MODELING AND VALIDATION OF A HEAVY TRUCK MODEL WITH ELECTRONIC STABILITY CONTROL

Thesis

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Validation was performed on an existing heavy truck vehicle dynamics model with electronic stability control. The first stage in this validation process was to study the effect of a torsional stiffness model on the dynamics of a bobtail tractor. By running static maneuvers comparing different torsional stiffness values to the original rigid vehicle model, it was found that the incorporation of a torsional stiffness model had only a minor effect on the overall vehicle response.

The next stage in validation was to compare the response of the simulated tractor to that of the experimental tractor. By looking at the steady-state gains of the tractor, adjustments were made to the model to more closely match the experimental results. These adjustments included suspension and steering compliances, as well as auxiliary roll moment modifications. By adding some new compliances to the model and modifying many of the current compliances, the simulated response of the tractor closely modeled the response seen experimentally.

Once the bobtail validation was completed for the current configuration, the existing 53-foot box trailer model was added to the vehicle model. Similarly to the bobtail tractor model, the effect of torsional stiffness was studied for the trailer model. By again simulating the vehicle response using varying values of torsional stiffness and
comparing these responses with the original rigid model, it was found that the overall effect on vehicle response was minimal for both static and dynamic maneuvers. The final stage in experimental validation for the current tractor-trailer model was to incorporate suspension compliances and modify the auxiliary roll stiffness to more closely model the experimental response of the vehicle.

Following the steady-state validation of the current tractor-trailer model, the dynamic response of the vehicle was tested using a ramp steer maneuver. The current simulation model worked in parallel with an electronic stability control (ESC) model which simulated the workings of a commercial ESC system. The vehicle model was simulated with the ESC system activated and deactivated to study the effectiveness of the ESC system in preventing tractor-trailer rollover. It was found that when the ESC control was activated, the vehicle experienced much greater dynamic forces and did not experience rollover until significantly greater speeds compared to simulations with the ESC switched OFF.

Another focus of the current research was to develop a new 28-foot flatbed control trailer model. Meeting this objective was comprised of determining the physical parameters, performing static and dynamic validation, and adapting the current ESC model to a trailer with only a single axle. RSM simulations were performed, and again showed the ESC-initiated braking to be a significant improvement toward increasing vehicle roll stability.
DEDICATION

To my family and friends for their endless support
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I would first like to express my gratitude to my advisor, Prof. Dennis Guenther, for introducing me to this research and for his guidance during my time at Ohio State. I am especially thankful for his planning which enabled me to finish my degree in a short amount of time, thus allowing me to pursue my true passion in industry. I would also like to thank Dr. Paul Grygier and Dr. Riley Garrott for providing me with the necessary funding and resources to conduct my research at the National Highway Traffic Safety Administration’s Vehicle Research and Test Center.

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CHAPTER 1

INTRODUCTION

1.1 MOTIVATION

In 2000, Americans traveled just over 2.7 trillion miles. As staggering as this number may seem, in 2007, this number increased to over 3 trillion miles as the total number of registered vehicles in the United States increased from 217 million in 2000 to 256 million in 2007 [1]. Taking population growth into consideration, the average American drove 300 additional miles in 2007 than in 2000.

Despite an increase in travel on America’s roadways, advancements in vehicle safety have reduced the number of fatalities due to crashes involving heavy trucks from 5,282 in 2000 to 4,808 in 2007 [1]. This improvement represents a decrease in the number of accidents involving heavy trucks per billion miles traveled from 1.82 in 2000 to 1.51 in 2007.

Even though a significant improvement in heavy truck safety has already been achieved, further safety advancements are necessary. Of the total fatal crashes involving heavy trucks, tractor-trailer jackknife attributed to 6.4% of these in 2007, down from
8.5% in 2000. One shortcoming, however, was that the percentage of fatal accidents involving rollover showed a slight increase from 13.1% in 2000 to 13.9% in 2007 [1]. One area of improvement is in the incorporation of electronic stability control (ESC) systems. ESC systems are comprised of both roll stability control (RSC) systems and yaw stability control (YSC) systems. These systems work to prevent vehicle rollover and to maintain the desired vehicle path as closely as possible.

A computer model has been developed to simulate the partial workings of a commercial ESC system for a heavy truck. Currently this model only contains the RSC component. This model is used to give a better understanding of the dynamics of heavy trucks for a wide range of vehicle maneuvers. Further improvement of this model will lead to a better assessment of heavy truck dynamics, as well as an increased understanding of the effectiveness of ESC systems.

Research was conducted at the National Highway Traffic Safety Administration’s (NHTSA) Vehicle Research and Test Center (VRTC) in East Liberty, Ohio. VRTC is located on the grounds of the Transportation Research Center, Inc. (TRC). NHTSA’s mission is to “save lives, prevent injuries and reduce economic costs due to road traffic crashes, through education, research, safety standards and enforcement activity” [2]. As outlined by the statistics above, the development of a complete heavy truck model ties in closely with NHTSA’s mission. Ultimately, this model will aid in the study of the effectiveness of heavy truck ESC systems as it will reduce the amount of time and money spent on the physical test track.
1.2 **Simulation**

As alluded to in the previous section, computer simulation is the heart of this research. When dealing with heavy trucks, a great amount of equipment is necessary to perform each test. This includes the tractor, trailer, replacement tires, instrumentation, outriggers, video recording, and a professional driver. Each test must be carefully conducted in order to collect the data properly. It is also common to perform many similar trials in order to find a certain threshold for the vehicle. By using a computer simulation, the amount of resources invested is greatly reduced.

Two commercial software packages were used for this research – MATLAB Simulink ® and Mechanical Simulation’s TruckSim ®. A great advantage of TruckSim is that it can be interfaced with Simulink in order to develop more complex vehicle models. TruckSim is used to model the physical vehicle characteristics while Simulink is used to develop additional models such as the ABS and ESC systems. The user can then setup different throttle, braking, steering, and road condition scenarios within TruckSim. This allows a single programmer to conduct a wide variety of tests without being limited by test track availability and environmental conditions. While computer simulations do not completely replace actual vehicle testing, largely due to the complexities of the physical system, they do provide a significant aid in understanding the physics of motion and help to reduce the number of physical experiments.
1.3 Previous Studies

The ABS model was first developed for a doctoral dissertation by Dr. Ashley Dunn [3]. Further detail about this system will not be discussed here, but the reader is referred to the work of Zaugg [4]. This original model, developed by Dunn, was also used for further master’s research conducted by Zagorski [5]. Zagorski’s research focused on the causes and effects of jackknifing for a tractor with two trailers. Additions made to the model for this research included brake chamber dynamics, brake torque and pneumatics systems, and new control algorithms to handle the additional trailer. In 2006, the ABS model was further improved by Shurtz to look at the effect of ABS control parameters on the overall braking of the truck [6]. More information about the development of this model is also outside the scope of the current research and is left to the reader. Most recently, in 2007, an ESC system was added to the heavy truck model by Chandrasekharan [7]. While ESC consists of both RSC and YSC, only the RSC component was developed by Chandrasekharan. While the current model is not a full ESC system, it will be referred to as ESC throughout this document. This model is shown below in Figure 1.1. A more in-depth look at the Tractor ESC block is shown in Figure 1.2.
Figure 1.1: Simulink/TruckSim Model Interaction
Figure 1.2: Current ESC Simulink Model
By looking at the loading of the heavy truck, the ESC model determines the maximum allowable lateral acceleration to safely keep the vehicle from rolling over. If the lateral acceleration surpasses this value, the wheels are braked accordingly in order to slow the vehicle. For example, if the vehicle was starting to enter into a counterclockwise spinout, a restoring braking moment would be applied in the clockwise direction using differential braking of the wheels. While all of the wheels are braked in order to slow the vehicle, greater pressure is applied to the wheels on the right-hand side of the vehicle to create the restoring moment. The ESC model works closely with the ABS model to brake the vehicle in the most effective way possible. If the ESC model calls for a level of brake pressure which leads to wheel lockup, the ABS system immediately reduces the applied brake pressure to allow the wheel to rotate. In addition to improving steering control, preventing lockup yields the maximum possible braking force. By slowing the vehicle, the lateral acceleration is decreased, and thus, the likelihood of a rollover event is decreased.

1.4 **OBJECTIVE**

The objective of the current study is to validate the existing vehicle model to more closely match the experimental results of current VRTC vehicles. The existing heavy truck model in TruckSim uses parameters corresponding to a 1991 Volvo tractor while the current vehicle used by VRTC is a 2006 Volvo. While these vehicles are similar, they do exhibit some physical differences. By looking at the behavior of the vehicle at steady-state, it is possible to determine which parameters need to be modified in order to
more closely match the experimental response of the vehicle. Another objective for the current vehicle model is to study the effect of torsional stiffness on the overall response of the vehicle.

A third objective of this research is to develop a new model in TruckSim for a 28-foot Great Dane flatbed control trailer. This model will then be used as another data point in which to implement the trailer torsional stiffness model and also to study validity of the current ESC model. At the completion of this research, the result will be a more realistic heavy truck model that compares more closely with experimental results. This will aid in the design of safer heavy trucks as well as prepare the stage for the development of a yaw stability control model.

1.5 Thesis Overview

Chapter 2 reviews the current TruckSim and Simulink models for the NHTSA vehicles. This chapter also briefly outlines the steering maneuvers, vehicle loading conditions, and conventions used for this research.

Chapter 3 reviews the parameterization of the current NHTSA vehicles. This chapter also discusses the modifications necessary within TruckSim and the development of a torsional stiffness model. Both bobtail and tractor-trailer combination vehicles are discussed in this chapter. Additionally, some changes to the current ESC model are presented.

Chapter 4 discusses the development of a new model for the Great Dane 28-foot control trailer. This includes the parameterization and the implementation of torsional
stiffness. Modification and implementation of the current ESC model is also discussed in this chapter.

Chapter 5 showcases the simulation results for both the modified Volvo tractor-Fruehauf trailer combination and the new 28-foot control trailer TruckSim models. Specifically, results include torsional stiffness validation, comparison of simulation and experimental vehicle gains, and the effect of ESC activation for a ramp steer maneuver.

Lastly, Chapter 6 discusses overall conclusions from the study. This chapter also outlines some recommendations for future study and development.
2.1 Volvo TruckSim Model

In 1995, The University of Michigan Transportation Research Institute (UMTRI) performed measurements of a 1991 Volvo 6x4 tractor [8]. This test included many parameters, including auxiliary roll moment, roll steer, and tractor compliances. More recently, however, NHTSA has replaced the 1991 Volvo tractor with a similar 2006 Volvo model. Since the current vehicle used by NHTSA is a newer model, the parameters for the 1991 tractor merely served as a rough estimate and were used as a starting point. The values of the parameters of the two tractors were expected to vary between the two designs. The current TruckSim heavy truck model is composed of a mixture between the 1991 and 2006 Volvo 6x4 tractor parameters. The model also includes a 1992 Fruehauf box trailer which is still in use by NHTSA.
2.2 **STEERING MANEUVERS**

2.2.1 *Slowly Increasing Steer Maneuver*

One steering maneuver used to evaluate vehicle performance is the slowly increasing steer, or SIS, maneuver. For this maneuver, the steering hand wheel angle is increased at a constant rate (VRTC uses 13.5 deg/sec) for a certain length of time as the vehicle travels at constant speed. This maneuver is shown in Figure 2.1.

![Graph of Steering Angle vs Time](image)

Figure 2.1: Sample Slowly Increasing Steer Maneuver

The SIS maneuver is useful for evaluating the steady-state, or static, steering gains of the vehicle. These gains are then determined according to a first-order polynomial regression within the linear range and provide useful insight into vehicle response and performances. Typical gains of interest are as follows:
- Roll Angle versus Lateral Acceleration
- Roll Angle versus Yaw Rate
- Roll Angle versus Steer Angle
- Lateral Acceleration versus Steer Angle

### 2.2.2 Ramp Steer Maneuver

Another maneuver commonly used for heavy trucks is the ramp steer maneuver, or RSM. For this maneuver, the steering wheel angle is increased at a steady rate (VRTC uses 175 deg/sec) to a maximum angle, held constant for a certain length of time, and then decreased at the same rate back to zero degrees. This maneuver is shown in Figure 2.2.

![Sample Ramp Steer Maneuver](image)

**Figure 2.2: Sample Ramp Steer Maneuver**
The severity of the maneuver can be adjusted by the rate of increase, the maximum steering wheel angle, and the length of the hold. Using the lateral acceleration versus steer angle gain, the steady-state data is extrapolated to estimate the steering angle which causes the vehicle to reach 0.5 g’s of lateral acceleration. This steering angle is then chosen as the maximum steering wheel angle for VRTC’s RSM maneuvers. Once the steering maneuver commences, the throttle is dropped so that the vehicle coasts through the remainder of the maneuver. Unlike the SIS maneuver, the RSM gives an indication of the dynamic response of the vehicle. Vehicle rollover propensity is analyzed by increasing the initial speed of the maneuver in successive tests until the vehicle rolls over. Experimentally, rollover is deemed as the point where outrigger contact occurs.

2.3 Fruehauf Loading Conditions

2.3.1 High CG Height Gross Vehicle Weight Rating Configuration

The gross vehicle weight rating, or GVWR, was the standard loading condition used for the tractor-trailer combination. This configuration consisted of front and rear ballasts whose characteristics are shown in Table 2.1.

<table>
<thead>
<tr>
<th></th>
<th>Load Mass (kg)</th>
<th>CG Distance Behind Kingpin (mm)</th>
<th>CG Height Above Ground (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front ballast</td>
<td>10464</td>
<td>822</td>
<td>2296</td>
</tr>
<tr>
<td>Rear ballast</td>
<td>8650</td>
<td>11833</td>
<td>2617</td>
</tr>
</tbody>
</table>

Table 2.1: High CG Height GVWR Loading Configuration
The overall trailer CG was 2099 mm above the ground. High CG height GVWR configurations are often used in heavy truck testing as they increase the likelihood of tractor-trailer rollover. The angle at which rollover occurs is depicted in Figure 2.3 and the approximation for small angles is given in Equation (2.1). Thus, as the vertical CG height increase, the critical angle for vehicle rollover decreases.

Figure 2.3: Tractor-Trailer Rollover Threshold Diagram
\[ \theta = \frac{t}{2h} \]  

(2.1)

Where:

\( t \) = track width, in m

\( h \) = vertical CG height, in m

\( \theta \) = critical roll angle, in radians

2.3.2 Low CG Height Lightly Loaded Configuration

Another configuration used in simulations was a tractor-trailer with a low CG height. This arrangement, outlined in Table 2.2, was modeled after the high CG height GVWR loading condition, but with the rear loading removed and the front loading lowered.

<table>
<thead>
<tr>
<th>Load Mass (kg)</th>
<th>CG Distance Behind Kingpin (mm)</th>
<th>CG Height Above Ground (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front loading</td>
<td>10464</td>
<td>822</td>
</tr>
</tbody>
</table>

Table 2.2: Low CG Height Loading Configuration

By lowering the CG location, the rollover propensity of the vehicle is drastically reduced. The overall trailer CG was 1153 mm above the ground – nearly half of the high CG height GVWR case. Without having to worry about rollover until much higher speeds, the tractor-trailer can be subjected to maneuvers in which the lateral acceleration reaches a maximum. This gives a better idea of how the vehicle responds over a wider range of possible lateral accelerations. The use of this configuration is discussed in more detail in §5.1.2.1.
2.4 **SIGN CONVENTION**

Because of differences in experimental data collection and model simulation, it was necessary to establish a common sign convention for the sake of consistency. Since each SIS maneuver has steering in only one direction, its sign convention is outlined below in Table 2.3.

<table>
<thead>
<tr>
<th></th>
<th>Steer to the Left</th>
<th>Steer to the Right</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Steering Angle</strong></td>
<td>(+)</td>
<td>(−)</td>
</tr>
<tr>
<td><strong>Yaw Rate</strong></td>
<td>(+)</td>
<td>(−)</td>
</tr>
<tr>
<td><strong>Roll Angle</strong></td>
<td>(−)</td>
<td>(+)</td>
</tr>
<tr>
<td><strong>Lateral Acceleration</strong></td>
<td>(+)</td>
<td>(−)</td>
</tr>
</tbody>
</table>

Table 2.3: Sign Convention for a Slowly Increasing Steer Maneuver

2.5 **TRUCK NOTATION**

Another difference between experimental testing and simulation is the labeling of the tractor and trailer wheels. The notation used for this thesis is shown in Figure 2.4. For the bobtail case, only wheels numbered 1 through 6 are used as these correspond to the tractor wheels. Additionally, only wheels numbered 1 through 8 are used for the Volvo tractor-Great Dane 28-foot control trailer since it has a single trailer axle.
Figure 2.4: Overhead View of Tractor-Trailer Combination
CHAPTER 3

CURRENT MODEL VALIDATION

3.1 Bobtail Validation

3.1.1 Tractor Torsional Stiffness Model

An important focus of the current research was to study the effect of torsional stiffness on the current model. This addition was also used as part of the validation of the existing Volvo model in TruckSim. Within TruckSim, a simulation box can be activated from the Vehicle: Lead Unit Sprung Mass screen, shown in Figure 3.1, in order to use the torsional stiffness model for the tractor and trailer. If this box is unchecked, TruckSim assumes that the selected vehicles are rigid – that is, that they do not deflect about their longitudinal axes as the vehicle rolls. The effect of torsional stiffness was studied first for the Volvo tractor, and then for the Volvo tractor-Fruehauf trailer combination.
At this point in the research, emphasis was placed on the effect of trailer torsional stiffness on the overall system response. As a result, a great deal of time was not spent in developing the torsional stiffness model for the tractor alone. For the tractor, the torsional stiffness was modeled as $1.0 \times 10^6$ N-m/deg. This value is discussed in more detail in §3.2.1. Simulations were run comparing the above tractor to a tractor with half this torsional stiffness and the original (rigid) tractor for a SIS maneuver. The results of these simulations are shown in Figures 5.1 through 5.8 and are discussed beginning on page 59. In order to achieve critical damping and maintain mathematical stability, the damping for the torsional model was set to a large value. Setting this value too large did
cause the simulation to become unstable, so the value of 1000 N-m/(deg-sec) was chosen as it applied a large amount of damping but did not cause numerical instability. It is also noted in the TruckSim user’s manual that the damping value is not highly influential to the model response, but is merely used for mathematical stability. With the tractor stiffness set to a large value, the vehicle was nearly rigid, and so the remaining setup parameters pertaining to the distance between points used for measurement and the location of the torsional node were not critical. Determination of these parameters will be discussed in more detail with the trailer model. Simulations were conducted to ensure that any changes to these parameters had a minimal effect on the overall torsional response of the tractor. Further validation for the torsional stiffness model was performed for the tractor-trailer combination. The tractor torsional model was setup only because the torsional flexibility box had to be checked for the lead unit in order to use the torsional model for the trailer.

### 3.1.2 Parameter Implementation

The next step toward validation was to look at the linear roll steer coefficients measured by UMTRI, shown in Table 3.1. The linear roll steer coefficient is used to relate the influence of vehicle roll angle to steer angle. These values represent the steer axle with a loading of 10,000 pounds and the drive axles with a loading of 16,000 pounds each. Throughout this document, the title “Original value” pertains to the value of the parameter used in the model prior to the current research whereas the title “Modified value” refers to the final value used in the model at the conclusion of this research.
<table>
<thead>
<tr>
<th></th>
<th>UMTRI</th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steer axle (deg/deg)</td>
<td>0.277</td>
<td>-0.225</td>
<td>0.3</td>
</tr>
<tr>
<td>Leading drive axle (deg/deg)</td>
<td>-0.069</td>
<td>0.11</td>
<td>0</td>
</tr>
<tr>
<td>Trailing drive axle (deg/deg)</td>
<td>0.028</td>
<td>-0.002</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 3.1: Linear Roll Steer Coefficients for Volvo Tractor

Since UMTRI measured both of the drive axle coefficients as having values near zero, these values were also set equal to zero in the model. Thus, the drive axles of the tractor do not contribute to roll steer. Within vehicle dynamics, the signs of parameters are often determined according to different conventions. For validation purposes, it was sometimes necessary to invert the sign and view the effect on the tractor response to determine which value was appropriate. Since the UMTRI value for the steer axle linear roll steer was opposite in sign to that of the current model, simulations were performed using both signs. The best improvement in vehicle response resulted from using the value shown in Table 3.1.

A second improvement to make in the TruckSim model was the steer to wrap ratio for the steer axle. This parameter represents how axle twist relates to steering angle. After some trial and error, it was found that the model performance was improved by changing the sign of this value. The best vehicle response was then found using the value in Table 3.2.

<table>
<thead>
<tr>
<th></th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steer / Wrap ratio</td>
<td>0.14 deg/deg</td>
<td>-0.2 deg/deg</td>
</tr>
</tbody>
</table>

Table 3.2: Axle Interaction Coefficient for Volvo Tractor
Another area of improvement for the model was to incorporate compliances for the steer and drive axles. The original model did not include any compliance information about the tractor. For the steer axle, the lateral compliance coefficient reported by UMTRI was fairly constant over the weight range of 10,000 to 14,000 pounds. This value was unchanged and entered into TruckSim with the value shown in Table 3.3.

<table>
<thead>
<tr>
<th>Lateral Compliance Coefficient (mm/N) [TruckSim: Lateral / Fy]</th>
<th>UMTRI</th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.00 (10^{-5})</td>
<td>0</td>
<td>4.00(10^{-5})</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.3: Modified Steer Axle Compliance for Volvo Tractor

Using TruckSim, it was found that the static loading for the Volvo tractor was approximately 4,300 pounds each for both drive axles. Since this was outside of the range reported by UMTRI and the compliances for the drive axles tended to vary by loading, regressions were performed from the given data and are shown in Figures 3.2 to 3.4. The regression was then extrapolated to determine the value of the coefficient at 4,300 pounds. These values were converted into SI units, entered into TruckSim, and are shown in Table 3.4.
Figure 3.2: Drive Axle Lateral Force Steer Coefficient Regression

Figure 3.3: Drive Axle Aligning Moment Steer Coefficient Regression
Figure 3.4: Drive Axle Lateral Compliance Coefficient Regression

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral Force Steer, deg/N</td>
<td>0</td>
<td>-6.12(10^{-6})</td>
</tr>
<tr>
<td>[TruckSim: Steer / Fy]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aligning Moment Steer, deg/(N-m)</td>
<td>0</td>
<td>2.34(10^{-5})</td>
</tr>
<tr>
<td>[TruckSim: Steer / Mz]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lateral Compliance Coefficient, mm/N</td>
<td>0</td>
<td>1.38(10^{4})</td>
</tr>
<tr>
<td>[TruckSim: Lateral / Fy]</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.4: Modified Drive Axle Compliances for Volvo Tractor

Since the determination of these compliances involved a fairly large extrapolation of the given data, it was necessary to break at this point and check the validity of the model. After running a slowly increasing steer maneuver at 30 mph, the maximum lateral force (Fy) and maximum aligning moment (Mz) for the drive axles were found to be 7765 N and 129 N-m, respectively. By multiplying these values by the coefficients
used from Table 3.4, the contribution of these compliances on vehicle performance is shown in Table 3.5.

<table>
<thead>
<tr>
<th>Maximum steer due to Fy</th>
<th>0.048 deg</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum steer due to Mz</td>
<td>0.0030 deg</td>
</tr>
<tr>
<td>Maximum lateral displacement due to Fy</td>
<td>1.0 mm</td>
</tr>
</tbody>
</table>

Table 3.5: Steering and Lateral Movement Caused by Compliances for Volvo Tractor

Also note that for the steer axle, the maximum lateral force was a little over 13,000 N, contributing to only about 0.5 mm of lateral displacement. Compared to the overall handwheel steering angle of 180 degrees and the displacement of the tractor during the maneuver, these compliances were shown to have a fairly small effect on vehicle response. Since the drive axle wheels are held rigidly, their contribution to vehicle steer should be minimal. This statement is validated since the drive axle compliances only contributed about 0.05 degrees to vehicle steer. While the value of these coefficients would change with the amount of loading for a GVWR configuration, these values provide a good estimate since their contribution to the overall vehicle dynamics is subtle. While the effect of adding compliances was small, the TruckSim tractor gains more closely followed the gains of the experimental data.

By looking at the model response at this point, it was determined that the TruckSim tractor was not experiencing large enough roll angles compared to the experimental tractor. Roll angle is often a function of the center of gravity (CG) height, but can also be controlled by the auxiliary roll moment. The auxiliary roll moment acts in parallel with the suspension to increase (or sometimes decrease) the overall vehicle roll
stiffness. The current model included an auxiliary roll moment for each axle which was linear up to ± 5 degrees, and then quickly became much stiffer outside of 5 degrees to simulate the physical bump stops of the tractor. To allow the tractor to experience a larger roll angle, the auxiliary roll moment of each axle was reduced. After a trial and error process of different reductions in roll stiffness, the tractor most closely matched experimental results for a 25% reduction in auxiliary roll moment for each axle. These values are outlined below in Table 3.6.

<table>
<thead>
<tr>
<th></th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steer axle</td>
<td>1045 N-m/deg</td>
<td>784 N-m/deg</td>
</tr>
<tr>
<td>Leading drive axle</td>
<td>1468 N-m/deg</td>
<td>1101 N-m/deg</td>
</tr>
<tr>
<td>Trailing drive axle</td>
<td>1130 N-m/deg</td>
<td>848 N-m/deg</td>
</tr>
</tbody>
</table>

Table 3.6: Modified Auxiliary Roll Moments for Volvo Tractor

The final area of improvement was in the determination of the tractor steering compliances. The compliance of the tie rod and the steering column are difficult parameters to measure by nature. With the values for these characteristics unknown, a trial and error approach was again taken to determine the values of the parameters which led to the model performance that most resembled the experimental data. The final values of these parameters are listed in Table 3.7.

<table>
<thead>
<tr>
<th></th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tie rod</td>
<td>0.0006 deg/(N-m)</td>
<td>0.0001 deg/(N-m)</td>
</tr>
<tr>
<td>Steering column</td>
<td>0.001 deg/(N-m)</td>
<td>0.00025 deg/(N-m)</td>
</tr>
</tbody>
</table>

Table 3.7: Steering Compliances for Volvo Tractor
Using all of the modified values derived above, the TruckSim and experimental tractor responses were compared for a SIS maneuver at 30 mph with a steering rate of 13.5 deg/sec. These results are shown in Figures 5.9 through 5.13 and are discussed beginning on page 65. Comparing these gains to the gains calculated from the experimental data was an effective way to verify the validity of the TruckSim model.

3.2 TRACTOR-TRAILER COMBINATION VALIDATION

3.2.1 Trailer Torsional Stiffness Measurements

Since one objective of the current research was to develop and implement a torsional stiffness model into the current TruckSim vehicle model, it is important to discuss how these measurements were performed experimentally. After attaching a beam near the front of the trailer, a moment was applied to the beam by hanging known weights on one end and by pulling upward using a pulley system on the other end. In addition to knowing the forces applied, the trailer wheels were placed on load scales to provide a secondary measurement for the applied moment. By measuring the angle at the beam and plotting this against the applied moment, the overall trailer torsional stiffness was determined as the best fit line across the loading range. While the overall trailer torsional stiffness was known, this value represented the parallel combination of the trailer torsional stiffness and the stiffness of the suspension. Since the TruckSim model was only interested in the trailer torsional stiffness by itself, further work was performed to determine this parameter. The trailer torsional stiffness was determined by performing
a linear fit of the applied moment against the trailer twist angle. The trailer twist angle was defined as the difference between the twist angle of the beam and the average of the angles measured directly above each trailer axle. The result of these measurements is shown in Figure 3.5.

![Figure 3.5: Measured Torsional Stiffness for Fruehauf Trailer with Outriggers](image)

**3.2.2 Trailer Torsional Stiffness Model**

Similarly to the tractor torsional stiffness model, the stiffness parameters for the Fruehauf trailer were entered on the *Vehicle: Trailer Sprung Mass* screen. Based on tests conducted at VRTC for the Fruehauf trailer, the torsional stiffness was measured to be 79,522 ft-lb/deg, or 107,824 N-m/deg [9]. Recall that the tractor stiffness was set to 1.0(10^6) N-m/deg, or about 10 times that of the trailer stiffness. In order to test the effect of the torsional stiffness model, TruckSim simulations were run comparing different

---

1 Plot courtesy of VRTC [9]
trailer stiffness values with the rigid tractor-trailer model for a SIS maneuver. Results from these simulations are shown in Figures 5.14 through 5.27 and are discussed beginning on page 70. The torsional damping of the trailer was again set to 1000 N-m/(deg-sec) for the same reasons discussed for the tractor torsional stiffness model. The distance between points used to measure stiffness refers to the setup of the method used to determine the torsional stiffness. The longitudinal distance represents the distance from the application of the weights used to twist the trailer to where the trailer is held stationary. For the Fruehauf trailer, the longitudinal distance was taken as the length of the trailer between the kingpin and the midpoint of the two trailer axles, or 11722.1 mm. The lateral distance used was 1219.2 mm, or half the width of the trailer. The location of the torsional node represents the location on the trailer where no twist occurs. For this trailer, this node occurred at the midpoint of the trailer length, or 5861.1 mm.

Simulations were also run for an RSM steering maneuver to study the dynamic response of the tractor-trailer. Results from these simulations are shown in Figures 5.28 through 5.33 and are discussed beginning on page 81.

### 3.2.3 Parameter Implementation

With the torsional stiffness model completed for the trailer, the next step was to incorporate compliances into the Fruehauf trailer model. Since little physical difference exists between the two trailer axles, the same compliances were used for both. These compliances were initially modeled after the drive axle compliances, and then modified to achieve optimal performance for a GVWR loading case.
Similarly to the bobtail validation, it was necessary to break at this point and verify the validity of the model. The maximum lateral force (Fy) and maximum aligning moment (Mz) for the trailer axles for a SIS maneuver at 25 mph were found to be 34,571 N and 2,396 N-m, respectively. By multiplying these values by the coefficients used from Table 3.8, the contribution of these compliances on vehicle performance is shown in Table 3.9.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral Force Steer, deg/N [TruckSim: Steer / Fy]</td>
<td>0</td>
<td>-4.00(10^{-5})</td>
</tr>
<tr>
<td>Aligning Moment Steer, deg/(N-m) [TruckSim: Steer / Mz]</td>
<td>0</td>
<td>2.64(10^{-6})</td>
</tr>
<tr>
<td>Lateral Compliance Coefficient, mm/N [TruckSim: Lateral / Fy]</td>
<td>0</td>
<td>2.41(10^{-5})</td>
</tr>
</tbody>
</table>

Table 3.8: Fruehauf Trailer Axle Compliances

Again, these values represent a subtle, but positive, improvement to the model response. After the incorporation of these compliances, the tractor-trailer model was still experiencing a greater roll angle than that of the experimental data. To reduce the amount of roll, the auxiliary roll moment of both trailer axles was increased. The auxiliary roll moment was increased by a factor of 10 in order to control the roll response of the trailer.
<table>
<thead>
<tr>
<th></th>
<th>Original value</th>
<th>Modified value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leading trailer axle</td>
<td>1468 N-m/deg</td>
<td>14680 N-m/deg</td>
</tr>
<tr>
<td>Trailing trailer axle</td>
<td>1468 N-m/deg</td>
<td>14680 N-m/deg</td>
</tr>
</tbody>
</table>

Table 3.10: Fruehauf Trailer Axle Auxiliary Roll Moments

Again, this solution was achieved by trial and error until the simulated response more closely modeled the experimental data. Similarly to the auxiliary roll moment for the tractor, the roll moment for the trailer axles was linear up to ± 5 degrees, and then quickly became much stiffer to simulate the physical bump stops of the trailer. The model was compared to experimental datasets for a SIS maneuver at 25 mph. Results from these simulations are shown in Figures 5.34 through 5.38 and are discussed beginning on page 86.

3.3 ESC Model Modifications

While the existing ESC Simulink model [7] formed a solid basis for the current research, some changes were implemented to more closely reflect the physical system. One immediate change to implement was to add the option to remove steer axle braking since a true RSC system does not use steer axle braking. Since the same model may serve as a basis in the future for other tractors with steer axle braking, the calculations in the Simulink block diagram were not removed, but a constant gain block was placed after the final determination of the brake chamber pressure. The value of this block is specified in the initialization script for the model. This block effectively removes any
steer axle braking if it is set equal to zero or applies steer axle braking if it is set equal to one.

After comparing to experimental results from a Volvo tractor which uses a Bendix ESC system, it was noted that the drive axle braking was often held at a higher pressure upon its first activation. Upon ESC activation, the current model quickly applied maximum braking pressure, but then immediately reduced this pressure before effectively slowing down the vehicle to a safe level. An extra subsystem, shown in Figure 3.6, was constructed to more closely match the braking of the physical system. The original ESC activation logic remained unchanged, but the new system tracks the time when the ESC activation takes place and how many times the ESC has been activated during the current simulation. The constant block titled “Hold time” represents the minimum amount of time the ESC will remain activated following its initial activation. The length of the hold was set to 0.3 seconds for this model. Note that holding this value for too long often caused the vehicle to jackknife. This logic works in conjunction with the Brake Severity Estimator, shown in Figure 3.7, to hold the “maximum” pressure for the amount of the “Hold time” block, assuming that it is the first time the ESC has activated during the simulation. The word maximum is used in quotations because the ABS system still decreases the brake pressure if wheel lockup is detected. Commanding the maximum possible pressure simply ensures that the brakes will receive maximum allowable pressure for the duration of the hold. The influence of this block can be effectively eliminated by setting the hold time to zero.
Figure 3.6: Modified ESC Activation Subsystem
Another change to the model involved the trailer braking system. The trailer axle braking was determined by a pulse generator, meant to simulate the workings of an ABS system. The change was to equate the pressures delivered to the left and right sides of the trailer wheels. This was necessary because the actual trailer braking system did not have a valve to apply different pressures to the two sides. Since the vehicle may turn either direction, the maximum of the two pressures was used. While the maximum delivery pressure of the tractor brakes was around 110 psi, only about 60 to 70 psi was applied to the trailer wheels in the physical system since the tractor ESC system has no knowledge of whether the trailer has an ABS system. In case the trailer does not have ABS, applying less pressure reduces the chances of wheel lockup. To limit the output of the
model, a gain of 0.7 was added to the final pressure output which limited the maximum pressure to the desired range.

The final change to the model was in the Lateral Acceleration Predictor block. The ESC system uses this block to look at the current road wheel angle and vehicle speed and then predict the resulting lateral acceleration. If the predicted lateral acceleration is greater than the calculated threshold, the ESC system activates. By using this steady-state preview, the ESC controller can activate sooner, thus helping to avoid a rollover event. In order to determine this threshold, the lateral acceleration gain was determined for a series of SIS tests at increasing speeds. This gain was calculated by dividing the lateral acceleration by the road wheel angle, where the road wheel angle was equal to the steering handwheel angle divided by the tractor steering ratio of 20.5 deg/deg. By plotting the lateral acceleration gain against vehicle speed, the slope of a linear regression yielded the coefficient for the Simulink model. Using experimental data, this coefficient was previously estimated to be 0.0013 g/(deg-mph) [7]. Thus, the lateral acceleration is calculated according to Equation (3.1).
\[ a_y = \alpha \times V \times \delta \]  

(3.1)

Where:
- \( a_y \) = the predicted lateral acceleration, in g’s
- \( \alpha \) = activation threshold coefficient, in g/(deg-mph)
- \( V \) = vehicle velocity, in mph
- \( \delta \) = road wheel angle, in deg

After running the existing model for a GVWR tractor-trailer combination, it was seen that the ESC system was not activating soon enough to keep the vehicle from rolling over. By the time the system activated, the brakes did not have enough time to effectively slow down the vehicle since it was already very close to wheel lift. Referring to Figure 5.38, some difference still existed between the experimental and simulation lateral acceleration gains. Since the TruckSim model experienced slightly higher lateral accelerations for the same steer angle, the coefficient used to predict the lateral acceleration should be slightly higher. In order to determine this coefficient from the model, SIS maneuvers were run at increasing speeds in the range of 20 to 35 mph. The simulated lateral acceleration gain was determined and plotted against the vehicle speed, as shown in Figure 3.8. The slope of this line defined the understeer properties of the heavy truck, so a modified system required a new slope.
While the model can simulate higher speeds, speeds were kept near 25 mph since this was the speed used for model validation. The coefficient for the model was thus shown to be 0.0029 g/(deg-mph), or a little more than double the current coefficient used in the model. Applying this coefficient to the model resulted in earlier braking and thus, a slightly higher rollover speed for the RSM maneuver.

### 3.4 Dynamic Response

With the validation of both the bobtail and the tractor-trailer combination models complete, the final step in model evaluation was to look at the dynamic response of the model. This was completed using an RSM steering maneuver and successively
increasing the initial speed until a rollover event occurred. The results for ESC OFF are provided in Figures 5.39 through 5.44 and are discussed beginning on page 90. Results for ESC ON are shown in Figures 5.45 through 5.59 and are discussed beginning on page 95.
CHAPTER 4

CONTROL TRAILER MODEL DEVELOPMENT

4.1 TRAILER SPRUNG MASS

A second focus of the current research was to develop a new TruckSim model for a Great Dane 28-foot flatbed control trailer. This is another trailer currently used by VRTC in combination with the Volvo tractor. The 28-foot trailer is commonly used in the trucking industry as a control trailer.

4.1.1 CG Characteristics

Since exact measurements of the trailer CG characteristics were unavailable, these properties were first approximated, and then verified with the experimental axle loading measurements, steady-state SIS maneuvers, and rollover characteristics. In order to estimate the sprung mass and longitudinal CG location of the trailer, linear scaling was performed on a 53-foot flatbed trailer model currently existing in TruckSim. These characteristics are shown in Table 4.1.
Similarly to the Fruehauf box trailer, the same outriggers are also used for testing with the Great Dane control trailer [10]. The approximate outrigger CG location was determined by measuring the trailer. The specifications for these outriggers are detailed below in Table 4.2.

<table>
<thead>
<tr>
<th>Mass</th>
<th>676 kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>CG longitudinal distance behind kingpin</td>
<td>3594 mm</td>
</tr>
<tr>
<td>CG vertical height above ground</td>
<td>533 mm</td>
</tr>
</tbody>
</table>

Table 4.2: Class 8 Tractor-Trailer Outrigger Specifications

To compare the final axle loading conditions to the experimental LLVW case, it was also necessary to account for any additional hardware attached to the trailer. This hardware included the ballast frame, anti-jackknife cables, and any other necessary instrumentation. Since these weights were also unknown, only the weight of the ballast frame was determined since it accounted for most of the additional weight. The contribution from the steel ballast frame was estimated to be 233 kg. The frame mass was added to the trailer mass and was used to estimate the vehicle CG location. At this point, it was necessary to perform iterations of the vehicle model until the vertical and longitudinal CG locations were approximately equal to their corresponding experimental values since the CG of the trailer mass was unknown. For reference, the sprung mass of
the trailer and ballast frame only had a combined CG location of 3421 mm behind the kingpin and 1194 mm above the ground. The overall CG location for the sprung mass, including the trailer, ballast frame, and outriggers, was determined according to Equation (4.1) and is shown in Table 4.3.

\[ \bar{x} = \frac{\sum x_i m_i}{\sum m_i} \]  

(4.1)

Where:
\[ \bar{x} = \text{distance from the reference point to the overall loading CG, in mm} \]
\[ x_i = \text{distance from the reference point to the CG of each load, in mm} \]
\[ m_i = \text{mass of each load, in kg} \]

<table>
<thead>
<tr>
<th>Longitudinal distance behind kingpin</th>
<th>3450 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical height above ground</td>
<td>1083 mm</td>
</tr>
</tbody>
</table>

Table 4.3: Overall CG Location of Great Dane Trailer Sprung Mass

After entering the characteristics of the sprung mass CG into TruckSim, comparisons were made for the overall trailer CG and axle loadings [11] between the simulated and experimental data. These results are shown in Table 4.4.
Table 4.4: LLVW Control Trailer Overall CG Position and Axle Weights

<table>
<thead>
<tr>
<th></th>
<th>Experimental</th>
<th>TruckSim</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>CG longitudinal distance behind kingpin</td>
<td>3912 mm</td>
<td>3965 mm</td>
<td>1.3%</td>
</tr>
<tr>
<td>CG vertical height above ground</td>
<td>1208 mm</td>
<td>994 mm</td>
<td>18%</td>
</tr>
<tr>
<td>Steer axle loading</td>
<td>4935 kg</td>
<td>4974 kg</td>
<td>0.8%</td>
</tr>
<tr>
<td>Combined drive axle loading</td>
<td>5811 kg</td>
<td>5775 kg</td>
<td>0.6%</td>
</tr>
<tr>
<td>Trailer axle loading</td>
<td>2785 kg</td>
<td>2786 kg</td>
<td>0.04%</td>
</tr>
<tr>
<td>Total loading</td>
<td>13531 kg</td>
<td>13535 kg</td>
<td>0.03%</td>
</tr>
</tbody>
</table>

Since some uncertainty existed in these experimental measurements, they were used mostly as a guideline. Some variation existed in the placement of the vertical CG, but it was also important to remember that the experimental value was only an estimation. Using a vertical CG height equal to 994 mm placed the CG of the trailer about 21 inches below the deck of the trailer, which should be a fair estimate given the thickness of the trailer frame and the inclusion of the outriggers and the suspension system. Also notably, the axle loading is an extremely good match. Given the number of assumptions made up to this point, these results are more than satisfactory.

Once the trailer sprung mass CG location was determined, the overall inertia for the trailer sprung mass was calculated. This inertia was comprised of the inertia from the trailer/frame combination and the outriggers. Without inertial measurements for the trailer sprung mass, it was necessary to perform more approximations. Inertial calculations were performed by assuming that the trailer sprung mass was approximately a rectangular prism in shape and had a uniform mass density. The measured dimensions of the trailer are shown below in Table 4.5.
<table>
<thead>
<tr>
<th>Length (X direction)</th>
<th>8534 mm (28 ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width (Y direction)</td>
<td>2438 mm (8 ft)</td>
</tr>
<tr>
<td>Thickness (Z direction)</td>
<td>457 mm (18 in.)</td>
</tr>
</tbody>
</table>

Table 4.5: Approximate Dimensions of Great Dane 28-Foot Flatbed Trailer

The length and width measurements of the trailer sprung mass were fairly straightforward. Recalling that the CG location of the trailer and frame only was 3421 mm behind the kingpin, this meant that the CG location was about 4297 mm behind the front of the trailer. Assuming that this measurement was near the center of the trailer, the overall trailer length was estimated to be about double, or 8594 mm. Since this was fairly close to the measured length of 8534 mm, the measured length should be a good estimation of the trailer length. The height measurement required a bit more estimation. The front half of the trailer was fairly thin, about 12 inches, while the rear of the trailer was about 24 inches thick because of an I-beam beneath the deck. Consequently, the overall thickness was estimated to be about 18 inches. Without a highly accurate method to easily measure these dimensions by hand, this simple approximation leaves a lot of room for improvement. The inertia of the trailer sprung mass was then calculated according to Equations (4.2), (4.3), and (4.4).
\[ I_{xx} = \frac{1}{12} m(y^2 + z^2) \quad (4.2) \]

\[ I_{yy} = \frac{1}{12} m(x^2 + z^2) \quad (4.3) \]

\[ I_{zz} = \frac{1}{12} m(x^2 + y^2) \quad (4.4) \]

Where:
- \( I_{xx} \) = roll inertia, in kg-m
- \( I_{yy} \) = pitch inertia, in kg-m
- \( I_{zz} \) = yaw inertia, in kg-m
- \( m \) = trailer mass, in kg
- \( x \) = longitudinal trailer length, in m
- \( y \) = lateral trailer width, in m
- \( z \) = vertical trailer height, in m

In order to calculate the overall inertia of the trailer and the outriggers, it was necessary to apply the parallel axis theorem. This theorem was used to take the inertia of an object at one location and formulate an equivalent inertia at the overall CG location of the trailer. This relationship is shown below in Equation (4.5).
\[ I_{CG} = I + md^2 \]  

(4.5)

Where:

\[ I_{CG} = \text{inertia about overall CG axis, in kg-m}^2 \]

\[ I = \text{inertia about individual component CG axis, in kg-m}^2 \]

\[ m = \text{mass, in kg} \]

\[ d = \text{perpendicular distance from individual component CG to overall CG, in m} \]

By applying the parallel axis theorem to the trailer/frame mass combination and the outriggers, the equivalent inertia about the overall trailer CG was determined. The results of these calculations are shown in Tables 4.6 and 4.7.

<table>
<thead>
<tr>
<th></th>
<th>Inertia about own CG</th>
<th>Distance to trailer CG</th>
<th>Inertia about overall trailer CG</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll inertia (I_{xx})</td>
<td>1857 kg-m^2</td>
<td>0.144 m</td>
<td>1871 kg-m^2</td>
</tr>
<tr>
<td>Pitch inertia (I_{yy})</td>
<td>16.3 kg-m^2</td>
<td>0 mm</td>
<td>16.3 kg-m^2</td>
</tr>
<tr>
<td>Yaw inertia (I_{zz})</td>
<td>1857 kg-m^2</td>
<td>0.550 m</td>
<td>2061 kg-m^2</td>
</tr>
</tbody>
</table>

Table 4.6: Outrigger Inertia Calculations

<table>
<thead>
<tr>
<th></th>
<th>Inertia about own CG</th>
<th>Distance to trailer CG</th>
<th>Inertia about overall trailer CG</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll inertia (I_{xx})</td>
<td>1725 kg-m^2</td>
<td>0.029 m</td>
<td>1728 kg-m^2</td>
</tr>
<tr>
<td>Pitch inertia (I_{yy})</td>
<td>20477 kg-m^2</td>
<td>0 mm</td>
<td>20477 kg-m^2</td>
</tr>
<tr>
<td>Yaw inertia (I_{zz})</td>
<td>22085 kg-m^2</td>
<td>0.111 m</td>
<td>22126 kg-m^2</td>
</tr>
</tbody>
</table>

Table 4.7: Trailer Inertia Calculations
Lastly, the overall characteristics of trailer sprung mass were determined. The equivalent sprung mass consisted of the trailer bed, the ballast frame, and the outriggers. The equivalent inertia was the sum of the trailer/frame inertia and the outrigger inertia and is shown below in Table 4.8.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Sprung mass</td>
<td>4040 kg</td>
</tr>
<tr>
<td>Roll inertia (I_{xx})</td>
<td>3599 kg-m(^2)</td>
</tr>
<tr>
<td>Pitch inertia (I_{yy})</td>
<td>20493 kg-m(^2)</td>
</tr>
<tr>
<td>Yaw inertia (I_{zz})</td>
<td>24188 kg-m(^2)</td>
</tr>
</tbody>
</table>

Table 4.8: Overall Trailer Sprung Mass and Inertia

4.1.2 Torsional Stiffness Model

The final consideration for the trailer sprung mass was to implement the torsional stiffness model. Parameters for this trailer were determined using a method similar to that used for the Fruehauf box trailer, discussed in §3.2.1, since both trailers were measured using the same basic procedure. The only main difference between the two was that since the Great Dane trailer had only a single axle, the trailer twist angle was defined as the difference between the beam angle and the angle measured directly above the lone trailer axle. The resulting trailer stiffness measurements are shown below in Figure 4.1.
The torsional node was again set equal to half the length of the trailer. The damping was set to achieve critical damping and maintain mathematical stability. The parameters entered into TruckSim are shown below in Table 4.9.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal distance between points used to measure stiffness</td>
<td>6794.5 mm</td>
</tr>
<tr>
<td>Lateral distance between points used to measure stiffness</td>
<td>1219 mm</td>
</tr>
<tr>
<td>X coordinate of torsional node</td>
<td>3397.3 mm</td>
</tr>
<tr>
<td>Stiffness about longitudinal axis</td>
<td>4790 N-m/deg</td>
</tr>
<tr>
<td>Damping about longitudinal axis</td>
<td>1000 N-m/(deg-sec)</td>
</tr>
</tbody>
</table>

Table 4.9: Great Dane Trailer Torsional Stiffness Parameters

---

Figure 4.1: Measured Torsional Stiffness for Great Dane Trailer with Outriggers

- Plot courtesy of VRTC [9]
4.2 Loading Configuration

With the development of the lightly-loaded trailer model complete, an additional loading model was created. The 28-foot trailer is tested for both mid and high vertical CG configurations [11], but only the mid CG height loading configuration was constructed for this model.

The mid CG height GVWR configuration consisted of two weight distributions – one near the front of the trailer and one located above the rear axle. Ballast tables, steel frames weighing about 495 lb each and denoted by X’s in the figure, were used to raise the ballast blocks as necessary. This loading condition is shown in Figure 4.2 where all weights are shown in pounds and all measurements are shown in inches.
Figure 4.2: Mid CG Height GVWR Loading Configuration

Each ballast block in this configuration is rectangular in shape (approximately 24 in. by 29 in. by 72 in.). The CG location of the front load was first calculated according to Equation (4.1). Next, the equivalent inertia at this location for each block was calculated using Equations (4.2), (4.3), and (4.4). This procedure was repeated for the rear ballast and the overall characteristics of both loads are listed in Table 4.10.

<table>
<thead>
<tr>
<th></th>
<th>Front ballast</th>
<th>Rear ballast</th>
</tr>
</thead>
<tbody>
<tr>
<td>Payload mass</td>
<td>11077 kg</td>
<td>3892 kg</td>
</tr>
<tr>
<td>CG longitudinal distance</td>
<td>827 mm</td>
<td>6775 mm</td>
</tr>
<tr>
<td>behind kingpin</td>
<td></td>
<td></td>
</tr>
<tr>
<td>CG vertical distance</td>
<td>2144 mm</td>
<td>2368 mm</td>
</tr>
<tr>
<td>above ground</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Roll inertia (I_{xx})</td>
<td>17229 kg-m²</td>
<td>1807 kg-m²</td>
</tr>
<tr>
<td>Pitch inertia (I_{yy})</td>
<td>844 kg-m²</td>
<td>296 kg-m²</td>
</tr>
<tr>
<td>Yaw inertia (I_{zz})</td>
<td>4616 kg-m²</td>
<td>1408 kg-m²</td>
</tr>
</tbody>
</table>

Table 4.10: Mid CG Height GVWR Ballast Characteristics

A comparison was then made between these calculations and the measured experimental loading. The results are shown in Table 4.11.

<table>
<thead>
<tr>
<th></th>
<th>Experimental</th>
<th>TruckSim</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Trailer CG longitudinal</td>
<td>2752 mm</td>
<td>2759 mm</td>
<td>0.3%</td>
</tr>
<tr>
<td>distance behind kingpin</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Trailer CG vertical</td>
<td>1891 mm</td>
<td>1909 mm</td>
<td>1.0%</td>
</tr>
<tr>
<td>height above ground</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steer axle loading</td>
<td>5420 kg</td>
<td>5568 kg</td>
<td>2.7%</td>
</tr>
<tr>
<td>Combined drive axle</td>
<td>15186 kg</td>
<td>14919 kg</td>
<td>1.8%</td>
</tr>
<tr>
<td>loading</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Trailer axle loading</td>
<td>8183 kg</td>
<td>8016 kg</td>
<td>2.0%</td>
</tr>
<tr>
<td>Total loading</td>
<td>28978 kg</td>
<td>28503 kg</td>
<td>1.0%</td>
</tr>
</tbody>
</table>

Table 4.11: Mid CG Height GVWR Experimental Comparison

4.3 ADDITIONAL TRAILER PARAMETERS

The Great Dane control trailer was modeled using many of the same parameters as the Fruehauf box trailer. The major difference was that the 28-foot control trailer had only a single axle, located 6794.5 mm behind the kingpin. A brief description of the suspension, brakes, and tires is listed below.
4.3.1 Suspension

The suspension of the control trailer was originally modeled using the Fruehauf suspension parameters. While the overall roll stiffness of this trailer was measured by VRTC to be 12011 ft-lb/deg [9], this is often a difficult parameter to accurately determine. After determining the auxiliary roll stiffness corresponding to the measured overall roll stiffness, steady-state simulations for a mid CG height GVWR loading configuration showed the roll angle of the trailer to be much too high. By increasing the stiffness of the linear range by about 20 times, the roll angle response of the TruckSim model was a much better match to the experimental data. The auxiliary roll stiffness was set equal to 178,430 N-m/deg. This does not represent an exact calculated number, but was simply just a dataset that already existed in TruckSim and was much larger than the calculated auxiliary roll stiffness. While a baseline auxiliary roll stiffness was calculated, it was only an estimate since the Great Dane trailer suspension system was likely different from the Fruehauf trailer suspension. Another change made to the model was the spring and shock lateral spacing. This was directly measured from the Great Dane trailer and was found to be 965 mm. Lastly, the suspension compliances were removed from the trailer model. With the number of approximations in the model, trying to fine tune compliances was not extremely practical at this point in time. The TruckSim suspension model parameters are shown in Figure 4.3.
Figure 4.3: Great Dane Suspension Parameters
4.3.2 Brakes

The brakes specified for the model were 7.5 kN-m air brakes. None of the maneuvers performed for this research, however, used braking unless it was initiated by the ESC system model. The ESC Simulink model determined the desired braking torque and then replaced the value calculated by the TruckSim model.

4.3.3 Tires

In the absence of a trailer tire model, steer axle tires were also used for the Great Dane trailer model. These are the same tires which are currently used by the Fruehauf trailer model. The tires used have an effective rolling radius of 506 mm and a spring rate of 999 N/mm. Visually, it was seen that the width of the tires on the Great Dane trailer were narrower than those on the Fruehauf trailer. This further suggests that using the steer axle tires from the Fruehauf trailer was only an approximation of the tires that were actually on the Great Dane.

4.4 Steady-State Validation

After developing the parameters for the sprung mass and suspension, the output of the TruckSim model was compared to that of the experimental data. This comparison was once again performed using a slowly increasing steer maneuver at 30 mph and the results are shown in Figures 5.60 through 5.64 and are discussed beginning on page 107.
4.5 **SIMULINK MODEL MODIFICATIONS**

Since the ESC model was previously only used for dual axle trailers, many modifications were necessary to accommodate a single axle trailer. These modifications, which include fairly major changes to the ABS system, are outlined in the subsequent sections.

4.5.1 **TruckSim-Simulink I/O**

The first major change to the model was in the list of inputs and outputs between TruckSim and Simulink. When running a model within TruckSim, the simulation only creates variables necessary to the model. For example, the simulation does not create any variables related to the trailing trailer axle (L5 and R5) for a trailer with only a single axle. Unfortunately, the Simulink model was expecting velocity, angular accelerations, lateral slip angle, and forces for these two wheels. All of these variables were removed from the list of TruckSim outputs and then adjustments were made to the signal sizes coming into the Simulink model. The Simulink model also calculated the braking pressure and surface coefficient of friction for the fifth axle tires. These were removed from the signal of Simulink outputs and from list of inputs in TruckSim.

4.5.2 **ABS Model Modifications**

The current ABS model consisted of five blocks – one for each vehicle axle. While each block worked fairly independently of the others, the modification process was a little more involved than just deleting the block for the second trailer axle. Many of the
signals in Simulink contained multiple channels and then used selectors to split the signal into components. Without the fifth axle, many of the signals now contained only two channels, rather than four. This required many of these selectors to be modified such that each subsequent block received the appropriate inputs. It was pertinent to check each subsystem one at a time and verify the dimensions of each signal to ensure that anything involving the fifth axle was removed from the model. Luckily, the ABS model was developed in such a way that calculations for the second trailer axle were easily removed without influencing the rest of the model. The only real issue was finding and removing every location in the model where calculations were performed for the fifth axle.

4.5.3 ESC Model Modifications

Two main changes were necessary to the ESC model for the Volvo-Great Dane combination. Both of these changes were within the Roll Stability Control block of the Tractor ESC system.

4.5.3.1 Threshold Selector

The first change to implement was in the determination of the critical roll threshold. This value represents the maximum allowed lateral acceleration before the ESC system initiates braking. Since this ESC model was previously used for the Fruehauf box trailer, this threshold had to be adjusted for the 28-foot control trailer. The roll threshold was previously determined from a study looking at the critical lateral acceleration values when the vehicle rolled over. After plotting this curve, the critical threshold was determined as a function of the overall weight of the vehicle. Since this
study had not been performed for the control trailer, the critical threshold was estimated as a linear function according to vehicle weight. The critical value was determined by looking at the maximum lateral acceleration attained during SIS maneuvers for LLVW and GVWR loading configurations. The final parameters chosen for this threshold are shown in Table 4.12.

<table>
<thead>
<tr>
<th>Vehicle Weight</th>
<th>Critical Lateral Acceleration Threshold</th>
</tr>
</thead>
<tbody>
<tr>
<td>13581 kg</td>
<td>0.60 g</td>
</tr>
<tr>
<td>28789 kg</td>
<td>0.33 g</td>
</tr>
</tbody>
</table>

Table 4.12: Great Dane Critical Lateral Acceleration Threshold

4.5.3.2 Lateral Acceleration Predictor

The other change implemented into the model was the coefficient used to predict the lateral acceleration for the GVWR configuration. After performing SIS maneuvers at increasing speeds, the lateral acceleration gain was estimated to be around 0.0032 g/(deg-mph), shown in Figure 4.4.
4.5.3.3 Trailer Axle Braking

Despite removing the calculations for the second trailer axle, no modifications were necessary within the ESC model for the trailer axle braking. Since the model applied the same pressure to all channels of the input signal, it made no difference whether this input was two channels or four channels.

4.6 Dynamic Response

After developing the TruckSim and Simulink models for the Volvo-Great Dane combination, comparisons were made between the experimental and simulation results.

Figure 4.4: Lateral Acceleration Gain for Volvo-Great Dane Combination

gain = 0.0032 g/(deg-mph)
It is important to make one distinction between the two models. The ESC system developed by Chandrasekharan includes only a roll stability component and thus, does not include any yaw stability control. Consequentially, vehicle braking is initiated based simply on the lateral acceleration and not on the yaw rate. With the ESC system activated, the simulation results represent the output from the roll stability model only, and do not include any steer axle braking. The experimental track data was for a complete ESC system, so it includes both RSC and YSC control. As a result, variation is expected between the simulated and experimental results. The dynamic response for the Volvo tractor-Great Dane control trailer is shown in Figures 5.65 through 5.88 and is discussed beginning on page 112.
CHAPTER 5

SIMULATION RESULTS

5.1 VOLVO-FRUEHAUF MODEL VALIDATION

The current model was validated for both bobtail and tractor-trailer combination configurations. Validation consisted of first comparing the effect of torsional stiffness and then on the steady-state gains of the vehicles during a slowly increasing steer maneuver. A dynamic ramp steer maneuver was also performed for the tractor-trailer combination to show the effect of torsional stiffness under dynamic conditions.

5.1.1 Volvo Bobtail Model

5.1.1.1 Static Effect of Torsional Stiffness

To study the effect of torsional stiffness, the Volvo bobtail model was simulated using a SIS maneuver at 30 mph for two different stiffness values. These stiffness values were $1.0 \times 10^6$ N-m/deg and $0.5 \times 10^6$ N-m/deg. The response of the model using these two stiffness values was then compared to the rigid model. Since this was a SIS maneuver, the speed controller was set to maintain a constant speed of 30 mph
throughout the maneuver and to increase the steering handwheel angle at a rate of 13.5 deg/sec.

The implementation of torsional stiffness did not cause any difference in the speed or steering angle throughout the maneuver. The first difference came in the plot of the chassis twist over time (Figure 5.2). As the SIS maneuver progressed, it was seen that the rigid vehicle had zero twist angle. For the tractor with the torsional stiffness model, the chassis twist was proportional to the degree of stiffness. As expected, a decrease in the torsional stiffness by a factor of two resulted in an increase in the amount of chassis twist by a factor of two. While simple, this did prove that the torsional stiffness model was working properly. Also note that the amount of twist was minimal for both cases, which was expected since the tractor stiffness was set to a large value.

Continuing the discussion to the steady-state gains of the vehicle, the current implementation of torsional stiffness in TruckSim showed effectively no difference on the overall vehicle response for any of the three cases. While it may appear that only the rigid model response was plotted, this was just because it was the last series plotted in MATLAB, so it displayed on top. Because of the importance of this result, each plot was zoomed many times to ensure that three separate lines did exist. Again, little difference was expected between the three cases since the torsional stiffness for the tractor was set to such a high value. Similarly, no effective difference was seen in the vehicle path or the vertical wheel forces for any of the three cases.
Figure 5.1: Effect of Torsional Stiffness on Speed and Steering Angle vs. Time for a 30 mph SIS on a Volvo Bobtail Tractor

Figure 5.2: Effect of Torsional Stiffness on Chassis Twist vs. Time for a 30 mph SIS on a Volvo Bobtail Tractor
Figure 5.3: Effect of Torsional Stiffness on Roll Angle vs. Lateral Acceleration for a 30 mph SIS on a Volvo Bobtail Tractor

Figure 5.4: Effect of Torsional Stiffness on Roll Angle vs. Yaw Rate for a 30 mph SIS on a Volvo Bobtail Tractor
Figure 5.5: Effect of Torsional Stiffness on Roll Angle vs. Steering Angle for a 30 mph SIS on a Volvo Bobtail Tractor

Figure 5.6: Effect of Torsional Stiffness on Lateral Acceleration vs. Steering Angle for a 30 mph SIS on a Volvo Bobtail Tractor
Figure 5.7: Effect of Torsional Stiffness on Position for a 30 mph SIS on a Volvo Bobtail Tractor

Figure 5.8: Effect of Torsional Stiffness on Vertical Wheel Force vs. Time for a 30 mph SIS on a Volvo Bobtail Tractor
5.1.1.2 Steady-State Validation

The TruckSim model was compared against the Volvo tractor experimental data collected by VRTC in April, 2008 [12]. Experimental datasets numbered 1763 through 1768 were chosen for the comparison. These datasets included three SIS maneuvers to the left (L) and three SIS maneuvers to the right (R). One minor difference between the model and the experimental results was in the speed input. The experimental test did not use an extremely precise speed controller, which resulted in a variation of about 2 mph for some runs. While this was not a huge difference, some variation was expected between the simulation and experimental results since the model maintained a more consistent speed throughout the entire run.

The simulation response shown in Figure 5.10 was a fairly good representation of the experimental data up to about ±0.4 g’s of lateral acceleration. The gain shown in the legend was determined across the linear range of the response. One important difference to note about the experimental results was the asymmetrical response of the tractor between the left and right-hand maneuvers. The tractor tended to experience greater roll angles for maneuvers to the right than to the left. For the model, the intent was to split the difference between the left and right-hand maneuvers as this created a good representation on average. Similar results are shown in Figures 5.11, 5.12, and 5.13. The simulated response of the vehicle shown in Figure 5.13 was an extremely good match with the experimental data. The asymmetry in roll angle was not present in this response as the tractor exhibited the same response for both left and right-hand maneuvers. A
summary of the approximate gains for the bobtail tractor, determined from the TruckSim predictions shown on the plots, is shown in Table 5.1.

<table>
<thead>
<tr>
<th></th>
<th>Tractor Gain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll Angle / Lateral Acceleration</td>
<td>-2.7 deg/g</td>
</tr>
<tr>
<td>Roll Angle / Yaw Rate</td>
<td>-0.064 deg/deg/s</td>
</tr>
<tr>
<td>Roll Angle / Steering Angle</td>
<td>-0.0067 deg/deg</td>
</tr>
<tr>
<td>Lateral Acceleration / Steering Angle</td>
<td>0.0025 g/deg</td>
</tr>
</tbody>
</table>

Table 5.1: Bobtail Tractor Steady-State Gains
Figure 5.9: Experimental Comparison of Speed and Steering Angle vs. Time for a 30 mph SIS on a Volvo Bobtail Tractor

Figure 5.10: Experimental Comparison of Roll Angle vs. Lateral Acceleration for a 30 mph SIS on a Volvo Bobtail Tractor
Figure 5.11: Experimental Comparison of Roll Angle vs. Yaw Rate for a 30 mph SIS on a Volvo Bobtail Tractor

Figure 5.12: Experimental Comparison of Roll Angle vs. Steering Angle for a 30 mph SIS on a Volvo Bobtail Tractor
Figure 5.13: Experimental Comparison of Lateral Acceleration vs. Steering Angle for a 30 mph SIS on a Volvo Bobtail Tractor
5.1.2 Volvo-Fruehauf Combination Model

5.1.2.1 Static Effect of Torsional Stiffness on Low CG Configuration

Similarly to the bobtail case, the Volvo-Fruehauf combination model was simulated using a SIS maneuver at 30 mph for three different trailer stiffness values. The steering controller again increased the steering handwheel angle at 13.5 deg/sec. The three stiffness values used were 107,824 N-m/deg, 53,912 N-m/deg, and 26,956 N-m/deg. Once again, the response of the model for all three stiffness values was compared to the rigid vehicle model. Note that during these and all subsequent simulations, the tractor stiffness was held constant at $1.0(10^6)$ N-m/deg. A low CG height lightly-loaded configuration was used to allow the vehicle to experience a larger range of lateral accelerations without having to worry about rollover.

As with the bobtail model, the different stiffness values did not have any effect on the vehicle speed or the steering input. Differences in the chassis twist were apparent in the trailer depending on the stiffness of the trailer. Similarly to the bobtail model, a fairly linear relationship existed between torsional stiffness and chassis twist angle. Looking forward to the steady-state vehicle gains starting with Figure 5.16, some differences did exist between the rigid vehicle model and the torsional stiffness vehicle model. Since the lines overlapped, each plot was zoomed to verify the location of each. For the roll angle relationships, all three trailer stiffness values showed the same response. This response was largely equal to the rigid vehicle model up to a tractor lateral acceleration of about 0.55 g’s, or a roll angle of about 1.6 deg. While this difference was present, it was important to remember this level of lateral acceleration was
only attainable because this simulation used a lightly-loaded vehicle with a low center of gravity. As will be shown in subsequent sections, a high CG height GVWR tractor-trailer combination cannot reach lateral acceleration this high without rolling over. The difference between the current TruckSim rigid and torsional stiffness models was not prevalent in the steady-state lateral acceleration gain shown in Figure 5.19, suggesting that the implementation of torsional stiffness affected mainly the roll angles of the tractor-trailer. Differences in vertical wheel force were also minimal between the four simulations.

The same maneuver was also run for the case of 45 mph. Again, since this was a low CG height loading configuration, the vehicle did not roll over, but instead achieved its maximum possible lateral acceleration before entering into a tractor-trailer jackknife. Little difference was shown between the four simulations up to a tractor lateral acceleration of around 0.6 g’s for this case. From this point, few anomalies existed between this case and the 30 mph case, so further discussion of the results is not necessary. The results from these simulations are included in Appendix A.
Figure 5.14: Effect of Torsional Stiffness on Speed and Steering Angle vs. Time for a 30 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.15: Effect of Torsional Stiffness on Chassis Twist vs. Time for a 30 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.16: Effect of Torsional Stiffness on Roll Angle vs. Lateral Acceleration for a 30 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.17: Effect of Torsional Stiffness on Roll Angle vs. Yaw Rate for a 30 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.18: Effect of Torsional Stiffness on Roll Angle vs. Steering Angle for a 30 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.19: Effect of Torsional Stiffness on Lateral Acceleration vs. Steering Angle for a 30 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.20: Effect of Torsional Stiffness on Vertical Wheel Force vs. Time for a 30 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination
5.1.2.2 Static Effect of Torsional Stiffness on GVWR Configuration

To continue looking at the effect of the torsional stiffness model for different stiffness values, simulations were conducted for the high CG height GVWR loading case. Again, the simulations used a constant speed of 30 mph, a slowly increasing steer maneuver with a steering rate of 13.5 deg/sec, and different trailer torsional stiffness values. For this case, however, the vehicle did roll over regardless of the torsional stiffness value due to the higher center of gravity. Looking at the response of the vehicle prior to rollover, it was seen that little difference existed between the rigid model simulations and the torsional stiffness simulations. Referring to Figure 5.23, the simulations with torsional stiffness showed slightly larger roll angles than the rigid model. Similar results are seen for the other steady-state gains for this testing configuration. One difference between these and the low CG height simulations was that the roll angle showed some variation between different torsional stiffness values. To conclude the results for the lightly-loaded low CG height and high CG height GVWR cases, the implementation of trailer torsional stiffness had only a minimal effect for a static slowly increasing steer maneuver.
Figure 5.21: Effect of Torsional Stiffness on Speed and Steering Angle vs. Time for a GVWR Volvo Tractor-Fruehauf Box Trailer Combination at 30 mph

Figure 5.22: Effect of Torsional Stiffness on Chassis Twist vs. Time for a GVWR Volvo Tractor-Fruehauf Box Trailer Combination at 30 mph
Figure 5.23: Effect of Torsional Stiffness on Roll Angle vs. Lateral Acceleration for a GVWR Volvo Tractor-Fruehauf Box Trailer Combination at 30 mph

Figure 5.24: Effect of Torsional Stiffness on Roll Angle vs. Yaw Rate for a GVWR Volvo Tractor-Fruehauf Box Trailer Combination at 30 mph
Figure 5.25: Effect of Torsional Stiffness on Roll Angle vs. Steering Angle for a GVWR Volvo Tractor-Fruehauf Box Trailer Combination at 30 mph

Figure 5.26: Effect of Torsional Stiffness on Lateral Acceleration vs. Steering Angle for a GVWR Volvo Tractor-Fruehauf Box Trailer Combination at 30 mph
Figure 5.27: Effect of Torsional Stiffness on Vertical Wheel Force vs. Time for a GVWR Volvo Tractor-Fruehauf Box Trailer Combination at 30 mph
5.1.2.3 Dynamic Effect of Torsional Stiffness on GVWR Configuration

To gain a more complete view into the trailer torsional stiffness model, simulations were also conducted for a ramp steer maneuver. As outlined previously, this maneuver represents how the vehicle responds in a more dynamic situation. To choose the speed for this comparison, the model was first run with the ESC system OFF at increasing speeds to rollover. 25 mph represented the maximum allowable speed prior to vehicle rollover. The maximum steering angle was set to 199 degrees for this maneuver, with a dwell time of six seconds before returning to zero.

For this maneuver, again no significant difference existed between the different torsional stiffness values. Beginning with the load transfer, shown in Figure 5.29, the implementation of torsional stiffness had little effect on the response of the trailer, but increased the load transfer on the tractor. The flexible model response was slightly more extreme because the maximum load transfer occurred for a longer duration than for the rigid model. Both models showed only minor differences in lateral acceleration. Continuing to Figure 5.31, the rigid model experienced a smaller roll angle for both the tractor and trailer compared to the flexible model. As the torsional stiffness was decreased, the roll angle of the tractor and trailer both showed slight increases over most of the simulation range. Both models also experienced about the same yaw rates for the tractor and trailer.

Further analysis came from the phase space diagram for the roll of the tractor and trailer. The roll phase space gave additional insight into the overall severity of the
maneuver, plotting the roll rate as a function of the roll angle. For each maneuver, each variable was initially around zero, and then increased or decreased accordingly throughout the maneuver. For stable ramp steer maneuvers, the ending point will also be around the origin as both the roll rate and roll angle return to near zero. If the vehicle were to roll over or become unstable, the phase space line would trail off drastically and would not form a nearly closed loop. Looking at Figure 5.33, it was seen that the flexible model experienced slightly larger roll angles and roll rates for most of the maneuver compared to the rigid model. At the maximum, the tractor and trailer roll about half of a degree more for the flexible model than for the rigid model.

Overall, some differences in dynamic vehicle response existed between the rigid model and the torsional stiffness model. The difference between the two was subtle enough that changing the trailer stiffness did not lead to drastically different vehicle responses between the cases. The two models were fairly comparable, but including the torsional stiffness represented a more complete vehicle model.
Figure 5.28: Effect of Torsional Stiffness on Speed and Steering Angle vs. Time for a 25 mph RSM on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.29: Effect of Torsional Stiffness on Load Transfer vs. Time for a 25 mph RSM on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.30: Effect of Torsional Stiffness on Lateral Acceleration vs. Time for a 25 mph RSM on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.31: Effect of Torsional Stiffness on Roll Angle vs. Time for a 25 mph RSM on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.32: Effect of Torsional Stiffness on Yaw Rate vs. Time for a 25 mph RSM on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.33: Effect of Torsional Stiffness on Roll Rate vs. Roll Angle for a 25 mph RSM on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
5.1.2.4 Steady-State Validation

The final step in validating the Volvo tractor-Fruehauf trailer combination torsional stiffness model was to compare the simulation results to experimental results collected by VRTC [12]. For this comparison, six experimental maneuvers including three to the left and three to the right, numbered 1633 through 1638, were considered. The data collected was for a 13.5 deg/sec SIS maneuver at 25 mph. The loading used for this test was a high CG height GVWR configuration.

While the deviation between the experimental and simulation models was slightly greater than that for the Volvo bobtail case, the model was still a largely accurate representation of vehicle motion. Again, it is worth noting that the experimental roll angle experienced by the tractor for maneuvers to the right was slightly greater than for maneuvers to the left. This difference was not quite as obvious here as for the bobtail case. The largest deviation between the model and experimental roll angles was only about 0.3 deg across the entire range. Also, the relationship between tractor lateral acceleration and steer angle was represented fairly well by the model. The model, however, tended to overestimate the trailer lateral acceleration across most of the steering range. A summary of the simulation gains for each case is shown in Table 5.2.

<table>
<thead>
<tr>
<th></th>
<th>Tractor Gain</th>
<th>Trailer Gain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll Angle / Lateral Acceleration</td>
<td>-4.7 deg/g</td>
<td>-8.1 deg/g</td>
</tr>
<tr>
<td>Roll Angle / Yaw Rate</td>
<td>-0.11 deg/deg/s</td>
<td>-0.16 deg/deg/s</td>
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<tr>
<td>Roll Angle / Steering Angle</td>
<td>-0.0063 deg/deg</td>
<td>-0.013 deg/deg</td>
</tr>
<tr>
<td>Lateral Acceleration / Steering Angle</td>
<td>0.0018 g/deg</td>
<td>0.0018 g/deg</td>
</tr>
</tbody>
</table>

Table 5.2: GVWR Volvo Tractor-Fruehauf Trailer Combination Steady-State Gains
Figure 5.34: Experimental Comparison of Speed and Steering Angle vs. Time for a 25 mph SIS on a GVWR Volvo Tractor-Fruhauf Box Trailer Combination

Figure 5.35: Experimental Comparison of Roll Angle vs. Lateral Acceleration for a 25 mph SIS on a GVWR Volvo Tractor-Fruhauf Box Trailer Combination
Figure 5.36: Experimental Comparison of Roll Angle vs. Yaw Rate for a 25 mph SIS on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.37: Experimental Comparison of Roll Angle vs. Steering Angle for a 25 mph SIS on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.38: Experimental Comparison of Lateral Acceleration vs. Steering Angle for a 25 mph SIS on a GVWR Volvo Tractor-Fruhauf Box Trailer Combination
5.2 VOLVO-FRUEHAUF MODEL ROLLOVER

Now that the validation of the Volvo tractor-Fruehauf trailer model was complete, the vehicle was subjected to a ramp steer maneuver with the ESC system set to OFF and to ON. A high CG height GVWR loading condition was used for this comparison. Once again, a ramp steer maneuver of 199 degrees was used for this maneuver. The initial speed was then increased in one mile per hour increments until vehicle rollover occurred.

5.2.1 ESC OFF

For the case with the ESC system deactivated, the model predicted tractor-trailer rollover at 26 mph. Experimental data was not available for comparison. Rollover was defined as the speed at which the maximum trailer roll angle surpassed six degrees. While the TruckSim model did not actually show vehicle rollover until higher roll angles, the simulation was in the nonlinear range at this point which challenged the validity of the simulation results. Experimentally, rollover was generally seen when the trailer roll reached around six degrees.

As the speed gradually increased, the amount of load transfer increased until wheel lift occurred, and then increased further to the point where the vehicle cannot recover. For this case, the vehicle rolled over when it reached about 0.37 g’s of lateral acceleration for the tractor and about 0.33 g’s for the trailer. This corresponded to a time about three seconds into the maneuver where the roll angle began to diverge more rapidly. From the plot of the yaw rates shown in Figure 5.43, it was also seen that the
simulation at 26 mph deviated from the previous simulations shortly after three seconds into the maneuver. As stated previously, the phase space diagram for tractor and trailer roll can be useful for comparing the severity of numerous maneuvers. In this case, the four simulations showed little variation near the beginning of the run, but the dynamics of the increase in speed quickly took over as the roll rate and roll angle increased more dramatically. Figure 5.44 also showed that the vehicle became unstable at 26 mph as the roll phase space curve reached much larger bounds than at the slower speeds.
Figure 5.39: Speed and Steering Angle vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.40: Load Transfer vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.41: Lateral Acceleration vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.42: Roll Angle vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.43: Yaw Rate vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.44: Roll Rate vs. Roll Angle for a RSM with ESC OFF on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
5.2.2 ESC ON

By activating the ESC system, the speed at which the vehicle rolled over was drastically increased. For the same ramp steer maneuver with the ESC activated, the vehicle did not roll over until 37 mph – 11 mph higher than with the ESC deactivated. The first impact to notice was that the ESC system greatly reduced the duration and magnitude of the load transfer for the tractor and trailer. This effect was important because this meant that the system was slowing the vehicle enough such that all 18 wheels remained in contact with the roadway for a greater amount of time, leading to increased stability and greater available braking force.

Considering Figure 5.48, the ESC system allowed the tractor to achieve much higher levels of lateral acceleration than seen for the case without ESC. The vehicle did not roll over until the tractor lateral acceleration was around 0.49 g’s. Note, however, that a trailer lateral acceleration of around 0.31 g’s caused the vehicle to roll over, thus reinforcing the notion that rollover is initiated by the trailer. Figure 5.50 showed the levels of longitudinal acceleration experienced by the tractor and trailer throughout each simulation. One feature to notice was that as the speed increased, the longitudinal acceleration of the tractor became much more active and remained at greater levels for a longer period of time. As load transfer began to take place and the wheels lost vertical force, some of the braking force was lost and consequently, longitudinal deceleration decreased.
While the roll angles and roll rates of the vehicle with the ESC system activated were similar in magnitude to those with the ESC system deactivated, it was important to remember that they correspond to speeds which are 11 mph higher. This increase is significant as the vehicle experiences much more powerful forces than at lower speeds, but still does not roll over or lose control.

Lastly, the braking initiated by the ESC system is shown in Figure 5.58. Because these steering maneuvers were all performed to the left, greater braking pressure was applied to the drive wheels on the right in order to apply a restoring moment for the vehicle, thus lowering the lateral acceleration and preventing rollover. While the braking pressures applied were nearly equal for the four simulations, the duration of the braking increased as the vehicle speed increased. Also note that no steer axle braking was present and that all four trailer tires were braked equally, as discussed previously in §3.3.

From the preceding analysis, the effect of electronic stability control on rollover prevention is incredible. Not only does ESC prevent vehicle rollover until much greater speeds, but it also controls the vehicle despite drastic increases in kinetic energy.

Comparing the responses of the simulation and the experimental data, both vehicles showed similar results as far as the rollover point was concerned. While the same steering profile was used for both, it was seen that the speed of the physical system was decreased a greater amount than the speed modeled by TruckSim. This was also seen in the duration of the brake pressure application, shown in Figure 5.59. One important difference to note was the difference in the braking model used for the vehicle. The current model did not apply any steer axle braking, while the physical vehicle did
apply steer axle braking, but no braking on either trailer axle. This was because the current model did not represent a full ESC system like the physical vehicle. The physical testing was also performed with the bogey offline, thus removing the trailer axle braking. Due to these differences, the two results could not be concretely compared, but served merely as a reference for the TruckSim model up to the current point in time.

Comparing the lateral acceleration between the two results, the lateral acceleration predicted by the model was slightly lower than seen in the physical data for both the tractor and trailer. Looking at the longitudinal acceleration, the model allowed for greater deceleration than the physical test. Remember that longitudinal acceleration is directly related to the braking pressure applied, so differences between the two models were expected. It was also seen that the physical system maintained the longitudinal acceleration near its maximum for a longer amount of time, thus further slowing the vehicle and decreasing the lateral acceleration. The roll angle predicted by the model was slightly larger than the experimental data, again caused by differences in speed and lateral acceleration. Similar response was seen in the roll angle phase space as the TruckSim model showed slightly higher roll rates and roll angles compared to the experimental data. The yaw rates predicted by the model were also comparable to those shown in the experimental data. Both sets attained nearly the same maximum yaw rate for the tractor and trailer, but some differences showed as the vehicle speeds began to differ.

Overall, the difference in response between the simulation and the experimental vehicles was fairly small, especially given the inherent differences in the braking system.
of the two vehicles. Both systems showed vehicle rollover around 37 mph. Since these results were in close agreement, it is reasonable to believe that the current ESC system can be expanded to also include a yaw stability control component.
Figure 5.45: TruckSim Speed and Steering Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.46: Experimental Speed and Steering Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.47: TruckSim Load Transfer vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.48: TruckSim Lateral Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.49: Experimental Lateral Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.50: TruckSim Longitudinal Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.51: Experimental Longitudinal Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.52: TruckSim Roll Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.53: Experimental Roll Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.54: TruckSim Yaw Rate vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.55: Experimental Yaw Rate vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.56: TruckSim Roll Rate vs. Roll Angle for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.57: Experimental Roll Rate vs. Roll Angle for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
Figure 5.58: TruckSim Brake Pressure vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination

Figure 5.59: Experimental Brake Pressure vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Fruehauf Box Trailer Combination
5.3 **Volvo-Great Dane Model Steady-State Validation**

To validate the steady-state response of the Volvo tractor-Great Dane 28-foot control trailer, a SIS maneuver was performed at 30 mph. The trailer loading for this comparison was a mid CG height GVWR configuration. Experimental datasets 2286 through 2291 were used for the comparison. While SIS maneuvers have been performed heretofore with the ESC system deactivated, this particular set of maneuvers was performed with the ESC ON. Since interest was only in the steady-state response of the vehicle, however, only the experimental data prior to the ESC system activation was used for the comparison. This was a valid simplification to make given that the current ESC Simulink model is known to be inherently different from the physical ESC system. Thus, the model was simulated in TruckSim with no external reference to the ESC Simulink model.

The speed profile was a very good match between the simulation and experiment. Comparing the roll angle versus lateral acceleration shown in Figure 5.61, the response of the vehicle model looked fairly good for the trailer, but did show some deviation for the tractor. For both cases, the simulation overestimated the amount of lateral acceleration seen experimentally. Continuing to the roll angle versus yaw rate comparison, the TruckSim simulation again matched the experimental data fairly well for the trailer, but again showed more variation for the tractor. Similar responses were seen when looking at the steer angle gains. Considering the roll angle versus steering angle plot, both the tractor and trailer were a good match to the experimental data. Part of the deviation for
the tractor was attributed to the asymmetry in the design, discussed previously. Lastly, the lateral acceleration versus steering angle plot showed a very linear relationship between the two variables, but again, it was seen that the model overestimated the amount of lateral acceleration for each steer angle. A summary of the simulation gains for each case is shown in Table 5.3.

<table>
<thead>
<tr>
<th></th>
<th>Tractor Gain</th>
<th>Trailer Gain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roll Angle / Lateral Acceleration</td>
<td>-7.2 deg/g</td>
<td>-8.0 deg/g</td>
</tr>
<tr>
<td>Roll Angle / Yaw Rate</td>
<td>-0.16 deg/deg/s</td>
<td>-0.19 deg/deg/s</td>
</tr>
<tr>
<td>Roll Angle / Steering Angle</td>
<td>-0.015 deg/deg</td>
<td>-0.021 deg/deg</td>
</tr>
<tr>
<td>Lateral Acceleration / Steering Angle</td>
<td>0.0026 g/deg</td>
<td>0.0026 g/deg</td>
</tr>
</tbody>
</table>

Table 5.3: GVWR Volvo Tractor-Great Dane Trailer Combination Steady-State Gains
**Figure 5.60:** Experimental Comparison of Speed and Steering Angle vs. Time for a 30 mph SIS on a GVWR Volvo Tractor-Great Dane Trailer Combination

**Figure 5.61:** Experimental Comparison of Roll Angle vs. Lateral Acceleration for a 30 mph SIS on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.62: Experimental Comparison of Roll Angle vs. Yaw Rate for a 30 mph SIS on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.63: Experimental Comparison of Roll Angle vs. Steering Angle for a 30 mph SIS on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.64: Experimental Comparison of Lateral Acceleration vs. Steering Angle for a 30 mph SIS on a GVWR Volvo Tractor-Great Dane Trailer Combination
5.4 Volvo-Great Dane Model Rollover

With the validation of the Volvo-Great Dane model complete, the vehicle was subjected to a ramp steer maneuver with the ESC system set to OFF and to ON. A mid CG height GVWR loading condition was used for this comparison. Like the experimental runs, ramp steer of 199 degrees at a rate of 175 deg/sec was used for the simulations of this maneuver. The speed was then increased in one mile per hour increments until vehicle rollover was obtained. As discussed previously, the vehicle was assumed to experience rollover when the trailer roll angle exceeded six degrees.

5.4.1 ESC OFF

The results of the TruckSim simulation were compared against those of the experimental data. For comparison, the TruckSim results are shown on the left and the experimental results are shown on the right on the following pages. For the cases with the ESC system deactivated, rollover was predicted by the TruckSim model at 27 mph while outrigger contact was predicted by the experimental data at 31 mph.

Overall, the shape of the speed profiles of the simulation and the physical test were in good agreement. The experimental data showed a slightly faster decay in speed, due to the increased drag at higher speeds. Continuing the discussion to the lateral acceleration, the simulation gave a fairly reasonable representation of the physical system for both the tractor and trailer. An offset in the maximum levels of lateral acceleration exists, but this was a result of the difference in speed. Looking at the yaw rate, the
simulated and experimental responses were in close accord for the tractor and trailer up to
the speed where rollover/outrigger contact occurred. Both increased at about the same
rate during the increasing steering maneuver and then held nearly constant during the
constant steering portion. The yaw rate of the experimental vehicle was again slightly
larger than that of the simulation due to the difference in speed.

While the vehicle response predicted by the model looked largely accurate up to
this point, the roll angle predicted by the simulation showed more deviation between
successive speeds compared to the physical data. The roll angle for the experimental data
tended to group together for successive speeds and then finally broke away when
outrigger contact occurred. Looking at the roll angle predicted by the simulation, this
transition was a little more gradual. This anomaly was also seen in the roll rate versus
roll angle phase space plots as the simulation showed a more gradual increase in the
maneuver severity than the experimental data.
Figure 5.65: TruckSim Speed and Steering Angle vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.66: Experimental Speed and Steering Angle vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.67: TruckSim Lateral Acceleration vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.68: Experimental Lateral Acceleration vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.69: TruckSim Yaw Rate vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.70: Experimental Yaw Rate vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.71: TruckSim Roll Angle vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.72: Experimental Roll Angle vs. Time for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.73: TruckSim Roll Rate vs. Roll Angle for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.74: Experimental Roll Rate vs. Roll Angle for a RSM with ESC OFF on a GVWR Volvo Tractor-Great Dane Trailer Combination
5.4.2 ESC ON

For the case of ESC ON, the TruckSim model results showed a fairly large amount of deviation to the experimental results. Some major differences were seen in the speed profiles between the two tests, and it is important to remember that the TruckSim model uses only roll stability control while the physical data represents a full ESC system. Consequently, differences are inherent between the simulated and experimental results.

Despite these differences, both sets of results did show similar patterns of lateral acceleration for the tractor and trailer up to about three seconds into the maneuver. After this point, the differences in the two responses were very distinct. The current Simulink ESC model calculated a maximum lateral acceleration threshold and then tried to keep the vehicle at or below this threshold. This was seen in Figure 5.77 between the times of four and six seconds. Looking at the experimental data showed a much different relationship, however. Once the ESC activated, it remained activated until about six seconds into the run, at which point the lateral acceleration was below 0.1 g’s. Looking at the longitudinal acceleration plots, the simulation did not maintain the maximum deceleration for nearly as long as the physical controller did. This difference was also seen in the brake pressure plots. While the simulation and physical data applied different maximum pressures, the physical controller maintained the maximum pressure for much longer than the model controller. The model controller logic prevented the vehicle from rolling over, but still allowed the vehicle to maintain its intended path and speed of travel as closely as possible.
As with the case of ESC OFF, the simulation results closely modeled the experimental yaw rates up to about two seconds into the maneuver. Again, differences after this time were due to the differences in speed and braking between the two sets. As seen previously, the roll angle and roll rates predicted by the model were similar to those seen in the physical data with the exception that the TruckSim model roll angle increased a little more gradually. This difference relates back to the reasons discussed in §5.4.1 and also the differences between the actual ESC system and the current ESC model which includes only an RSC component.

For the simulations with the ESC OFF, the overall vehicle response was not bad except for an offset in vehicle speed of about 4 mph. The model with ESC ON showed an offset of about 9 mph, but again, remember that some of this offset was caused by differences between the current Simulink ESC model and the physical ESC controller.

When dealing with a dynamic maneuver like the RSM, differences in physical parameters become more important. This is especially the case as the vehicle is pushed to its limits to induce rollover. Differences between the model and physical response can be largely attributed to the absence of an accurate trailer tire model and the fact that many of the 28-foot trailer parameters were simple estimations.
Figure 5.75: TruckSim Speed and Steering Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.76: Experimental Speed and Steering Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.77: TruckSim Lateral Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.78: Experimental Lateral Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.79: TruckSim Longitudinal Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.80: Experimental Longitudinal Acceleration vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.81: TruckSim Brake Pressure vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.82: Experimental Brake Pressure vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.83: TruckSim Yaw Rate vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.84: Experimental Yaw Rate vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.85: TruckSim Roll Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.86: Experimental Roll Angle vs. Time for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination
Figure 5.87: TruckSim Roll Rate vs. Roll Angle for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination

Figure 5.88: Experimental Roll Rate vs. Roll Angle for a RSM with ESC ON on a GVWR Volvo Tractor-Great Dane Trailer Combination
CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

With the conclusion of this research, a torsional stiffness model was successfully implemented into TruckSim. After conducting numerous modifications and simulations with the current Volvo-Fruehauf model, the implementation of the torsional stiffness model was validated. Both SIS and RSM steering maneuvers were used for this validation. For the SIS runs, little difference was seen between the model which used torsional stiffness and the original rigid model. A slight amount of difference was seen for the RSM runs, but these differences generally took place at extreme levels for low CG height loading conditions. These extreme levels are not experienced by a high CG height tractor-trailer as the vehicle is already experiencing a rollover event. The overall simulation results were validated using the TruckSim torsional stiffness chassis model.

For the Volvo-Fruehauf model, ramp steer maneuvers were conducted with the ESC system switched ON and OFF. With the ESC OFF and ON, the model showed vehicle rollover at 26 mph and 37 mph, respectively. The braking initiated by the ESC
system was shown to make a significant contribution to vehicle stability and rollover prevention.

Another outcome of this research was the development of a Great Dane 28-foot control trailer model within TruckSim and Simulink. While many of the physical parameters for this trailer were unknown, engineering estimates led to a reasonably accurate model. The model was first validated against experimental data for a SIS maneuver to ensure that the estimated parameters were reasonable. After the static response looked to be acceptable, the dynamic runs were to be conducted. Before these runs could commence, the current ESC Simulink model was modified to work for a trailer with only one axle. At the conclusion of these RSM runs, it was found that the Volvo tractor-Great Dane trailer experienced rollover at 27 mph with the ESC OFF and at 32 mph with the ESC ON. These results showed some deviation from the experimental results, but it is important to remember that the current ESC model only included a roll stability component and no yaw stability control. Another important shortcoming of the Great Dane model was that many of the physical parameters were estimated.

6.2 RECOMMENDATIONS

The primary recommendation from this research is to develop a yaw stability control model. Now that the current ESC model (which includes only RSC) has been validated, the inclusion of the YSC component will yield a complete ESC model. Before some of the braking changes were made to the Simulink model, the tractor-trailer combination would experience a jackknife condition rather than rolling over for the ramp
steer maneuver. Adding a yaw stability component would help to avoid any jackknifing and allow for more controlled and powerful braking.

Another important recommendation is to update the tire model used for the trailer models. Currently, these tires are modeled after the steer axle tires which are not an accurate portrayal of the physical system. Incorporating more accurate trailer tire measurements into the model would affect both the steady-state gains of the system and the response of the vehicle during a dynamic maneuver like the RSM. It is also known that the rollover speed predicted by the model is lower than the experimental rollover speed for both Volvo-Fruehauf and the Volvo-Great Dane models. Between the addition of YSC control and a better tire model, the simulation would become more representative of the physical system.

A third recommendation is to perform a more complete parameterization of the 28-foot control trailer since many of the physical parameters were estimated for this model. Parameters of interest include mass, inertia, CG location, and suspension characteristics. Implementing these parameters would result in more accurate simulations.

One final recommendation is to perform additional measurements on the actual trailer twist which occurs during a dynamic maneuver. It is known that varying levels of trailer twist are observed depending on the trailer type and loading configuration. By having experimental chassis twist measurements available, this knowledge could be used to better evaluate and modify the torsional stiffness model.
REFERENCES


APPENDIX A

ADDITIONAL SIMULATION RESULTS
Figure A.1: Effect of Torsional Stiffness on Speed and Steering Angle vs. Time for a 45 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination

Figure A.2: Effect of Torsional Stiffness on Chassis Twist vs. Time for a 45 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination
Figure A.3: Effect of Torsional Stiffness on Roll Angle vs. Lateral Acceleration for a 45 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination

Figure A.4: Effect of Torsional Stiffness on Roll Angle vs. Yaw Rate for a 45 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination
Figure A.5: Effect of Torsional Stiffness on Roll Angle vs. Steering Angle for a 45 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination

Figure A.6: Effect of Torsional Stiffness on Lateral Acceleration vs. Steering Angle for a 45 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination
Figure A.7: Effect of Torsional Stiffness on Vertical Wheel Force for a 45 mph SIS on a Low CG Volvo Tractor-Fruehauf Box Trailer Combination