an articulated vehicle to estimate hitch articulation and predict the onset of jackknife instability. A detailed introduction to the Kalman Filter and Extended Kalman Filter can be found in Chapter 1 of [10].

The Dual Extended Kalman Filter (DEKF) algorithm is a promising method for simultaneous state estimation and parameter identification given only noisy observations. First described by Nelson [11] this algorithm consists of two separate Extended Kalman Filters, one for state estimation and the other for parameter identification running in parallel. The DEKF has been used in neural network applications and in vehicle dynamics applications. For example Resaeian et al. [12] employed cascaded DEKFs to simultaneously estimate vehicle mass, tire longitudinal and vertical forces and CG position. The schematic of a DEKF is shown in Fig.3.12 [10]. A detailed explanation of the DEKF can be found in Chapter 5 of [10] authored by Wan and Nelson.
The scheme of the DEKF consists of two EKFs as shown in 3.12, the equations for the first EKF, for the state estimation for the nonlinear system 3.7.12 are shown in Equations 3.7.14 - 3.7.19. The equations for the second EKF, for the parameter estimation are detailed in Eqs. 3.7.21 - 3.7.25.

\[
x_{k+1} = F(x_k, u_k, w) + v_k, \tag{3.7.12}
\]

\[
y_k = C(x_k, w) + \eta_k \tag{3.7.13}
\]

Where \(x_k\) is the state vector, \(y_k\) is the observation vector and \(w\) is the parameter vector. \(v_k\) is the state noise vector with covariance \(R^v\) and \(\eta_k\) is the measurement noise vector with covariance \(R^m\).
State Estimation EKF

Initialization: Initializing the Kalman estimates for the state estimator,

\[ \hat{x}_0 = E[x_0], \quad P_{x_0} = E[(x_0 - \hat{x}_0)(x_0 - \hat{x}_0)^T] \] (3.7.14)

Computation: For \( k \in \{1, \ldots, \infty\} \), the time-update equations for the state filter;

\[ \hat{x}_k^- = F(\hat{x}_{k-1}, u_k, \hat{w}_k^-), \] (3.7.15)
\[ P_{x_k^-} = A_{k-1} P_{x_{k-1}} A_{k-1}^T + R^w. \] (3.7.16)

The measurement-update equations for the state filter are;

\[ K_k^z = P_{x_k^-} C^T (CP_{x_k^-} C^T + R^z)^{-1}, \] (3.7.17)
\[ \hat{x}_k = \hat{x}_k^- + K_k^z (y_k - C\hat{x}_k^-), \] (3.7.18)
\[ P_{x_k} = (I - K_k^z C) P_{x_k^-}, \] (3.7.19)

where
\[ A_{k-1} \triangleq \frac{\partial F(x, \hat{w}_k^-)}{\partial x} \bigg|_{\hat{x}_{k-1}}, \quad e_k = (y_k - C\hat{x}_k^-) \] (3.7.20)

Parameter Estimation EKF

Initializing the parameter estimation EKF,

\[ \hat{w}_0 = E[w], \quad P_{w_0} = E[(w - \hat{w}_0)(w - \hat{w}_0)^T], \] (3.7.21)

For \( k \in \{1, \ldots, \infty\} \), the time-update equations for the parameter filter are

\[ \hat{w}_k^- = \hat{w}_{k-1}, \] (3.7.22)
\[ P_{w_k^-} = P_{w_{k-1}} + R_r^w = \lambda^{-1} P_{w_{k-1}} \] (3.7.23)
The measurement update equations for the parameter filter are

\[ K^w_k = P^{-1}_{w_k}(C^w_k)^T[C^w_k P^{-1}_{w_k}(C^w_k)^T + R^e]^{-1}, \]  

(3.7.24)

\[ \hat{w}_k = \hat{w}_k^- + K^w_k e_k \]  

(3.7.25)

\[ P_{w_k} = (I - K^w_k C^w_k) P^{-1}_{w_k} \]  

(3.7.26)

where

\[ C^w_k \triangleq -\frac{\partial e_k}{\partial w} = \frac{C \partial \hat{x}_k^-}{\partial w} \bigg|_{w = \hat{w}_k^-} \]  

(3.7.27)

It is to be noted that \( R^e \) is the measurement noise covariance matrix and \( R^r \) is the process noise covariance matrix for the parameter estimation EKF. If the measurement noise covariance \( R^e \) is a constant diagonal matrix, then it cancels out of the algorithm. The process noise covariance \( R^r \) on the other hand, affects the convergence rate and tracking performance.

Setting

\[ R^r_k = (\lambda^{-1} - 1) P_{w_k}, \quad where \quad \lambda \in (0, 1] \]  

(3.7.28)

Provides for an approximate exponentially decaying weighting on past data. \( \lambda \) is referred to as the "forgetting factor".

### 3.7.3 Trailer Roll Angle and Roll Stiffness Estimation

Using the roll angle measured from Eq.3.7.2, a DEKF is designed to simultaneously estimate trailer roll angle and roll stiffness of the trailer. Since Eq. 3.7.2 does not give a clean measure of the roll angle, the DEKF scheme provides an elegant framework to simultaneously estimate the trailer roll angle and roll stiffness.

The roll dynamics of the trailer are modeled as a linear second order system shown in Eq.3.7.29. Since this is a linear model, a linear Kalman Filter is used for
the state estimation instead of an Extended Kalman Filter. This means that the
Equations 3.7.5 - 3.7.11 are used instead of Equations 3.7.14 - 3.7.19 in the Dual
Extended Kalman Filter routine. Since the model 3.7.29 is a continuous time system,
the Matlab function c2d is used to convert it to discreet time. Constant process and
measurement noise covariance matrices are used to tune the filter.

\[
\begin{bmatrix}
\dot{\phi}_2 \\
\dot{\phi}_2
\end{bmatrix} = \begin{bmatrix}
0 & 1 \\
\frac{K_s}{I_{xx}} & \frac{C_s}{I_{xx}}
\end{bmatrix} \begin{bmatrix}
\phi_2 \\
\dot{\phi}_2
\end{bmatrix} + \begin{bmatrix}
0 \\
\frac{m_{2sp}h_2}{I_{xx}}
\end{bmatrix} [a_{y2}] 
\] (3.7.29)

The parameters of interest are the trailer roll stiffness \( K_s \) and roll damping \( C_s \).
The roll inertia \( I_{xx} \), though important, cannot be estimated reliably under normal
driving scenarios. To get a good estimate of roll inertia, it would require the system
to be excited in a highly dynamic maneuver that produces sustained roll acceleration,
like a slalom test. Since tractor trailers are not subjected to such maneuvers during
everyday use scenarios, the \( I_{xx} \) value is estimated using the trailer mass and CG height
as shown in the Equation 3.7.30. With a reasonable guess for radius of gyration, this
equation provides an estimate of the roll moment of inertia of sufficient accuracy.
This accuracy is sufficient since the parameter of interest, trailer roll stiffness \( K_s \) is
estimated during steady state turning conditions when the inertia does not affect the
estimation.

\[
I_{xx} = m_{2sp}(r_{gy}^2 + (h_2 - h_{rc})^2);
\] (3.7.30)

Where;
\( r_{gy} \) - Radius of gyration of roll inertia about an axis through the CG (m)

Since roll angle is the state that is measured (indirectly, using axle loads), it is
expressed as a function of the parameters being estimated;

\[
\phi_2 = \frac{1}{K_s} \left( m_{2sp}h_2a_{y2} - I_{xx}\ddot{\phi}_2 - C_s\dot{\phi}_2 \right) 
\] (3.7.31)
Equation 3.7.31 is then incorporated into the parameter estimation EKF scheme described in Equations 3.7.21 - 3.7.25 where $C^w_k$ is given by Equation 3.7.32. Roll acceleration is calculated from the estimated roll rate using backward difference formula.

$$C^w_k = \left[ \frac{\partial \phi_2}{\partial K_s} \frac{\partial \phi_2}{\partial C_s} \right]_{w=\hat{w}_k} = \left[ -\frac{m_{2,sp}a_{g_2}h_2 - I_{xx}\dot{\phi}_2 - C_s\dot{\phi}_2}{K_s^2} - \frac{\dot{\phi}_2}{K_s} \right]_{w=\hat{w}_k}$$  

(3.7.32)

The roll parameter estimator needs the system to be exited, i.e. a roll angle to be present, to be able to estimate the roll stiffness. Hence the roll parameter estimator is put inside a conditional if loop, and executed only when the trailer lateral acceleration exceeds a preset threshold (0.02 m/s$^2$). The results of the estimator are discussed in the next section.

3.7.4 Roll Angle and Roll Stiffness Estimation

The simulated vehicle is run through a long left turn maneuver at 25 mph (40.2 kph), simulating the truck taking an exit ramp off a highway. The planar model equations (Section 3.6) are used to calculate the trailer lateral acceleration which is used as an input to the roll estimator DEKF. The trailer roll angle estimate is shown in Figure 3.13 followed by the roll stiff and roll damping estimates in Figure 3.14.
The estimation results for a shorter, left turn maneuver are shown in Figures 3.15 and 3.16.
It is to be noted that the trailer roll angle calculation and estimation are both very sensitive to the accuracy of the CG height and roll center height. With an accurate estimate of the CG height, a good estimate of the roll angle is achieved. Starting from an initial guess equal to the roll stiffness of the drive axles of the tractor, the parameter estimator, during the maneuver, converges to a higher value due to the added stiffness of the trailer roll stiffness. Though this is not extremely accurate (the algorithm needs an even longer maneuver to converge to the actual stiffness), it provides insight into the trailer roll angle gain.
3.8 Conclusion

In this chapter the various techniques used to estimate trailer parameters and states are detailed. The vehicle mass and road grade are estimated, with good accuracy, using RLS with multiple forgetting factors. This is followed by the estimation of trailer CG using the tractor drive axle load. The trailer CG longitudinal position is estimated fairly accurately, but the CG height estimate deteriorates as the length of the trailer increases. A planar model of the tractor trailer is then used to develop kinematic expressions for trailer yaw plane states using articulation angle.

The parameters and states estimated are then used for trailer roll calculation. Trailer roll angle calculation using the drive axle loads was unsuccessful, and hence a more general approach using corner loads of all the axles of the vehicle was chosen. The roll angle measured using the corner loads is then used in a Dual Extended Kalman Filter to simultaneously estimate the roll angle, roll stiffness and roll damping of the trailer. A fairly accurate estimate of the roll angle is achieved using the DEKF estimator.
Chapter References


CHAPTER 4
HARDWARE IN THE LOOP SETUP

4.1 Introduction

Electronic Stability Control systems are an essential and proven tool to improve vehicle safety for the automotive industry. ESC technology is hence becoming mandatory on vehicles world-wide. The US requires ESC systems in all passenger vehicles under 10,000 pounds (4536 kg) for all models produced since 2012 [1]. According to NHTSA, a large number of crashes can be prevented by this technology (single vehicle crashes of passenger cars reduced by 34% and by 59% for SUVs) [1]. ESC systems, now available for heavy trucks and tractor-trailer combinations are expected to further reduce fatal crashes on the road. Hence, testing heavy truck ESC systems to study their effectiveness is strategically important.

Physical testing of heavy vehicles is limited for many reasons that include: the inability to test at highway speed for safety considerations; the lack of specialized testing facilities for heavy vehicles; numerous different tractor and trailer configurations; and the prohibitive expenses associated with field testing. For these reasons, Hardware-in-the-Loop (HIL) testing has an indispensable role in the development and validation of vehicle electronics and software systems. HIL technology is absolutely essential to the continued development of modern vehicle dynamic stability and safety analysis.
4.2 System Description

In this HIL system, the truck dynamics states required by the ESC are generated by software while the brake actuation and brake pressures are produced by the hardware, as it would be in the actual truck. Brake chamber pressures are measured and fed back into the TruckSim truck dynamics, forming a closed loop.

Figure 4.1 shows the HIL system architecture, and the different components of the system. The figure also outlines the general data transfer scheme. The blue lines indicate simulated signals while the green lines indicate signals generated by the hardware. The different systems and their functions are described briefly below. Figure 4.2 shows the actual HIL system hardware.

![HIL Schematic Diagram](image_url)

Figure 4.1: HIL Schematic Diagram
Figure 4.2: HIL Hardware Setup
4.2.1 Host PC

The Host PC is where the software portion of the HIL is housed. In this case, the Host PC has the TruckSim - Simulink cosimulation model which has been integrated with Control Desk. Control Desk is the software provided by dSPACE, which provides the user with complete control over the dSPACE Simulator Hardware and enables the user to change simulation parameters during run time.

4.2.2 dSPACE Simulator and I/O Board

The real-time simulator consists of a quad-processor computer (DS1006 processor board) and is connected to the I/O board through which the software and hardware are connected. Once the simulation is set up on the host PC, it is sent to the real-time simulator from where it is run in conjunction with the hardware. At this stage, the Control Desk software offers the user control of the simulation from the host PC.

4.2.3 ESC System

The ESC system currently used with the HIL system is the EC-60 Advanced ESC controller from Bendix, programmed for the specific 2006 Volvo tractor that has been modeled. The system provides ABS-based stability features, including automatic traction control in addition to yaw control and roll stability programs. This ESC system is identical to the actual ESC module present on the modeled Volvo tractor and is an actual production unit (not modified for use with the HIL system). This assists in the validation process as experimental results can be compared one for one with the HIL simulation results.

The various signals that are simulated/measured that are transmitted between the software, hardware and ESC are listed in detail in Appendix A. Lateral and longitudinal accelerations, yaw rates, hand-wheel steering angle are transmitted
through the high speed CAN (Sensor CAN), other states for engine, powertrain, stop light switch, are transmitted through J1939 CAN bus. Other signals like air bag pressure sensor, wheel speed, brake treadle pressures (primary and secondary) are hardwired to the EC-60.

4.2.4 Brake Hardware

The brake hardware consists of 10 brake chambers corresponding to the five axles, three on the tractor and two on a trailer. The brake chambers and hose lengths are sized according to measurements from the Volvo tractor that was modeled and a 53 foot Fruehauf box trailer.

The pneumatic brake circuit is based on the schematic shown in Figure 4.4. The system is designed to comply with the Federal Motor Vehicle Safety Standards 121 (FMVSS 121) regulations. This includes stopping distance (Section 6.2.1), brake reaction time, reservoir size and push rod displacement.

Figure 4.3: Brake Chamber, Push Rod and Rubber Bushing Assembly

Figure 4.3 shows the brake chamber and push rod assembly of one of the chambers on the HIL. The push rod extends and pushes into a rubber bushing. The rubber bushings were chosen so that they afford similar hysteresis and pressure versus deflection characteristics as seen on real brakes, and conform to FMVSS 121 deflection regulations. The brake circuit in the HIL system does not include the parking brakes.
and the S-cam drum brakes present on the actual tractor and trailer. The brake chamber pressures are measured and sent to the dSPACE real-time system where they are converted to torque values in the software environment.

4.3 Conclusion

This chapter briefly explains the HIL system design, layout and data flow. In this HIL system, in broad terms, the truck dynamics are generated by TruckSim software while the brake actuation and brake pressures are produced by the hardware which are fed back into the truck dynamics forming a closed loop. This is a short overview of the HIL system developed for this research.
Air disc & drum brakes combined on a single axle are shown for pictoral purposes only.

Figure 4.4: HIL Pneumatic Circuit Diagram
Chapter References

CHAPTER 5
TRUCKSIM MODEL DEVELOPMENT

5.1 Introduction

The HIL simulation environment incorporates multiple software environments interacting with commercial hardware. This requires that the software models have sufficient fidelity and accuracy to produce reliable results. This chapter outlines the modeling process to create TruckSim models for use in the HIL simulations to test ESC effectiveness. For this purpose, a 2006 Volvo tractor along with a Fruehauf box trailer and a Great Dane flatbed trailer were modeled. Various loading conditions for the two tractor-trailer combinations were also modeled. The data used for the various parameters are sourced from various reports and measurements. The Tractor model is described first, followed by the models for the two trailers and their loading conditions. The modeling process is described in the paper [1].

Figure 5.1 shows a typical home screen of one of the models used with the HIL. The Test Specifications section at the top left of the screen specifies the vehicle and procedure that is being tested. The Procedure link is empty since the vehicle is controlled through a driver model in the Simulink environment.

The blue link under the Run Control for DSpace specifies the input and output variables for Trucksim. These are discussed in detail later in this chapter. The rest of the links are self explanatory.
The tractor modeled for use with the HIL system is a 2006 Volvo 6x4 VNL64T630 model equipped with a Bendix ESC system (EC-60, 2011). Figure 5.2 below shows the Tractor model page on TruckSim. The various aspects of modeling the tractor in TruckSim are detailed in this section.
The various links correspond to different modeling parameter sets required to completely define a tractor. These links are explained below.

### 5.2.1 Tractor Mass and Inertia Properties

Figure 5.3 shows the mass, center of gravity (CG) and inertia properties of the Volvo Tractor’s sprung mass. The CG and inertia properties were measured at the U.S Army TARDEC facility [2]. The CG and inertia values of the sprung mass are calculated by adjusting for the contribution of the unsprung masses. The mass and inertia properties of unsprung components were measured by The University of Michigan Transportation Institute (UMTRI) [3] from a 1991 Volvo tractor suspension. Since the suspension geometry is similar, there parameters are assumed for the current model.
5.2.2 Steering System

The steering system model page is shown in figure 5.4. The steering ratio was measured to be 20.5. The steering compliance was measured to be 0.000819 deg/Nm in the linear range [4].
5.2.3 Tractor Steer Axle Suspension

The steer axle of the tractor has a solid axle design. Figure 5.5 shows the front suspension model page of the tractor. The mass and inertia properties for this axle are taken from a 1991 Volvo 6x4 tractor measured by UMTRI [3]. Since the 1991 Volvo tractor suspension links and geometry are similar to the 2006 VRTC Volvo tractor, these parameters were assumed for the current model.

Figure 5.6 shows the spring stiffness curve used in the model. This was calculated using the bounce test results from the SEA report[4]. Figure 5.7 shows the auxiliary roll stiffness model. The linear range auxiliary roll stiffness is calculated using equation 5.2.1 and the bounce and roll test data. This equation is derived using the fact that the auxiliary roll stiffness and the suspension springs are in parallel to each other and these two together are in series with the tires. The calculated auxiliary
roll stiffness applies between -5 to 5 degrees of roll, after which the stiffness increases drastically due to the bump-stops. The overall roll stiffness and suspension stiffness values required to calculate the auxiliary roll stiffness are taken from the SEA report [4]. These values are shown in Table 5.1

\[ K_{aux} = \frac{K_{\phi}K_T \frac{T^2}{2} + K_{\phi}K_S \frac{T^2}{2} - K_T K_S \frac{T^2}{4}}{K_T \frac{T^2}{2} - K_{\phi}} \]  (5.2.1)

Where

- \( K_{aux} \) - Auxiliary roll stiffness (Nm/rad)
- \( K_{\phi} \) - Overall roll stiffness (Nm/rad)
- \( K_S \) - Suspension spring stiffness (N/m)
- \( K_T \) - Tire stiffness (N/m)
$T_r$ - Vehicle track width (m)

$T_s$ - Distance between springs (m)

Figure 5.6: Steer Axle Spring Stiffness Model
Figure 5.7: Steer Axle Auxiliary Roll Stiffness

<table>
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<td>$K_t$</td>
<td>1225000 (N/m)</td>
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<td>$K_\phi$</td>
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<td>$K_{aux}$</td>
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</table>

Table 5.1: Parameters for Tractor Steer Axle Auxiliary Roll Stiffness Calculation

5.2.4 Tractor Leading Drive Axle

Figure 5.8 shows the leading drive axle model for the tractor. The inertia properties and roll center data are taken from the UMTRI report [3]. The roll center data corresponds to the leaf spring suspension from the 1991 Volvo tractor, and is used as
an approximation for the 2006 Volvo tractor with air suspension. Figure 5.9 shows the spring stiffness model used for the leading drive axle. This spring model has two curves each for compression and extension corresponding to two different normal load conditions of 5000N and 17000N. These curves were calculated from the bounce test results from the SEA report [4]. Figure 5.10 shows the auxiliary roll stiffness curve calculated using Equation 5.2.1, and the data used to calculate the same is shown in Table 5.2.
Figure 5.9: Leading Drive Axle Spring Force Model

Figure 5.10: Leading Drive Axle Auxiliary Roll Stiffness
Table 5.2: Parameters for Tractor Leading Drive Axle Auxiliary Roll Stiffness Calculation

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<td>$K_\phi$</td>
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<td>$T_s$</td>
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<td>$T_r$</td>
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<tr>
<td>$K_{aux}$</td>
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</tr>
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5.2.5 Tractor Trailing Drive Axle

Figure 5.11 shows the trailing drive axle model for the tractor. The inertia properties and roll center data are taken from the UMTRI report [3]. Figure 5.12 shows the spring stiffness model used for the trailing drive axle. This spring model has two curves each for compression and extension corresponding to two different normal load conditions of 5000N and 17000N. These curves were calculated from the bounce test results from the SEA report [4]. Figure 5.13 shows the auxiliary roll stiffness curve calculated using equation 5.2.1, and the data used to calculate the same is shown in Table 5.3.
Figure 5.11: Trailing Drive Axle Model Page

Figure 5.12: Trailing Drive Axle Spring Force Model

73
Table 5.3: Parameters for Tractor Trailing Drive Axle Auxiliary Roll Stiffness

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</tr>
</thead>
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<td>$T_s$</td>
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<tr>
<td>$K_{aux}$</td>
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</table>

5.2.6 Tires

The tire data used in this model was originally collected in a Society of Automotive Engineers (SAE) Cooperative Research Program investigation and used in the
National Advanced Driving Simulator (NADS) tire model. This data along with
the NADS tire model is discussed in detail by C.Derian [5]. Though the tractor and
trailers modeled have different tires than the ones discussed in [5], we are interested
in the general physics of the system, and not to replicate the exact physics of the
modeled tractor trailer. The tire data used represents real tires still available on the
market and are used on tractor trailers on the road. Hence it is acceptable to use a
representative tire model rather than the model of the exact tires on the test vehicle.

The steer axle is fitted with single tires while both the drive axles are equipped
with dual tires. Figure 5.14 shows the steer axle tire model page. Figure 5.15 shows
the longitudinal and lateral force curves for the steer axle tire model. Figure 5.16
shows the drive axle tire model page. Figure 5.17 show the longitudinal and lateral
force plots for the drive axle dual tires which are used for both the drive axles.

![Figure 5.14: Steer Axle Tire Model Home Page](image)
Figure 5.15: Steer Axle Tire Force Plots

Figure 5.16: Tandem Drive Axle Tire Model Home Page
5.2.7 Brakes

In this HIL system, brake pressures are measured from the hardware and fed into TruckSim. The brake model in TruckSim is simply a look-up table relating brake chamber pressures to brake torque. Since the HIL simulation is used for testing ESC effectiveness, the accuracy of this model is critical.

The Vehicle Research and Test Center (VRTC) and The Ohio State University have conducted extensive tests on pneumatic brakes [6], [7], [8]. However, this data is dated since advances in technology and regulations have made brakes on newer tractors, like the one modeled, more effective. Hence representative data provided by the manufacturer was used for the brake model.

The steer axle of the Volvo tractor has brake chambers with an effective area of 20in$^2$ (129cm$^2$) while the drive axles have chambers with an effective area of 30in$^2$ (193.5cm$^2$). Due to the difference in cross sectional area, these brakes have different torque vs. pressure characteristics. Figure 5.18 shows the brake characteristics used for the front axle whereas the Figure 5.19 shows the model for the drive axle brakes.
Figure 5.18: Steer Axle Brake Model

Figure 5.19: Drive Axle Brake Model
5.3 Trailer Model - Fruehauf Box Trailer

This section details the modeling of a 1992 Fruehauf box trailer. This 53 feet long trailer was chosen for modeling since most of its suspension parameters had already been measured and could be incorporated into the modeling process for a fairly accurate model. Figure 5.20 shows the trailer model home page in TruckSim.

![Trailer with 2 Axles Home Page](image)

Figure 5.20: Fruehauf Box Trailer Model Home Page

5.3.1 Trailer Mass and Inertia Properties

Figure 5.21 shows the mass and inertia properties of the trailer. The mass and dimensions of the trailer were measured at VRTC and the lateral and longitudinal coordinates of the CG were calculated from those measurements. The CG height and inertia values are estimated using the dimensions and mass of the trailer.
5.3.2 Fruehauf Trailer Axles

Figure 5.22 shows the leading and trailing trailer axle model page for the Fruehauf Box trailer. Figure 5.23 shows the spring stiffness curves for both the axles calculated from the bounce test results from the SEA report [4]. Figure 5.24 shows the auxiliary roll stiffness curves for both the axles whose linear range is calculated using Equation 5.2.1. The data used to calculate the auxiliary roll stiffness is shown in Table 5.4.
(a) Leading Axle
(b) Trailing Axle

Figure 5.22: Trailer Axles Model Home Page

(a) Leading Axle
(b) Trailing Axle

Figure 5.23: Trailer Spring Stiffness Curves

(a) Leading Axle
(b) Trailing Axle

Figure 5.24: Trailer Auxiliary Roll Stiffness Curves
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<th>Trailing Axle</th>
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<td>41910.3 (Nm/deg)</td>
</tr>
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</table>

Table 5.4: Parameters for Fruehauf Trailer Axles Auxiliary Roll Stiffness Calculation

5.3.3 Tires

The Fruehauf box trailer uses dual tires on its two axles. The dual tire model used on the drive axles of the tractor, described in Section 5.2.6 are used for the trailer tires as well.

5.3.4 Brakes

The trailer brake chambers on the 1996 Fruehauf box trailer have an effective area of $30in^2$ ($193.5cm^2$). Since these brakes are older, the older data from [7], [6], [8] were used to model these brakes. The chamber pressure versus torque curve used for the trailer brakes is shown in Figure 5.25
5.3.5 Loading

The Fruehauf box trailer was modeled with two different loading conditions:

- High CG
- 60% Gross Axle Weight Rating (GAWR)

5.3.5.1 High CG Loading

Figure 5.27 shows the loading diagram for the High CG loading condition. The inertia calculations for the individual load blocks are detailed in Appendix A. The load is divided into two parts, the front ballast loading and rear ballast loading, for modeling purposes. Tables 5.5 and 5.6 show the CG and inertia calculations for the front and rear ballast loading. The CG is calculated using Equations B.0.1, B.0.2 and B.0.3.
shown in Appendix B. The total inertias of the load are calculated using the parallel axis theorem shown in Equations B.0.5, B.0.6 and B.0.7 shown in Appendix B.

<table>
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<th>Mass (lbs)</th>
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<th>$CG_y$ (in)</th>
<th>$CG_z$ (in)</th>
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<th>$I_{yy}$ ($lb.in^2$)</th>
<th>$I_{zz}$ ($lb.in^2$)</th>
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</table>

| SI Units | 9398.4 (kg) | 818.0 (mm) | 0 (mm) | 1050.0 (mm) | 4715.3 (kg.mm²) | 9980.0 (kg.mm²) | 10,503.5 (kg.mm²) |

- **Table 5.5**: Front Ballast CG and Inertia Calculation

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<th>$CG_y$ (in)</th>
<th>$CG_z$ (in)</th>
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<td>3</td>
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<td><strong>483.5</strong></td>
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<td><strong>19,523,970.6</strong></td>
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| SI Units | 7565.9 (kg) | 12281.9 (mm) | 0 (mm) | 1377.8 (mm) | 2855.5 (kg.mm²) | 4351.6 (kg.mm²) | 5713.4 (kg.mm²) |

- **Table 5.6**: Rear Ballast CG and Inertia Calculation

The CG height calculated in the above tables uses the trailer bed as the datum, hence the height of the trailer bed (48in, 1219mm) is added to this value to get the
total CG height. Figure 5.26 show the TruckSim model pages for the front and rear ballast loading for the High CG case.
Figure 5.26: High CG Loading Model

(a) Front Load

(b) Rear Load
Fruehauf Box Van Trailer
GVWR (24” Tables, Concrete Blocks & OLD Outriggers)
High CG Ballast Condition
Used with Freightliner (VR1) and Volvo (VR2)

Figure 5.27: High CG Loading Diagram
5.3.5.2 60% GAWR Loading

In this loading condition, load blocks are placed on the trailer bed such that the tractor drive axles are loaded to 60% of the rated axle weight. This loading condition is modeled as a single load ballast in TruckSim. Figure 5.28 shows the axle loads for 60% GAWR loading and unloaded conditions for the tractor trailer. Since the actual loading diagram for this case is unavailable, the CG is calculated using the axle loads.

![Figure 5.28: 60% GAWR Loading Axle Weights](image)

From Figure 5.28, it is clear that the load on the trailer is 9390lbs. The load blocks are placed on the bed of the trailer, hence the CG height can be easily determined as the sum of the trailer bed height (48") and half the height of the load blocks used (12"). Hence the load and CG coordinates are found to be:

Load = 4260 kg (9390 lbs)

\[ CG_x = 512 \text{ mm (20.17 in)} \]

\[ CG_y = 0 \text{ mm (0.0 in)} \]

\[ CG_z = 1524 \text{ mm (60.0 in)} \]

These coordinates have the origin on the ground plane at the position of the hitch.
This is the origin used to define loads on the trailer in TruckSim. Figure 5.29 shows the load model in TruckSim.

![Figure 5.29: 60% GAWR Load Model Page](image)

This concludes the modeling of the Fruehauf Box Trailer and its loading conditions.
5.4 Trailer Model - Great Dane Trailer

This section details the modeling of a 28’ Great Dane Trailer. This trailer was chosen for the study since the 28 foot trailer is used as an industry standard for various truck regulations. It is also referred to as the Control Trailer. VRTC also has a vast cache of experimental data for this trailer that is helpful in validating this model. Figure 5.30 shows the trailer model home page in TruckSim.

![Figure 5.30: Control Trailer Model Home Page](image)

5.4.1 Trailer Mass & Inertia Properties

Figure 5.31 shows the mass and inertia properties of the trailer. The mass and dimensions of the trailer were measured at VRTC and the lateral and longitudinal coordinates of the CG were calculated from those measurements. The CG height and inertia values are estimated using the above measurements.
5.4.2 Great Dane Trailer Axle

The Control trailer has a single solid axle suspension at the rear at a distance of 6.8 m from the hitch. Figure 5.32 shows the TruckSim model page for this axle. Figure 5.33 shows the spring stiffness curves used for the axle. Since there was no measured data for the bounce test, the spring stiffness curves were carried over from a previous model developed by Patrick McNaull [9]. Figure 5.34 shows the auxiliary roll stiffness curve. This curve is calculated using the overall roll stiffness value published by David Mikesell et al. [10]. The equation 5.2.1 is used for this calculation and the data used is shown in Table 5.7.
Figure 5.32: Trailer Axle Model Home Page
Figure 5.33: Control Trailer Spring Stiffness Curve
Figure 5.34: Control Trailer Auxiliary Roll Stiffness

Table 5.7: Parameters for Great Dane Trailer Axle Auxiliary Roll Stiffness Calculation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_s$</td>
<td>400000 (N/m)</td>
</tr>
<tr>
<td>$K_t$</td>
<td>1000000 (N/m)</td>
</tr>
<tr>
<td>$K_\phi$</td>
<td>15541.74 (Nm/deg)</td>
</tr>
<tr>
<td>$T_s$</td>
<td>0.965 (m)</td>
</tr>
<tr>
<td>$T_r$</td>
<td>1.197 (m)</td>
</tr>
<tr>
<td>$K_{aux}$</td>
<td>25447.3 (Nm/deg)</td>
</tr>
</tbody>
</table>

5.4.3 Tires

The Great Dane Control trailer uses dual tires on its axle. The dual tire model used on the drive axles of the tractor, described in Section 5.2.6 are used for the trailer tires as well.
5.4.4 Brakes

The brakes on the Great Dane Control trailer are similar to the ones on the Fruehauf Box trailer and hence the same brake model, described in Section 5.3.4 are used for this trailer as well.

5.4.5 Loading

Three loading conditions were modeled for the the Great Dane Control trailer;

- 121 Style Loading
- 60% Gross Axle Weight Rating (GAWR)
- Mid CG Loading

5.4.5.1 121 Style Loading

The 121 style loading is called so since it is used in the Federal Motor Vehicles Safety Standard 121 (FMVSS 121) regulation. Here, load ballasts are placed on the trailer so that the tractor axles are loaded to the vehicle’s Gross Vehicle Weight Rating (GVWR). Since the actual loading diagram is not available, the CG is calculated using the axle loads that were measured prior to testing. These are shown in Fig.5.35.

![Figure 5.35: Control Trailer 121 Loading Diagram](image-url)
From the above figure it is inferred that the load on the trailer is 23113 lbs. The load blocks are placed on the bed of the trailer, hence the CG is determined to be the sum of the trailer bed height (56") and half the height of the load blocks used (12"). Hence the load and CG coordinates are found to be:

Load = 10484 Kg (23113 lbs)

\[ CG_x = 191 \text{ mm (7.5 in)} \]
\[ CG_y = 0 \text{ mm (0 in)} \]
\[ CG_z = 1724 \text{ mm (68 in)} \]

These coordinates have the origin on the ground plane at the position of the hitch. This is the origin used to define loads on the trailer in TruckSim. Figure 5.36 shows the load model in TruckSim.

![Figure 5.36: Control Trailer 121 Loading TruckSim Model](image-url)
5.4.5.2 60% Gross Axle Weight Rating (GAWR)

The 60%GAWR is also called the 58.5% Gross Vehicle Weight Rating (GVWR). The table 5.8 shows the load block positions and CG and inertia calculations. The equations outlined in Appendix B are used for the calculations. The Loading diagram is not available for this load.

<table>
<thead>
<tr>
<th>Ballast</th>
<th>Mass (lbs)</th>
<th>$CG_x$ (in)</th>
<th>$CG_y$ (in)</th>
<th>$CG_z$ (in)</th>
<th>$I_{xx}$ (lb.in²)</th>
<th>$I_{yy}$ (lb.in²)</th>
<th>$I_{zz}$ (lb.in²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4150</td>
<td>-19.5</td>
<td>0</td>
<td>12</td>
<td>1,992,000</td>
<td>490,045</td>
<td>2,083,645</td>
</tr>
<tr>
<td>2</td>
<td>4114</td>
<td>15.5</td>
<td>0</td>
<td>12</td>
<td>1,974,720</td>
<td>485,794</td>
<td>2,065,570</td>
</tr>
<tr>
<td>3</td>
<td>1760</td>
<td>44.5</td>
<td>0</td>
<td>6</td>
<td>844,800</td>
<td>207,826</td>
<td>883,666</td>
</tr>
<tr>
<td>Total</td>
<td>10024</td>
<td>6.1</td>
<td>0</td>
<td>10.9</td>
<td>4,863,755.3</td>
<td>6,914,385.2</td>
<td>10,711,365.9</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>SI Units</th>
<th>Mass (kg)</th>
<th>$CG_x$ (mm)</th>
<th>$CG_y$ (mm)</th>
<th>$CG_z$ (mm)</th>
<th>$I_{xx}$ (kg.mm²)</th>
<th>$I_{yy}$ (kg.mm²)</th>
<th>$I_{zz}$ (kg.mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>4546.8</td>
<td>154.9</td>
<td>0</td>
<td>278.0</td>
<td>1423.3</td>
<td>2023.4</td>
<td>3134.5</td>
</tr>
</tbody>
</table>

Table 5.8: 60%GAWR CG and Inertia Calculation

The CG height calculated in the above tables uses the trailer bed as the datum, hence the height of the trailer bed (48in, 1422mm) is added to this value to get the total CG height. Figure 5.37 shows the TruckSim model page for this loading condition.
5.4.5.3 Mid CG Loading

Figure 5.38 shows the mid CG loading diagram for the control trailer. Table 5.9 shows the CG and inertia calculation for this loading condition. The equations outlined in Appendix B are used for these calculations.
Figure 5.38: Control Trailer Mid CG Loading Diagram
### Table 5.9: Control Trailer Mid-CG Load CG and Inertia Calculation

The CG height calculated in the above tables uses the trailer bed as the datum, hence the height of the trailer bed (48in, 1422mm) is added to this value to get the total CG height. Figure 5.39 shows the TruckSim model for this loading case.
5.5 Conclusion

This chapter details the creation of TruckSim models of a Volvo 6x4 tractor and two trailers; the Fruehauf box trailer and the Great Dane flatbed trailer, with different loading conditions. The model parameters were deduced using experimentally measured data where available and engineering estimations were made for parameters where experimental data was not available. Attention to detail has been given to ensure that the models have sufficient fidelity to be used in HIL simulations to test ESC systems. The validation process of the models developed and the HIL simulations are discussed in the following chapter.

Figure 5.39: Control Trailer Mid CG Loading TruckSim Model
Chapter References


6.1 Introduction

In any model, the accuracy of the simulation relies on the accuracy of the model and the vehicle parameters used to build the model. Hence it is important to check the simulation against experimental results and ensure good correlation exists before the model can be used to extrapolate the behavior of the actual vehicle. In this case, it is all the more critical due to the existence of the hardware component in the simulation environment. The two tractor-trailer models are simulated for various load conditions and the results are compared with experimental data collected by NHTSA’s VRTC at the Transportation Research Center (TRC).

Since the HIL system incorporates the brake hardware, the brake hardware is first validated according to the FMVSS 121 regulations. The validation of the TruckSim models is then done in two steps; the model is first validated using a quasi steady state maneuver, followed by a more severe dynamic maneuver. This is done to ensure that the model is first able to replicate steady state responses of the vehicle. Dynamic validation maneuvers examine the ability of the model to replicate transient responses of the vehicle, like response time and overshoot. This validation process is discussed in detail in the paper [1].
6.2 Volvo Tractor plus Great Dane Control Trailer

This section details the validation process of the Volvo tractor and Great Dane trailer combination. The brake test results are discussed first, to ensure the HIL simulation conforms to the FMVSS 121 standard. The Slowly Increasing Steer (SIS) test results are discussed next, to compare the steady state response of the HIL simulation and finally the Sine with Dwell (SWD) test results are discussed to compare the dynamic response of the HIL simulation.

6.2.1 Brake Test - FMVSS 121

The brake test simulation is done according to the FMVSS 121 regulation [2] and the stopping distance for the various speeds are compared with the regulation. In this test, the tractor is connected to the Control trailer with the 121 style loading. The trailer brakes are disconnected according to the regulation. The vehicle is driven up to speed in a straight line, and full brakes are applied until the vehicle comes to a stop while maintaining a straight heading. The distance covered by the vehicle while coming to a stop is measured.

This test is critical, especially because the HIL system incorporates brake hardware which includes the whole circuit from the air tanks to the brake chambers, but the brake torques are calculated in the software environment as a function of the measured brake chamber pressure. This test helps to validate this complex system and ensure that the brake system on the HIL system is representative of the actual tractor trailer.
Figure 6.1: Stopping Distance Comparison for Various Speeds

Figure 6.1 shows the stopping distance comparison between the HIL simulation and the Federal Motor Vehicle Safety Standards (FMVSS) 121 requirements. The Table 6.1 shows the same comparison. Overall the performance of the HIL simulation meets the requirement and hence is deemed satisfactory.
<table>
<thead>
<tr>
<th>Entry Speed (mph)</th>
<th>FMVSS 121 requirement (feet)</th>
<th>HIL Stopping Distance (feet)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>NA</td>
<td>18.54</td>
</tr>
<tr>
<td>20</td>
<td>32</td>
<td>30.49</td>
</tr>
<tr>
<td>30</td>
<td>65</td>
<td>64.47</td>
</tr>
<tr>
<td>35</td>
<td>89</td>
<td>85.46</td>
</tr>
<tr>
<td>40</td>
<td>114</td>
<td>110.39</td>
</tr>
<tr>
<td>45</td>
<td>144</td>
<td>141.61</td>
</tr>
<tr>
<td>50</td>
<td>176</td>
<td>172.65</td>
</tr>
<tr>
<td>55</td>
<td>212</td>
<td>209.86</td>
</tr>
<tr>
<td>60</td>
<td>250</td>
<td>245.77</td>
</tr>
<tr>
<td>65</td>
<td>NA</td>
<td>286.24</td>
</tr>
<tr>
<td>70</td>
<td>NA</td>
<td>331.60</td>
</tr>
<tr>
<td>75</td>
<td>NA</td>
<td>379.14</td>
</tr>
<tr>
<td>80</td>
<td>NA</td>
<td>428.45</td>
</tr>
</tbody>
</table>

Table 6.1: Stopping Distance Comparison

Figures 6.2 and 6.3 show the tractor speed and acceleration time history for the brake tests at various speeds.
Figure 6.2: Tractor Longitudinal Speed vs. Time

Figure 6.3: Tractor Acceleration vs. Time
6.2.2 Slowly Increasing Steer (SIS)

The slowly increasing steer (SIS) maneuver is used to characterize the vehicle behavior in steady state. This is a constant speed maneuver in which the steering angle is increased from 0 deg to 270 deg at a constant rate of 13.5 deg/s. 30 mph (48 kph) was chosen as the maneuver speed. For this maneuver, the tractor axles are loaded to the vehicle’s GAWR using the 121 style loading, and the tractor ESC is on, while the trailer brakes are disconnected.

The Figures 6.4- 6.7 show the comparison between the HIL simulation and three experimental runs each for the left and right turn SIS maneuvers. In both the HIL simulation and the experimental run, the ESC system gets activated at around the 10 s mark, as can be seen by the drop in longitudinal velocity in Fig. 6.4.

![Figure 6.4: SIS Longitudinal Velocity Comparison](image)

Figure 6.4: SIS Longitudinal Velocity Comparison
Figure 6.5: SIS Lateral Acceleration Comparison

Figure 6.6: SIS Yaw Rate Comparison
It is apparent from the plots that there is very good correlation between the HIL simulation and the experimental runs. The SIS maneuver is also used as a characterization maneuver to find the steering wheel angle that will produce 0.5 g lateral acceleration. This is done by using a linear regression fit on the steering wheel angle versus lateral acceleration data up to 0.3 g. The regression equation is then used to extrapolate the steering wheel angle for 0.5g lateral acceleration. This steering wheel angle is called the 0.5g Steering Wheel Angle (0.5g SWA) and is used as the benchmark for the Sine with Dwell (SWD) maneuver discussed later.

The 0.5 g SWA for the HIL simulation was calculated to be 220 deg, compared to 209 deg for the experimental runs. This is a difference of 5%. This confirms the very good correlation between the simulation and the experimental runs with regards to steady state response.
6.2.3 Sine with Dwell

The Sine with Dwell maneuver developed and used by NHTSA was designed to impart large disturbance to a vehicle in order to study ESC performance. The SWD is used here to validate the dynamic response of the TruckSim models. This is a drop throttle maneuver, where the vehicle is allowed to coast down from a preset initial speed while the steering input is provided. The steering input used is a 0.5 Hz sinusoidal wave with a 1 s pause after the third quarter-cycle. The maneuver entry speed is chosen to be 40 mph. The test is repeated for various steering amplitudes, ranging from 25% of the 0.5g SWA to when instability is noticed, in 8.5% increments. Instability is said to occur when wheel lift is noticed (roll instability) or if the vehicle spins out/jackknifes (yaw instability).

6.2.3.1 Sine with Dwell without ESC

It is important to validate the transient dynamics of the model independently before incorporating it in the HIL simulation environment. For this purpose, the TruckSim model was simulated through the SWD maneuver with out the ESC system.

Figures 6.8 - 6.12 show the comparison between the TruckSim simulation and experimental run at 68% of the 0.5 g SWA. This corresponds to a hand wheel steering angle of 150 deg for the simulation and 143 deg for the test vehicle. The 121 style loading is used to load the tractor axles to the vehicle GVWR. This is a near limit maneuver since both the simulation and test vehicles are unstable in the run of the next higher severity of 76% of 0.5 g SWA, where wheel lift was noticed in both simulation and test runs.
Figure 6.8: SWD Steering Angle Comparison

Figure 6.9: SWD Longitudinal Velocity Comparison
Figure 6.10: SWD Lateral Acceleration Comparison

Figure 6.11: SWD Yaw Rate Comparison
It is apparent from the above plots (Figures 6.8 - 6.12) that even for this near limit maneuver, there is very good correlation between the HIL simulation and the experimental run. These results reiterate the fact that the model predicts the vehicle behavior accurately.

### 6.2.3.2 Sine with Dwell with ESC

Once the transient dynamics of the model were validated, the model was incorporated into the HIL simulation environment with the ESC system. The plots below show the comparison between the HIL simulation and experimental run for the 110% of the 0.5g SWA which corresponds to a steering angle of 233 deg for the test vehicle and 243 deg for the HIL simulation. This is a near limit maneuver since both the simulation and experimental vehicles produce wheel lift in the maneuver of higher severity. The tractor ESC is turned on for these runs, while the trailer brakes are off line.
Figure 6.13: SWD Steering Angle Comparison

Figure 6.14: SWD Longitudinal Velocity Comparison
Figure 6.15: SWD Lateral Acceleration Comparison

Figure 6.16: SWD Yaw Rate Comparison
Figure 6.17: SWD Roll Angle Comparison

Figure 6.18: Tractor Brake Pressure Comparison
The Figures 6.14 - 6.18 illustrate that even for this near limit maneuver, there is very good correlation between the HIL simulation and the experimental run. With the ESC system, the SWD maneuver stability limit has increased from a steering angle of 68% of 0.5 g SWA to 110% of 0.5 g SWA. The results from the other SWD maneuvers are compared in Appendix C. These results reiterate the fact that the model, with the HIL simulation environment, predicts the vehicle behavior accurately and can be used for making further predictions about the behavior of the vehicle.
6.3 Volvo Tractor plus Fruehauf Box Trailer

This section deals with the validation process of the Volvo tractor and Fruehauf Box trailer combination. The Slowly Increasing Steer test results are discussed, to compare the steady state response of the HIL simulation followed by the Ramp Steer maneuver results to validate the dynamic response of the HIL simulation.

6.3.1 Slowly Increasing Steer Maneuver

The SIS maneuver discussed in this section is very similar to the one discussed in Section 6.2.2. In this case the constant speed is chosen to be 25 mph, and the High CG loading is used. The tractor ESC is turned on. Figures 6.19 - 6.22 show the comparison between the HIL simulation and three experimental runs each for left and right side SIS maneuvers.

![Graphs showing comparison between HIL simulation and experimental runs.](image)

Figure 6.19: SIS Longitudinal Velocity Comparison
Figure 6.20: SIS Lateral Acceleration Comparison

Figure 6.21: SIS Yaw Rate Comparison
Figure 6.22: SIS Roll Angle Comparison

From the above plots, it is apparent that there is very good correlation between the HIL simulation and the experimental runs for this maneuver. The SIS results are used to calculate the 0.5g SWA as discussed in Section 6.2.2. The 0.5g SWA calculated from the HIL simulation is 303 deg, which compares to 291 deg from the experimental data. Here again the difference is less than 4% and this reiterates that the model predicts the vehicle behavior accurately.

6.3.2 Ramp Steer Maneuver

The Ramp Steer maneuver (RSM) is used to validate the dynamic response of the model. This maneuver is a drop throttle maneuver, where the vehicle is let to coast down from a predetermined speed while the steering input is provided. The steering angle is increased from 0 to 179 deg at a rate of 110 deg/s and held there for 5 seconds and brought back to 0 at the same rate. The maneuver is repeated for maneuver entry speeds starting from 20 mph in increments of 2 mph till the vehicle
fails the test. Failure is when wheel lift is noticed. The High CG load is used for this maneuver. Tractor ESC is turned on, while the trailer brakes are disconnected.

### 6.3.2.1 Ramp Steer Maneuver without ESC

Here again, it is important to validate the transient dynamics of the TruckSim model before incorporating it in the HIL simulation environment. For this purpose, the TruckSim model is simulated through the RSM maneuver without the ESC system and the data is compared to test runs.

Figures 6.23 - 6.27 show the comparisons for the 24 mph (40 kph) RSM test between the simulation and test.

![Hand Wheel Steering Angle Input](image)

Figure 6.23: RSM Steering Angle Comparison
Figure 6.24: RSM Longitudinal Velocity Comparison

Figure 6.25: RSM Lateral Acceleration Comparison
Figure 6.26: RSM Yaw Rate Comparison

Figure 6.27: RSM Roll Angle Comparison

The plots show that the model generates accurate predictions and that there
is good correlation between the model and the test vehicle. The model is now incorporated into the HIL simulation to further validate the HIL system.

6.3.2.2 Ramp Steer Maneuver with ESC

After the transient response of the Volvo with the Fruehauf trailer was validate, the model was incorporated into the HIL simulation environment to validate the whole system. Again the RSM maneuvers were performed.

The Figures 6.28 - 6.33 show the results for the 36 mph test. This is a limit maneuver since wheel lift was observed in the 38 mph test in the HIL simulation and at 40 mph in the test vehicle.

![Figure 6.28: RSM Steering Angle Comparison](image)

Figure 6.28: RSM Steering Angle Comparison
Figure 6.29: RSM Longitudinal Velocity Comparison

Figure 6.30: RSM Lateral Acceleration Comparison
Figure 6.31: RSM Yaw Rate Comparison

Figure 6.32: RSM Roll Angle Comparison
The plots show that the ESC is activated at the same time in both the HIL simulation and the experimental run, as can be seen in Figure 6.33. It is also apparent that the model generates accurate predictions and can be used to make further predictions about the behavior of the vehicle. The results from the other RSM maneuvers are compared in Appendix D.

### 6.4 Conclusion

This chapter discusses the validation process of the TruckSim model and the HIL simulation. The validation was done in two steps, first, using a quasi steady state...
maneuver (the SIS maneuver) and second, using dynamic maneuvers (the SWD and RSM maneuvers).

Once the models were validated, they were incorporated into the HIL simulation environment which includes the complete brake hardware along with the Bendix ESC system. Very good correlation was achieved even for near limit cases of the dynamic maneuvers for both the tractor-trailer configurations. This gives us confidence that the HIL simulation can be used to perform further, more severe tests which are not possible in the real world.

Thus the models have been validated for both static and dynamic tests with the HIL system. Good correlation between the test runs and the HIL simulations confirm that the HIL simulation system can now be used to extrapolate the behavior of the test vehicles.
Chapter References


CHAPTER 7
HIL APPLICATIONS

7.1 Introduction

After the HIL simulation environment has been tested and validated, various difficult to test conditions were simulated in the HIL environment and studied. This chapter details the various results including high speed maneuvers, low coefficient of friction conditions, high CG loading and scenario regeneration. These details are discussed in the following sections.

7.2 SWD Maneuver Sensitivity and ESC Performance Study

The SWD steering maneuver is becoming a de facto performance testing method in all regulation for both light vehicles and heavy vehicles. Its face validity, objectivity and relative ease make it an attractive test/maneuver for assessing the directional instability mitigation and ESC added safety values.

This section studies the applicability of SWD maneuver to evaluate ESC effectiveness in heavy trucks using the HIL simulation environment. The validated tractor trailer model is put through the SWD maneuver and the sensitivity of the vehicle stability to vehicle speed, CG height and road friction conditions was studied along with the improvements afforded by the ESC system. All the SWD maneuvers
discussed in this section were performed using the Volvo tractor and 28-foot control trailer model with the 121 style loading detailed in Chapter 5.

### 7.2.1 SWD Maneuver Trends with Increasing Speed

This section discusses simulation results of three different brake configurations of the tractor with the control trailer for simulations with maneuver entry speed from 45 to 65 mph. The three brake configurations are:

1. **ESC TRL**: ESC enabled with trailer braking commanded from ESC
2. **ESC**: ESC enabled with no trailer braking
3. **NO**: ESC disabled

The SWD maneuver was conducted on a dry road with a coefficient of friction equal to 0.9 for all three brake configurations. Maneuver severity is increased by increasing steering magnitude. Figure 7.1 shows the steering angle magnitude at different testing severity. The steering angle scaling is obtained from a Slowly Increasing Steer maneuver with a rate of 13.5 deg/sec and vehicle speed at 30 mph. The 100% steering angle, also called the steering scalar (STR), is estimated at a lateral acceleration of 0.5g [Section 6.2.2].

![Figure 7.1: Sine With Dwell Steering Angles as % of Steering Scalar (STR)](image)

The tests are repeated for maneuver entry speeds ranging from 45 mph to 65 mph in 5 mph increments for each brake configuration. The results are listed in Table 133.
7.1. The letters “S” and “U” in Table 7.1 indicate vehicle stability and instability respectively. “S∗” indicates that wheel lift occurred but the vehicle completed the maneuver without rollover. Vehicle instabilities are defined in general terms and no specific algebraic measure is used. Three types of instabilities may occur:

1. **Rollover**: It is indicated when the outrigger on the trailer touches the road surface. Instead of using ratio of lateral load transfer, or axle lifts, a simple approach is applied by adding an outrigger to the trailer model. If the outrigger touches the surface, then a rollover is indicated. This approach is simple to comprehend and use during real-time runs. Moreover, it enacts the actual testing of the vehicle system where outriggers are always added for safety concerns.

2. **Spinning out**: Yaw rate increases exponentially, as a manifestation of an oversteering state. Tractor yaw rate attenuation is used to indicate this unstable state.

3. **Plow out**: The tractor is not able to follow steering commands because of excessive understeer, lateral position measure is used to determine this unstable condition.

The results shown on Table 7.1 confirm the stability benefits gained from using ESC systems. The trailer brakes add more stability margin as speed increases, but no safety benefit is noticed at 45 mph for the steering amplitudes tested. With ESC turned on, the tractor trailer is stable up to maximum steering scalar of 130% for the 45 mph, with or without trailer braking. Through the different speeds, trailer braking marginally increases the safety benefit. This is because the control trailer has a moderately loaded single rear axle, hence its braking capacity is not as aggressive as
it would be when heavily loaded. Moreover, tractors connected to trailers with more axles would benefit more from trailer braking due to the additional braking power.

Safety benefit trends of ESC are consistently maintained through the higher speed runs. The results show that the tractor trailer system equipped with ESC can be steered with a steering scalar substantially greater than the ones with ESC disabled.
<table>
<thead>
<tr>
<th>Steering</th>
<th>45 mph</th>
<th>50 mph</th>
<th>55 mph</th>
<th>60 mph</th>
<th>65 mph</th>
</tr>
</thead>
<tbody>
<tr>
<td>STR Deg</td>
<td>ESC¹</td>
<td>ESC²</td>
<td>ESC NO</td>
<td>ESC TRL</td>
<td>ESC NO</td>
</tr>
<tr>
<td>30% 66.9</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
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<tr>
<td>40% 89.2</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
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<tr>
<td>50% 111.5</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td>60% 133.8</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td>70% 156.1</td>
<td>S</td>
<td>S</td>
<td>U</td>
<td>S</td>
<td>S</td>
</tr>
<tr>
<td>80% 178.4</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>U</td>
</tr>
<tr>
<td>90% 200.7</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>S</td>
<td>U</td>
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<td>130% 289.9</td>
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Table 7.1: SWD Maneuver and ESC Effectiveness Sensitivity to Speed
The SWD at 45 mph with ESC and trailer braking enabled constitutes a baseline maneuver for evaluating ESC stability performance. In contrast to HIL environment, field testing at speeds higher than 45 mph is not practical because of limited field spaces designated for vehicle dynamic testing and safety concerns.

Figure 7.2 shows the steering wheel angle input and the vehicle speed profile during the maneuver. As the steering magnitude increases, vehicle speed decreases accordingly, as a result of a more severe brake intervention from the ESC system.

Figures 7.3 to 7.5 show brake line pressures for all simulations done at steering scalars ranging from 30% to 130%. At steering scalar of 30%, the ESC system did not activate. As the steering scalar increases to 40% and 50%, the ESC was activated just at the end of the one second dwell time. At steering scalar equal or greater than 60%, the ESC system was activated and peak brake pressures occurred at the start and end of the one second dwell time for the steer axle. With positive steering input, the pressure at the left side is greater than the right side. The trailer pressures are the same for left and right, and they just follow a 'pulse' wave with a medium magnitude, around 50% of maximum pressure level. This is because the tractor assumes a trailer with no 'ABS', and generates pressure signals that would not lock up the trailer wheels.

---

1 ESC TRL: ESC enabled with trailer braking

2 ESC: ESC enabled with no trailer braking

3 NO: ESC disabled
Figure 7.2: Steering Angles and Vehicle Speed for 45mph SWD Runs (ESC On with Trailer Braking)

Figure 7.3: Tractor Steer Axle Brake Pressures for 45mph SWD Runs (ESC On with Trailer Braking)
Figures 7.4 to 7.9 show vehicle kinetics that define the tractor trailer state of motion. With the increase of steering scalar, the simulated vehicle yaw rate and lateral acceleration approach the stability limit, and the vehicle roll angle reaches 7
degrees, near the roll limit. HIL simulations showed that it is possible to bring the yaw plane and vertical plane instabilities closer while increasing maneuver severity through steering input augmentation from 30% to 130%. This is an important concept that justifies using yaw plane metrics for categorizing stability performance.

Figure 7.6: Longitudinal Acceleration for 45mph SWD Runs (ESC On with Trailer Braking)
Figure 7.7: Lateral Acceleration for 45mph SWD Runs (ESC On with Trailer Braking)

Figure 7.8: Yaw Rate for 45mph SWD Runs (ESC On with Trailer Braking)
Appendix E includes the plots for the 50 to 65 mph runs. As speed increases, the kinetic energy of the vehicle increases, and requires more time to be attenuated after the end of steering. In all stable cases, up to 65 mph, the yaw rate and lateral acceleration were completely attenuated at about 1.75 sec, and the vehicle maintained stability through ESC activation. Lateral displacements were maintained within the reasonable understeer responsiveness for all cases. The vehicle behavior follows similar trends as in the 45mph case with instability occurring at lower steering scalars due to increased system kinetic energy.

7.2.2 SWD Maneuver Analysis as a Measure of Vehicle Stability

To better understand the planar dynamics of the tractor trailer and to characterize its yaw directional stability state, the tractor yaw rate and lateral acceleration are normalized to maximum values within the dwell time for each run. This procedure provides the ability to quantify the attenuation of lateral dynamics after the end of
the maneuver. A stable vehicle would have decaying yaw rate and lateral acceleration tending to stable null values.

Figure 7.10 shows the normalized tractor lateral acceleration and yaw rate plots for the 45 mph runs (with ESC on and trailer brakes enabled). At a steering scalar equal to 130% the tractor trailer is at the vicinity of rolling over as shown in Figure 7.9. At 1.0 sec after the end of SWD steering input, the yaw rate ratio decreases to less than 30%, and for the following second (two seconds after end of steer) it is attenuated to less than 5%. The same observation can be made for the lateral acceleration ratio; it is under 30% after 1.0 sec after the end of steer, and completely attenuated during the following second. ESC essentially augments vehicle stability by applying and adjusting brake torques individually to induce a correcting yaw moment to the vehicle. Hence a performance measure based on plane dynamics (yaw rate and lateral acceleration) is sufficient to determine ESC’s ability to mitigate imminent instability incidents.

Figure 7.11 shows the tractor and trailer side slip and articulation angle phase diagrams. The ESC system was able to mitigate excessive oversteer, which is recognized by an 'exponential' increase of side slip rate in the phase diagram. This phase diagram shows that for steering scalars equal to 120% and 130%, the vehicle was on the verge of oversteering, which is indicated by the inflection point at about 4 deg of slip angle, and time of 3 sec. As we examine the effects during the dwell time, from 1.5 sec to 2.5 sec the side slip angle rate increases from 5 to 15 deg/sec. Furthermore, during the steering reversal after the dwell period, instead of decaying to zero, the slip angle rate reverses direction from 15 to -15 deg/sec. The articulation angle phase diagram shows that at steering scalar of 130%, the articulation rate was on the onset of 'jackknife' at about the end of the dwell interval.
The benefits gained from the ESC system should not be at the expense of the
vehicle’s responsiveness to the driver’s steering inputs. An extreme example of this could potentially be an ESC system locking the front wheels as the driver tries to avoid an obstacle through sudden and drastic steering input. Locking the front wheels would suppress spinning out and most probably rolling over, however it would prevent the vehicle from responding to the driver’s steering command and intended path of travel. The vehicle would then plow straight ahead and collide with the obstacle. Therefore, a balance between lateral stability and the ability of the vehicle to effectively respond to the driver’s inputs should be maintained. Hence a 'responsive' metric must supplement the lateral stability criteria based on yaw rate attenuation. Several metrics can be used for this purpose which include; lateral acceleration, lateral displacement, lateral velocity, and side slip angle.

The lateral displacement of the tractor with respect to the initial straight path can be calculated using two methods. The first is the FMVSS 126 [1] method, which involves double integrating measured body fixed lateral acceleration at the CG position from the beginning of the steer up to the start of dwell time (1.5 sec) for the sine with dwell applied to tractor-trailer vehicles (FMVSS 126 lists 1.07 sec that corresponds to 0.7 Hz of sine with dwell frequency), Figure 7.12. The second method is to compute the displacement using a ground-fixed coordinate system, Figure 7.13. The planar lateral and longitudinal accelerations were transformed to earth fixed coordinate with the corresponding vehicle yaw angle, and double integrated to get the displacement with respect to a fixed position. Figure 7.14 shows the difference between these two methods. The results indicate that the difference is very small and does not affect the results of lateral displacement measured at 1.5 sec after the beginning of the maneuver. The FMVSS 126 method is simple and provides accurate predictions for the intended purpose of this measure. It has face validity since it refers to the real world relevance of the metric, and can be explained with ease. Moreover,
the method is objective, and can be applied and generalized to vehicles traveling on highways with no inclusion of special cases.

Figure 7.12: Lateral Displacement (FMVSS126 method) for 45mph SWD Runs (ESC On with Trailer Braking)

Figure 7.13: Ground Fixed Lateral Displacement for 45mph SWD Runs (ESC On with Trailer Braking)
Figure 7.14: Lateral Displacement Calculation Difference Between the FMVSS126 and Ground Fixed Methods

7.2.3 SWD Maneuver Sensitivity to Trailer CG Height Variations

The HIL system offers a quick means to simulate repeatable tests, it is an effective tool to study the effectiveness of the ESC when parameters related to weight, or its mechanical properties like CG and inertia change.

HIL simulation runs were done to test the effects of CG height placement on the trailer, and how they would affect dynamics measures for assessing effectiveness of the ESC mitigation intervention. The goal was to define a reasonable tolerance for CG height that would not affect the performance measures. The load vertical positioning is varied as follows;

(a) Payload on deck, and the CG height above ground is 1.72 m, and centered about the fifth wheel or trailer kingpin. [Chapter 5]

(b) Payload raised 2 inches above deck

(c) Payload raised 6 inches above deck
(d) Payload raised 12 inches above deck

(e) Payload raised 18 inches above deck

For each load position, simulations were executed for steering scalar ranging from 30% to 130% or up to loss of stability, at a maneuver entry speed of 45 mph. When ESC was enabled, no major differences were seen for the lateral acceleration and yaw rate, up to a scalar of 120%, Figures 7.15 to 7.17. For 130% steering scalar, lateral acceleration and yaw rate measures show differences attributed to payload placement, Figures 7.18 to 7.20. Despite the steering input normalization through the slowly increasing steer maneuver, for the 130% steering scalar, the placement of the payload noticeably affects the stability measures when the load CG height differs by 6 inches or more. Figures 7.17 and 7.20 show roll angle at 120% and 130% steering scalar respectively. HIL simulations suggest that a tolerance of ±2 inches is admissible for the CG height placement without affecting performance measures at the limit.

Figure 7.15: CG Height Effects on Normalized Tractor Lateral Acceleration and Yaw Rate for 120% Steering Scalar Input
Figure 7.16: CG Height Effects on Lateral Displacement for 120% Steering Scalar Input

Figure 7.17: CG Height Effects on Tractor Trailer Roll Angle for 120% Steering Scalar Input
Figure 7.18: CG Height Effects on Normalized Tractor Lateral Acceleration and Yaw Rate for 130% Steering Scalar Input

Figure 7.19: CG Height Effects on Lateral Displacement for 130% Steering Scalar Input
7.2.4 SWD Maneuver Speed Sensitivity and ESC Effectiveness on Low Coefficient of Friction ($\mu = 0.5$) Surface

Testing heavy vehicles on low $\mu$ surfaces is a cumbersome process due to difficulty in producing consistent surface friction and repeatable tests. The HIL simulation environment overcomes these limitations provides consistent repeatable tests. Table 7.2 shows results of the SWD maneuver on a slippery road surface with a coefficient of friction equal to 0.5. The tractor trailer was simulated for various SWD entry speeds ranging from 45 mph to 70 mph with ESC on (with trailer brakes enabled) and ESC off. The HIL simulations demonstrate a substantial safety benefit from using ESC technology. At this friction level, loss of stability is exhibited through loss of directional control. Rollovers are not possible due to low lateral acceleration levels (limited to about 0.3 g) due to the inability of the tires to produce sufficient lateral forces. The ESC system is able to maintain directional stability at all speeds from 45 mph up to 70 mph, and up to maximum steering input.
Figures 7.21 to 7.25 show the motion kinetics of the vehicle for runs done at 45 mph. Due to the low lateral acceleration values, the roll angles of the tractor and trailer are also low. Plot 7.22 shows that tractor lateral acceleration persists for 3 seconds after the end of steering. Though this is the case on a surface with low coefficient of friction, the yaw rates decay to values below 50% of the maximum within 1 second of the end of steering due to ESC intervention. The phase diagrams shown in Figure 7.24 also demonstrate that the vehicle maintained basic directional stability. The vehicle system was not shown to be driven into spinning or excessive plowing, Figure 7.25. Appendix F shows HIL simulation results for 50 to 70 mph runs.

<table>
<thead>
<tr>
<th>Steering</th>
<th>45 mph</th>
<th>50 mph</th>
<th>55 mph</th>
<th>60 mph</th>
<th>65 mph</th>
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Table 7.2: Low µ SWD Maneuver Sensitivity to Speed and ESC Effectiveness
Figure 7.21: Low $\mu$ SWD Testing, Steering Input and Speed (45 mph, ESC On with Trailer Braking)

Figure 7.22: Low $\mu$ SWD Testing, Tractor Lateral Acceleration and Yaw Rate (45 mph, ESC On with Trailer Braking)
Figure 7.23: Low $\mu$ SWD Testing, Tractor and Trailer Roll Angle (45 mph, ESC On with Trailer Braking)

Figure 7.24: Low $\mu$ SWD Testing, Articulation Angle and Side Slip Phase Diagram (45 mph, ESC On with Trailer Braking)
7.3 Scenario Regeneration

This study was performed to show the capabilities of the HIL simulation environment and to ascertain the speed advantage afforded by ESC systems in an actual crash scenario reported in the Large Truck Crash Causation Survey (LTCCS). A single vehicle loss of control accident scenario was chosen from the LTCCS database (Case Number: 2002-078-004) [2]. An interpretation of the accident scenario was reconstructed in TruckSim considering the initial conditions of the tractor-trailer before loss of control. The tire marks from the accident scene were then used to formulate three possible driver intended paths which led to loss of control.

Since the performance of the vehicle in these scenarios is highly dependent on the driver models used, two different path following algorithms are used. The scenarios were simulated for various entry speeds and the difference in entry speeds for stable runs with and without ESC are reported.
7.3.1 Accident Scenario

According to the LTCCS case files [2],

The crash occurred on a rural interstate that consisted of four lanes divided by a depressed median. There were two lanes on each side of the median. The shoulders were paved with depressed rumble strips. The asphalt roadway was dry, level and straight. The crash occurred in the early morning hours before dawn. There were no sightline restrictions on the roadway deficiencies. The speed limit was 75 mph (121 kph). The roadway accommodated east/west traffic.

This is a crash involving a single tractor truck pulling a flatbed semi-trailer loaded with large steel pipe. Vehicle one was a 2001 Freightliner Conventional FLD 120 CBE tractor with sleeper pulling a 2000 "East" flatbed trailer loaded with eleven (11) sections of steel pipe, each 22-feet long (6.7-meters), 14-inch diameter (35.6-cm) with 1.5 inches thickness (3.8-cm) sidewall. The vehicle was traveling east in the outside lane when the driver apparently went to sleep. The truck began to exit the roadway on the right and came in contact with the depressed rumble strip. There was an immediate left steer input causing the truck to yaw back across the road toward the left side. The vehicle crossed both eastbound travel lanes, exited the roadway on the left, rotated in a counterclockwise direction and the power unit and trailer jackknifed. As the trailer left the roadway the vehicle tripped to the right and began to overturn about the longitudinal axis towards the right. The vehicle was moving northeast in the median. While on the right side the top of the trailer came in contact with the end of a concrete wall. The vehicle then overturned onto the top plane and came to final rest facing northwest in the median. The heavy pipe cargo
had been secured with chain and nylon web straps, but broke loose and scattered throughout the depressed median, over and under the cab of the tractor. The cab and sleeper were completely destroyed. The truck and trailer were upside down at final rest.

The unrestrained driver was entrapped inside the overturned cab under the tractor. He was pronounced dead at the scene from unspecified injuries. The tractor and trailer were towed due to disabling damage.

The path described by the vehicle immediately before roll over is shown in Figure 7.26 [2]. To simulate this case using path following driver models, three intended driver paths were designed using Figure 7.26 as the basis. The three paths are shown in Figure 7.27. Path 1 shown in Figure 7.27 is when the driver tries to stay in the right lane, Path 2 is when the driver over corrects and tries to get back into the right lane while Path 3 is where the driver corrects to get back onto the road and changes lanes in the process.
Figure 7.27: Estimated Driver Intended Paths
7.3.2 Vehicle Model

The vehicle model developed in Chapter 5 was modified for simulating the crash scenario, to reflect the vehicle involved in the crash. The tractor model was retained unchanged since the tractor involved in the crash is similar to the Volvo tractor modeled. The trailer model was modified to reflect the fact that the East trailer is a spread axle trailer. The loading condition was also modified to closely reflect the loading data available in the crash report.

Figure 7.28 shows the load and inertia values used with the model. The trailer load is reported in the case file, and the inertia is calculated considering that the pipes were laid out in two rows of five and six pipes on the bed of the trailer.

![Figure 7.28: Load Model for the Accident Vehicle](image)

7.3.3 Driver Model

To recreate the scenario, a path following algorithm is necessary. Since the performance of the vehicle in the crash scenario is extremely sensitive to the path
following algorithm used, the performance of the vehicle is evaluated using two
different path following algorithms.

Since the algorithm needs to be capable of running in real time along with the
simulation the TruckSim inbuilt path following algorithm was used as one of the
options and another was designed in Simulink applying published research. The path
following algorithms are briefly discussed in the following sections.

7.3.3.1 Simulink Driver Model

For real-time application, the path following algorithm needs to be computationally
inexpensive. Ozguner et al. [3] discuss the design of path following algorithm under
steady state assumptions using a single look ahead point. Jalali et al. [4] build on
this work, modifying the algorithm to work with multiple look ahead points. This
approach is used to design a path following algorithm for the scenario regeneration
purposes. The design is described below.

![Bicycle Model of Tractor for Path Following Algorithm](image)

Figure 7.29: Bicycle Model of Tractor for Path Following Algorithm

A linear bicycle model is used as shown in Figure 7.29 to obtain a linear state-space
representation of the lateral dynamics of the vehicle (Equation 7.3.1)

\[
\begin{bmatrix}
\dot{v}_1 \\
\dot{r}_1 
\end{bmatrix} =
\begin{bmatrix}
-\frac{C_{af} + C_{ar}}{m_1 u_1} & -u_1 + \frac{b_1 C_{ar} - a_1 C_{af}}{m_1 u_1} \\
\frac{b_1 C_{ar} - a_1 C_{af}}{I_z u_1} & -\frac{a_1^2 C_{af} + b_1^2 C_{ar}}{I_z u_1}
\end{bmatrix} \cdot 
\begin{bmatrix}
v_1 \\
r_1
\end{bmatrix} +
\begin{bmatrix}
\frac{C_{af}}{m_1} \\
\frac{a_1 C_{af}}{I_z}
\end{bmatrix} \delta
\]

(7.3.1)

Where;

\(C_{af}\) - Cornering Stiffness of the Steer Axle tires
\(C_{ar}\) - Cornering Stiffness of the Drive Axle tires
\(I_z\) - Yaw inertia of the vehicle
\(V_1\) - CG velocity
\(\delta\) - Road wheel steering angle

Figure 7.30: Steady-state Vehicle Motion along a Circular Path of radius R
Figure 7.30 [4] shows the vehicle moving along a desired circular path of radius $R$, where the distance between the center of gravity and the look-ahead point is defined as look-ahead distance $d$, the distance between the look-ahead offset $o$, and the distance between the look-ahead point and the center of the curve is defined as $h$. By considering the steady-state motion of the vehicle along the curve, where the vehicle perfectly tracks the desired path, Equation 7.3.1 reduces to Equation 7.3.2 by setting $\dot{v}_1 = \dot{r}_1 = 0$.

\[
\begin{bmatrix}
-C_{af} + C_{ar} \\ m_1u_1 \\
-b_1C_{ar} - a_1C_{af} \\ I_zu_1 \\
\end{bmatrix}
\begin{bmatrix}
-u_1 + \frac{b_1C_{ar} - a_1C_{af}}{m_1u_1} \\
-a_1^2C_{af} + b_1^2C_{ar} \\ I_zu_1 \\
\end{bmatrix}
\begin{bmatrix}
v_{1ss} \\
r_{1ss} \\
\end{bmatrix}
= \begin{bmatrix}
\frac{C_{af}}{m_1} \\
\frac{a_1C_{af}}{I_z} \\
\end{bmatrix}\delta_{ss} \tag{7.3.2}
\]

Where the subscript $ss$ denotes the steady state values of the variables. From Equation 7.3.2, the steady state lateral velocity is calculated as;

\[
v_{1ss} = T.r_{1ss} \tag{7.3.3}
\]

where;

\[
T = \left[ b_1 - \frac{a_1.m_1.u_1^2}{C_{ar}.(a_1 + b_1)} \right] \tag{7.3.4}
\]

In general, for steady-state circular motion,

\[
V_{ss} = \sqrt{u_1^2 + v_{1ss}^2} \tag{7.3.5}
\]

\[
V_{ss} = R.r_{1ss} \tag{7.3.6}
\]

From Equations 7.3.2 and 7.3.6, new expressions for $r_{ss}$ and $\delta_{ss}$ can be obtained that are only in terms of the vehicle longitudinal speed $u_1$, the radius of curvature $R$ and vehicle parameters;

\[
r_{ss} = \frac{u_1}{\sqrt{R^2 - T^2}} \tag{7.3.7}
\]

\[
\delta_{ss} = \frac{1}{\sqrt{R^2 - T^2}} \left( a_1 + b_1 - \frac{m_1.u_1^2.(a_1.C_{af} - b_1.C_{ar})}{(a_1 + b_1).C_{af}.C_{ar}} \right) \tag{7.3.8}
\]
In order to calculate an appropriate expression for the steady-state look-ahead offset \(o_{ss}\), where \(o_{ss} = h_{ss} - R\), an expression for \(h_{ss}\) is first defined as:

\[
h_{ss} = \sqrt{d^2 + R^2 - 2R.d\cos\left(\frac{\pi}{2} + \beta\right)} = \sqrt{d^2 + R^2 - 2R.d\frac{V_{1ss}}{V_{1ss}}} \quad (7.3.9)
\]

Using Equations 7.3.3, 7.3.6 and 7.3.9, the expressions for \(h_{ss}\) and \(o_{ss}\) are obtained as:

\[
h_{ss} = \sqrt{d^2 + R^2 + 2dT} \quad (7.3.10)
\]
\[
o_{ss} = \sqrt{d^2 + R^2 + 2dT} - R \quad (7.3.11)
\]

Using Equations 7.3.8 and 7.3.11, the ratio between the desired steering input \(\delta_{ss}\) and the look-ahead offset \(o_{ss}\) is given by:

\[
\frac{\delta_{ss}}{o_{ss}} = \frac{\frac{1}{\sqrt{R^2 - T^2}} \left( a_1 + b_1 - \frac{m_1.a_1^2(a_1C_{af} - b_1.C_{ar})}{(a_1+b_1).C_{af}.C_{ar}} \right)}{\sqrt{d^2 + R^2 + 2dT} - R} \quad (7.3.12)
\]

Now applying Taylor’s series expansion for the denominator, by assuming that \(\frac{|d.(d + 2T)|}{R} << 1\), i.e. \(\sqrt{d^2 + R^2 + 2dT} - R = \frac{d.(d + 2T)}{2R}\)

Equation 7.3.12 can be rewritten as:

\[
\frac{\delta_{ss}}{o_{ss}} = \frac{2 \left( a_1 + b_1 - \frac{m_1.a_1^2(a_1C_{af} - b_1.C_{ar})}{(a_1+b_1).C_{af}.C_{ar}} \right)}{\sqrt{1 - \frac{T^2}{2R^2}.d.(d + 2T)}} \quad (7.3.13)
\]

Next, by assuming that \(\frac{|T|}{R} << 1\) and, thus, \(\sqrt{1 - \frac{T^2}{R^2}} \approx 1\), Equation 7.3.13 can be further simplified as:

\[
\delta_{ss} \approx \frac{2 \left( a_1 + b_1 - \frac{m_1.a_1^2(a_1C_{af} - b_1.C_{ar})}{(a_1+b_1).C_{af}.C_{ar}} \right)}{d.(d + 2T)}.o_{ss} \quad (7.3.14)
\]

Equation 7.3.14 indicates that the steering angle required to keep the vehicle on a circular path, in steady state motion is a function of look-ahead offset \(o_{ss}\), the
vehicle longitudinal velocity $u$, the look-ahead distance $d$ and vehicle parameters. The equation is independent of the radius of curvature $R$, which makes suitable for use with every possible road profile. Moreover, since Equation 7.3.14 is a function of longitudinal velocity, it updates itself with change in velocity. The stability of the steering controller has been proven analytically using the Routh-Hurwitz technique by Ozguner et al. [3].

Jalali et al. [4] point out that using a single preview point for describing a driver model is unrealistic and unsatisfactory. They argue that if the look-ahead point is too far in front of the vehicle, it will be inappropriate to act on the preview information at the time of its acquisition, and the information has been lost by the time it is useful. On the other hand, if the look-ahead point is too close to the vehicle, it necessarily causes very poor control, especially at higher speeds. Moreover, a single preview point is not conducive for complex road profiles. Hence Jalai et al. propose a method using multiple look-ahead points defined as a function of vehicle longitudinal velocity (Equation 7.3.15).

$$d_{\text{look-ahead}_i}(t) = W_i(d_{\text{const}} + t_{\text{driver}}.u_1(t)) \quad (7.3.15)$$

Where, $d_{\text{look-ahead}_i}$ is the $i^{th}$ look-ahead point, $d_{\text{const}}$ is a constant distance that the driver will look ahead, even at lower velocities, $t_{\text{driver}}$ is the reaction time of the driver and $W_i$ is the relative distance between the $i^{th}$ preview point and the vehicle.
Figure 7.31: Vehicle Look-Ahead Points and Lateral Offsets

Figure 7.31 shows the look-ahead points and the lateral offsets corresponding to each of those points. The new look-ahead offset is defined as the weighted sum of all the lateral offsets:

\[ o(t) = \sum_{i=1}^{5} (G_i e_i(t)) \quad (7.3.16) \]

Where \( e_i \) is the lateral offset for the the \( i^{th} \) preview point and \( G_i \) is the lateral offset gain for the \( i^{th} \) preview point. The lateral offset gains are decided in an ad hoc fashion based on intuition rather than any formal optimization scheme. Using the new lateral offset and look-ahead distance, Equation 7.3.14 becomes;

\[ \delta_{ss} \approx \frac{2 \left( a_1 + b_1 - \frac{m_1 a_f^2 (a_1 C_{af} - b_1 C_{ar})}{(a_1 + b_1) C_{af} C_{ar}} \right)}{d_{\text{look-ahead}}(t) \left( d_{\text{look-ahead}}(t) + 2T \right)} \cdot \sum_{i=1}^{5} (G_i e_i(t)) \quad (7.3.17) \]

Equation 7.3.17 describes the driver model equation as used by Jalali et al. [4]. This steady state formulation for steering angle, when implemented to control the vehicle during high speed, dynamic maneuvers, caused sharp and unrealistic steering angle changes, leading the vehicle to instability. To improve the performance of the steering controller for high speed dynamic maneuvers, a feed forward term was introduced to smooth the controller response.
Figure 7.32 shows a schematic of the final steering controller model. The steering rate multiplied by a gain is added to the steering angle which effectively works like a feed forward term. The feed forward gain is determined empirically. This driver model is used in the scenario regeneration study. The validation of the steering controller is discussed in the following section.

![Simulink Driver Model with Feed Forward](image)

**Figure 7.32: Simulink Driver Model with Feed Forward**

### 7.3.3.2 Validation of Simulink Driver Model

The tractor trailer model is put through a circle drive maneuver to validate the path following driver model. A circular path of radius 68 m is used and the vehicle is set to a constant speed of 45 mph (72.4 kph). Figures 7.33 to 7.35 show the validation of the driver model. It is clear from Figure 7.33 that the driver model is able to track the desired path accurately.
Figure 7.33: Comparison of Design Path and Path Followed by Vehicle

Figure 7.34: Driver Model Steering Input
7.3.3.3 TruckSim Driver Model

The TruckSim inbuilt driver model page is shown in Figure 7.36. The driver preview time is set to be constant at 1.5 seconds since the TruckSim help file for the driver model specify this value to be a realistic number. No further technical explanation about the driver model is given. The three driver intended paths discussed earlier are modeled as the target paths to be followed.
7.3.4 Scenario Regeneration Results

The three intended driver paths were simulated using the two different path following algorithms with and without ESC in the HIL simulation environment. The vehicle speed is increased (up to 75 mph) until the vehicle loses stability during the maneuver. A run is deemed stable if the vehicle completes the maneuver and lateral acceleration decays to less than 0.1 g within 10 seconds of the start of the maneuver. The maximum speed at which the vehicle is able to complete the maneuver without losing stability is recorded. The results are shown in Table 7.3.

<table>
<thead>
<tr>
<th>Driver Model</th>
<th>Path1 ESC Off</th>
<th>Path1 ESC On</th>
<th>Path2 ESC Off</th>
<th>Path2 ESC On</th>
<th>Path3 ESC Off</th>
<th>Path3 ESC On</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulink</td>
<td>66</td>
<td>66</td>
<td>55</td>
<td>63</td>
<td>63</td>
<td>63</td>
</tr>
<tr>
<td>TruckSim</td>
<td>62</td>
<td>64</td>
<td>50</td>
<td>56</td>
<td>50</td>
<td>58</td>
</tr>
</tbody>
</table>

Table 7.3: Comparison of Accident Scenario Stable Speeds With and Without ESC
It is important to note that the parameters of both the driver models were kept the same for all the paths tested. It is possible to optimize both the driver models for each individual path, but since this study is focused on studying the effects of ESC systems during severe maneuvers, optimizing the driver model was not a priority. It is clear from the table that ESC system offers an advantage during this severe high speed scenario. The advantage varies depending on the driver model used and the intended driver path.

Figure 7.37 shows the comparison between the two driver models for Path 1. The TruckSim driver driven vehicle enters the maneuver at 64 mph while the Simulink driver driven model enters the maneuver at 66 mph. In spite of the higher entry speed, the Simulink driver driven model has a higher exit speed. Figure 7.38 shows the comparison for Path 2 at 63 mph for both the driver models. For this path, since the speed difference between stable runs for the two driver models was high, the TruckSim driver model was run at the higher speed and this data was plotted for easy comparison between the driver models. Figure 7.39 shows the comparison for Path 3.
Figure 7.37: Simulink and TruckSim Driver Models Performance Comparison for Path1
Figure 7.38: Simulink and TruckSim Driver Models Performance Comparison for Path2
Figure 7.39: Simulink and TruckSim Driver Models Performance Comparison for Path3
7.4 Conclusions

This chapter details the various studies that have been conducted using the validated HIL system. These include studying the effects of speed, surface friction, and CG height on the SWD maneuver results. Further, a path control algorithm was designed and used to study the advantage afforded by an ESC system in an actual crash scenario.
Chapter References


CHAPTER 8
CONCLUSIONS

8.1 Contributions to the Engineering Community

A suite of general purpose analytical tools and computer models have been developed to facilitate the detailed study of heavy truck ESC systems implemented on a state-of-the-art hardware-in-the-loop system. Specifically;

1. A framework for estimating trailer parameters and states, using sensors on the tractor, was developed and implemented in a simulation environment to enhance ESC performance.

   (a) Vehicle mass and road grade were estimated using Recursive Least Square estimation with multiple forgetting factors. Results showed that very good estimates of mass and road grade were achieved.

   (b) Trailer CG position was estimated using Recursive Least Square estimation, and the findings showed very robust estimates of trailer CG longitudinal position. Since the effect of CG height on drive axle loads is minimal compared to CG longitudinal position, this poses problems in sufficiently exciting the estimator to provide robust CG height estimates. As a result CG height estimates were not as robust as CG longitudinal estimates.

   (c) Planar model kinematics for a tractor trailer system was used to calculate
the longitudinal velocity, lateral acceleration and yaw rate of the trailer. The results from the planar model matched the TruckSim vehicle output very well.

(d) A Dual Extended Kalman Filter was set up to estimate trailer roll angle and roll parameters using measured axle loads. The filter was able to accurately predict the trailer roll angle though the maneuvers need to be much longer for the roll parameters to converge to the correct values.

2. Contributions to a state-of-the-art Hardware-in-the-Loop setup for testing heavy truck ESC systems were made. Tractor and trailer models developed using TruckSim were used to validate the HIL system by comparing results to actual test data.

3. The validated HIL simulation system was used to conduct an extensive ESC effectiveness study for cases that are hard to test on the track, indicating that ESC can improve stability limits of heavy truck even at highway speeds. Particularly, vast gains in vehicle stability were noticed on low coefficient of friction surfaces.

4. A path following driver model was created and used to test ESC effectiveness in an actual crash scenario, indicating that ESC can improve vehicle safety and stability in real-world-like accident avoidance maneuvers.

8.2 Dissertation Summary

The research presented in this dissertation involved the development of a scheme to estimate trailer states and parameters using sensors on the tractor and the development of a Hardware in the Loop simulation setup to test heavy truck ESC
systems. Chapter 1 introduced the motivation behind the research and the need for a HIL setup. An extensive literature review in Chapter 2 shows the lack of well validated HIL systems for heavy truck ESC testing in the research arena. The literature review also revealed a need for trailer state and parameter estimation to enhance the ESC capabilities. Chapter 3 details the trailer parameter and state estimation process. The vehicle mass and road grade are first estimated during vehicle motion along a straight line using recursive least square estimation with multiple forgetting factors. This is followed by trailer CG estimation again using recursive least square estimation. It was noticed that the CG height estimation deteriorated for the longer trailer. The CG longitudinal position is then used in a tractor trailer bicycle model to calculate trailer velocities and accelerations. Finally trailer roll angle is estimated using a dual Kalman Filter. Chapter 4 briefly describes the design of the HIL system. Chapter 5 describes in detail the tractor and trailer models in TruckSim. A Volvo tractor along with a 53 foot Fruehauf box trailer and a 28 foot Great Dane trailer were modeled. Different loading conditions were modeled for the trailer models as well. Chapter 6 details the validation process of the vehicle models first, followed by the validation of the HIL simulation. A rigorous validation process was followed, first validating the vehicle models using quasi steady state maneuver (SIS maneuver) followed by dynamic maneuvers (SWD and RSM maneuvers). The validated vehicle models were then incorporated into the HIL simulation setup to validate the whole system by comparison with test results. Finally, Chapter 7 outlines some of the current applications of the HIL system which include a detailed ESC effectiveness study at higher speeds and low friction surfaces. Finally a path following algorithm was developed and used to study the advantage afforded by the ESC system in an actual ESC case.
8.3 Recommendations and Future Research

As indicated in Chapter 3, the robustness of trailer CG height estimation deteriorated for the longer trailer. The effectiveness of this technique for signals measured on an actual vehicle needs to be investigated. The trailer roll angle estimation using DEKF technique, was performed offline and using simulated measurements. The extension of this research would be to implement the trailer parameter and state estimation techniques on an actual vehicle. The author believes that, successful implementation would enhance the effectiveness of the ESC system, increase the adoption of ESC technology and as a consequence save more lives on the road.

The current HIL system is connected to a Bendix ESC system. Since the purpose of the HIL system is to test ESC systems, the next step would be to equip the system so that ESC systems from other manufacturers can be plugged in and tested with minimal This author also believes that there is vast scope for HIL technology with regards to testing advanced driver aids like forward crash warning, crash imminent braking, and even vehicle-to-vehicle communication. Testing these technologies requires expanding capabilities of the HIL simulation setup which would be the next step.
References


APPENDIX A

BALLAST INERTIA CALCULATION

Figure A.1 shows the dimensions of the standard block of weight used to load the trailer. The inertia of the blocks are calculated using the dimensions and the mass of the block using the equations A.0.1, A.0.2 and A.0.3

![Figure A.1: Ballast Block Dimensions](image)

\[ I_{xx} = \frac{1}{12} m \cdot (Y^2 + Z^2) \]  \hspace{1cm} (A.0.1)
\[ I_{yy} = \frac{1}{12} m \cdot (X^2 + Z^2) \]  \hspace{1cm} (A.0.2)
\[ I_{zz} = \frac{1}{12} m \cdot (X^2 + Y^2) \]  \hspace{1cm} (A.0.3)

Where

X - is the length of the block along the x axis
Y - is the length of the block along the y axis
$Z$ - is the length of the block along the $z$ axis
APPENDIX B

CENTER OF GRAVITY AND PARALLEL AXIS

THEOREM EQUATIONS

The center of gravity is calculated using these formulas:

\[
\bar{X}_{cg} = \frac{\sum m_i \cdot X_{cg_i}}{\sum m_i} \tag{B.0.1}
\]

\[
\bar{Y}_{cg} = \frac{\sum m_i \cdot Y_{cg_i}}{\sum m_i} \tag{B.0.2}
\]

\[
\bar{Z}_{cg} = \frac{\sum m_i \cdot Z_{cg_i}}{\sum m_i} \tag{B.0.3}
\]

The parallel axis theorem is given by the equation:

\[
I_{\text{parallel\_axis}} = I_{CG} + m \cdot d^2 \tag{B.0.4}
\]

Now adapting the above equation to the case of the trailer loads, we get:

\[
I_{X_{cg}} = \sum I_{X_{cg_i}} + m_i \cdot [(\bar{Y}_{cg} - Y_{cg_i})^2 + (\bar{Z}_{cg} - Z_{cg_i})^2] \tag{B.0.5}
\]

\[
I_{Y_{cg}} = \sum I_{Y_{cg_i}} + m_i \cdot [(\bar{X}_{cg} - X_{cg_i})^2 + (\bar{Z}_{cg} - Z_{cg_i})^2] \tag{B.0.6}
\]

\[
I_{Z_{cg}} = \sum I_{Z_{cg_i}} + m_i \cdot [(\bar{X}_{cg} - X_{cg_i})^2 + (\bar{Y}_{cg} - Y_{cg_i})^2] \tag{B.0.7}
\]

Where

\( m_i \) - Mass of block \( i \)

\( X_{cg_i} \) - CG X coordinate of mass \( i \)

\( Y_{cg_i} \) - CG Y coordinate of mass \( i \)
$Z_{cg_i}$ - CG Z coordinate of mass $i$

$X_{cg}$ - Aggregate CG $X$ coordinate of all the masses put together

$Y_{cg}$ - Aggregate CG $Y$ coordinate of all the masses put together

$Z_{cg}$ - Aggregate CG $Z$ coordinate of all the masses put together

$I_{CG}$ - Moment of inertia about an axis at the CG

$I_{parallel\_axis}$ - Moment of inertia about an axis parallel to the axis $I_{CG}$

$d$ - Perpendicular distance between the two axes

$I_{X_{cg}}$ - Moment of inertia about the $X_{cg}$ axis

$I_{Y_{cg}}$ - Moment of inertia about the $Y_{cg}$ axis

$I_{Z_{cg}}$ - Moment of inertia about the $Z_{cg}$ axis
APPENDIX C
VOLVO TRACTOR + GREAT DANE TRAILER SINE
WITH DWELL RESULTS

In the figures below, the HIL simulation results on the left are juxtaposed against the experimental results on the right for the sake of comparison.

(a) HIL Simulation  
(b) Experimental Run

Figure C.1: Sine with Dwell Steering Angles
Figure C.2: Sine with Dwell Longitudinal Velocity Comparison

Figure C.3: Sine with Dwell Lateral Acceleration Comparison
Figure C.4: Sine with Dwell Yaw Rate Comparison

Figure C.5: Sine with Dwell Roll Angle Comparison
APPENDIX D
VOLVO TRACTOR + FRUEHAUF BOX TRAILER
RAMP STEER MANEUVER RESULTS

In the figures below, the HIL simulation results on the left are juxtaposed against the experimental results on the right for the sake of comparison.

Figure D.1: Ramp Steer Maneuver Steering Angles
Figure D.2: Ramp Steer Maneuver Longitudinal Velocity Comparison

Figure D.3: Ramp Steer Maneuver Lateral Acceleration Comparison
Figure D.4: Ramp Steer Maneuver Yaw Rate Comparison

Figure D.5: Ramp Steer Maneuver Roll Angle Comparison
APPENDIX E

HIGH SPEED SWD MANEUVER RESULTS (HIGH $\mu$)

E.1 Introduction

This section shows results for the high speed SWD maneuver simulations, for speeds ranging from 50 mph to 65 mph. The simulations were conducted using the HIL simulation system on a high friction surface ($\mu = 0.9$) with ESC On and trailer braking.
E.2 Results for 50 mph SWD Tests ($\mu = 0.9$)

Figure E.1: 50 mph SWD Simulation Results (ESC On with Trailer Braking)
E.3 Results for 55 mph SWD Tests ($\mu = 0.9$)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate

(b) Tractor and Trailer Roll Angles

(c) Tractor Lateral Displacement

Figure E.2: 55 mph SWD Simulation Results (ESC On with Trailer Braking)
E.4 Results for 60 mph SWD Tests ($\mu = 0.9$)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate

(b) Tractor and Trailer Roll Angles

(c) Tractor Lateral Displacement

Figure E.3: 60 mph SWD Simulation Results (ESC On with Trailer Braking)
E.5 Results for 65 mph SWD Tests ($\mu = 0.9$)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate

(b) Tractor and Trailer Roll Angles

(c) Tractor Lateral Displacement

Figure E.4: 65 mph SWD Simulation Results (ESC On with Trailer Braking)
APPENDIX F
HIGH SPEED SWD MANEUVER (LOW $\mu$)

F.1 Introduction

This section shows results for the high speed SWD maneuver simulations, for speeds ranging from 50 mph to 70 mph. The simulations were conducted using the HIL simulation system on a low friction surface ($\mu = 0.5$) with ESC On and trailer braking.
F.2 Results for 50 mph SWD Tests (μ = 0.5)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate
(b) Tractor and Trailer Roll Angles
(c) Tractor Lateral Displacement

Figure F.1: 50 mph SWD Simulation Results (ESC On with Trailer Braking)
F.3 Results for 55 mph SWD Tests ($\mu = 0.5$)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate

(b) Tractor and Trailer Roll Angles

(c) Tractor Lateral Displacement

Figure F.2: 55 mph SWD Simulation Results (ESC On with Trailer Braking)
F.4 Results for 60 mph SWD Tests ($\mu = 0.5$)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate

(b) Tractor and Trailer Roll Angles

(c) Tractor Lateral Displacement

Figure F.3: 60 mph SWD Simulation Results (ESC On with Trailer Braking)
F.5 Results for 65 mph SWD Tests ($\mu = 0.5$)

Figure F.4: 65 mph SWD Simulation Results (ESC On with Trailer Braking)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate

(b) Tractor and Trailer Roll Angles

(c) Tractor Lateral Displacement
F.6 Results for 70 mph SWD Tests ($\mu = 0.5$)

(a) Normalized Tractor Lateral Acceleration and Yaw Rate

(b) Tractor and Trailer Roll Angles

(c) Tractor Lateral Displacement

Figure F.5: 70 mph SWD Simulation Results (ESC On with Trailer Braking)