MODELING, SIMULATION, AND ACTIVE CONTROL OF TRACTOR-SEMITRAILER COMBINATIONS

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submitted by **SINA ALAMDARI MILANI** in partial fulfillment of the requirements for the degree of **Master of Science in Mechanical Engineering Department, Middle East Technical University** by,

Prof. Dr. Gülbin Dural Ünver ______________
Dean, Graduate School of **Natural and Applied Sciences**

Prof. Dr. Tuna Balkan ______________
Head of Department, **Mechanical Engineering**

Prof. Dr. Y. Samim Ünlüsoy ______________
Supervisor, **Mechanical Engineering Dept., METU**

Examining Committee Members:

Prof. Dr. Metin Akkök ______________
Mechanical Engineering Dept., METU

Prof. Dr. Y. Samim Ünlüsoy ______________
Mechanical Engineering Dept., METU

Prof. Dr. Mehmet Çalışkan ______________
Mechanical Engineering Dept., METU

Prof. Dr. Tuna Balkan ______________
Mechanical Engineering Dept., METU

Asst. Prof. Dr. Kutluk Bilge Arıkan ______________
Mechatronics Engineering Dept., Atılım University

Date: 24.12.2015

I hereby declare that all information in this document has been obtained and presented in accordance with the academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

Name, Last name: SINA ALAMDARI MILANI

Signature:

ABSTRACT

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Alamdari Milani, Sina

M.S., Department of Mechanical Engineering

Supervisor: Prof. Dr. Y. Samim Ünlüsoy

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Articulated heavy vehicles (AHVs) are among the most common means of road transportation. They present, however, some specific performance limitations and safety risks due to their rather special dynamic characteristics. AHVs have maneuverability problems at low speeds as manifested by their Path Following Off-Tracking (PFOT). Further, AHVs are most likely to lose their stability at high speeds in three basic manners: trailer swing, jackknifing, and roll over which are the most common causes of many severe accidents. Such undesired behavior make the AHVs less stable and more dangerous vehicles compared to single unit vehicles.

In this study, the potential of Active Steering Control (ASC) of the semitrailer in improvement of the maneuverability and stability of tractor-semitrailer combinations is investigated. Existing AHV dynamic handling simulation models in the literature are studied and a model suitable for the aims of this study is presented in detail and implemented in the simulation environment (MATLAB). The vehicle handling model is then validated using a commercial software. The controller for the ASC is determined through the use of the Linear Quadratic Regulator (LQR) optimal state-feedback control which aims to minimize the low-speed PFOT as well as the high-speed roll motion and the Rearward Amplification (RA) of the semitrailer's lateral acceleration. The weighting factor selection for the control system is performed by means of Quantum Particle Swarm Optimization (QPSO) technique. Basic ASC along with two additional strategies are applied to the vehicle model and the results are compared to the baseline vehicle without ASC. The results from simulations show that the combination with ASC exhibits desirable improvements compared to the baseline vehicle.

Keywords: Tractor-Semitrailer, Active Steering, Optimal Control, Optimization, Vehicle Stability

ÖZ

ÇEKİCİ-YARITREYLER KOMBİNASYONLARININ MODELLENMESİ, SİMÜLASYONU VE AKTİF KONTROLÜ

Alamdari Milani, Sina

Yüksek Lisans, Makina Mühendisliği Bölümü

Tez Yöneticisi: Prof. Dr. Y. Samim Ünlüsoy

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Çekici-yarıtreyler (ÇYT) kombinasyonları karayolu taşımacılığında en yaygın olarak kullanılan araçlardır. Bu araçların dinamik davranışları, performanslarını kısıtlayan ve güvenlik riskleri oluşturan özelliklere sahiptir. ÇYT ler düşük hızlarda yörünge kayması ile ifade edilen kısıtlı manevra yeteneğine sahiptir. Diğer taraftan, yüksek hızlarda ÇYT ler ciddi kazalara neden olabilen üç ayrı şekilde denge kaybına uğrayabilir: treyler dönmesi, çekici dönmesi, devrilme. Bu istenmeyen özellikler ÇYT leri güvenlik açısından tek üniteden oluşan araçlara göre daha riskli hale getirmektedir.

Bu çalışmada, yarıtreyler tekerleklerine uygulanan aktif yönlendirme kontrolünün (AYC) ÇYT lerin manevra yeteneklerini ve dengesini iyileştirmekteki potansiyeli araştırılmıştır. Bu amaçla, literatürde mevcut ÇYT dinamik sürüş simulasyonu modelleri incelenmiş ve çalışmanın amaç ve kapsamına uygun bir model tüm ayrıntıları ile sunularak MATLAB ortamında uygulanmıştır. Model daha sonra ticari bir yazılım kullanılarak gerçekleştirilen simülasyonlarla doğrulanmıştır. Çalışmada "Doğrusal Quadratik Regülatör" (LQR) optimum durum geribeslemesi ile düşük hızlardaki yörünge kaymasının en aza indirilmesi amaçlanmıştır. Aynı zamanda, yüksek hızlı seyir koşullarında aracın yalpa hareketinin ve yarı treyler

yanal ivmesindeki artış ta mümkün olduğu kadar azaltılmaya çalışılmıştır. Kontrol sisteminde kullanılan ağırlık katsayıları, "Kuantum Parçacık Sürüsü Optimizasyonu" tekniği ile elde edilmiştir. Temel AYC ile iki farklı kontrol stratejisi araç modellerine uyguanmış ve elde edilen sonuçlar AYC uygulanmayan referans modelin dinamik davranışı ile karşılaştırılmıştır. Simülasyon sonuçları AYC uygulamasının ÇYT kombinasyonunda olumlu gelişmeler sağladığını göstermektedir.

Anahtar kelimeler: Çekici-Yarıtreyler, Aktif Yönlendirme, Optimal Kontrol, Optimizasyon, Araç stabilitesi

To My Beloved Family

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SYMBOLS:

Quantum Particle Swarm Optimization:

M	Population size
i	Index of the particle
D	Number of dimensions
f	Fitness value (the smaller, the better)
G	A parameter constrained to be greater than $\ln\sqrt{2}$
$\bar{x}_i$	Position vector for the i^{th} particle
Φ_1	Personal cognitive factors (positive random)
Φ_2	Social cognitive factors (positive random)

Vehicle Modeling & Control:

m_{is}	Sprung mass of the i^{th} vehicle unit
m_{1uf}	Tractor's front unsprung mass
m_{1ur}	Tractor's rear unsprung mass
m_{2u}	Semitrailer's unsprung mass
h_{is}	Sprung mass CG height of the i^{th} vehicle unit
h_{ir}	Roll center height of the i^{th} vehicle unit
h_c	Articulation point height
l_{1c}	Distance between tractor CG to 5^{th} wheel
l_{2c}	Distance between semitrailer CG to 5^{th} wheel
a	Distance between tractor CG to front axle
b	Distance between tractor CG to rear axle
d	Distance between semitrailer CG to its axle
I_{izz}	Total yaw moment of inertia of the i^{th} vehicle unit
I_{ixz}	Total yaw-roll product of inertia of the i^{th} vehicle unit
$I_{ix'x'}$	Sprung mass roll moment of inertia of the i^{th} vehicle unit

$I_{ix'z'}$	Sprung mass yaw-roll product of inertia of the i^{th} vehicle unit
K_{12}	5^{th} wheel coupling roll stiffness
K_{1f}^{*}	Tractor's front axle roll stiffness
K_{1r}^{*}	Tractor's rear axle roll stiffness
K_{2}^{*}	Semitrailer's axle roll stiffness
C_{1f}	Tractor's front axle roll damping
C_{1r}	Tractor's rear axle roll damping
C_{2}	Semitrailer's axle roll damping
C_{t1f}	Tractor's front axle cornering stiffness
C_{t1r}	Tractor's rear axle cornering stiffness
C_{t2}	Semitrailer's axle cornering stiffness
ϕ_i	Roll angle of the i^{th} vehicle unit
β_i	Side-slip angle of the i^{th} vehicle unit
ψ_i	Yaw angle of the i^{th} vehicle unit
Y_{β_i}	Derivative of the tire cornering force with respect to side-slip angle for the i^{th} vehicle unit
$Y_{\dot{\psi}_i}$	Derivative of the tire cornering force with respect to yaw rate for the i^{th} vehicle unit
Y_{δ_i}	Derivative of the tire cornering force with respect to steering angle for the i^{th} vehicle unit
N_{β_i}	Derivative of the moment of tire cornering force with respect to side-slip angle for the i^{th} vehicle unit
$N_{\dot{\psi}_i}$	Derivative of the moment of tire cornering force with respect to yaw rate for the i^{th} vehicle unit
N_{δ_i}	Derivative of the moment of tire cornering force with respect to steering angle for the i^{th} vehicle unit
u_i	Forward speed of the i^{th} vehicle unit
V_{yi}	Lateral velocity of the i^{th} vehicle unit

α_{1f}	Tractor's front tire slip angle
α_{1r}	Tractor's rear tire slip angle
α_2	Semitrailer's tire slip angle
a_{yi}	Lateral acceleration of the i^{th} vehicle unit
δ_i	Steering angle of the i^{th} vehicle unit
F_{y1f}	Tractor's front tire cornering force
F_{y1r}	Tractor's rear tire cornering force
F_{y2}	Semitrailer's tire cornering force
F_{cy}	Lateral between vehicle units at the 5^{th} wheel
T_{AR2}	Active anti-roll bar torque
X_i	Longitudinal coordinate of the i^{th} vehicle unit
Y_i	Lateral coordinate of the i^{th} vehicle unit

ABBREVIATIONS:

ASC	Active Steering Control
ARC	Active Roll Control
DB	Differential Braking
FLC	Fuzzy Logic Control
LQR	Linear Quadratic Regulator
RA	Rearward Amplification
PFOT	Path Following Off-Tracking
HSTO	High-Speed Transient Off-tracking
AHV	Articulated Heavy Vehicle
DOF	Degree of Freedom
PBS	Performance Based Standards
GUI	Graphical User Interface
SPW	Swept Path Width
ECU	Electronic Control Unit
[Q]PSO	[Quantum] Particle Swarm Optimization

CHAPTER 1

INTRODUCTION

1.1. Motivation

Articulated heavy vehicles (AHVs) are widely used in transport of goods mainly because of their desirable economic benefits in road transport; they are cost effective means of road transport in both labor requirements and fuel consumption compared to single unit heavy commercial vehicles. They also reduce the greenhouse emissions due to their large load carrying capacity [1]. AHVs are known as commercial vehicles with the primary purpose of goods and materials transport in the most efficient manner both logistically and economically. Usually, in the commercial sector, the cost-effectiveness of the trucks are directly related to the size of the vehicle and the larger vehicles with higher load carrying capacities are treated as better vehicles for road transport. The main cost elements are usually the labor (driver) and fuel expenses which could be dramatically reduced by use of AHVs [2].

On the other hand, there are some issues associated with AHVs that make them different from single-unit vehicles. They impose some highway safety concerns due to their large sizes, heavy weights, complex configurations, high centers of gravity and inner forces acting on their articulation points [3]. Between the years 1993 to 1998 about 35,000 people were killed in road accidents in the United States among which about 10% were caused by unstable motions of AHVs [1],

[4]. In Canada, about 2,700 people were killed during the year 2004 where the articulated vehicles were mostly involved in the accidents [5].

Poor directional control and the large length of AHVs are the main causes of some undesired characteristics that may lead to unstable motions and other dangerous behaviors. Besides such traffic safety concerns associated with AHVs, there is also a potential of excessive tire and road wear by use of AHVs. Modern AHVs are usually equipped with multiple-axle groups in order to reduce the vertical tire load on each wheel; this specific configuration which is an inevitable characteristic of large AHVs results in large tire slip due to its geometry causing the tires to scrub against the road surface and results in both tire and road wear [6]. The details of unstable motions and undesired dynamic behavior of AHVs are discussed in the following sections.

Such concerns have made AHVs more of concentration for the designers and researchers; since the year 1985, there has been a huge progress in the analyses and discussions about AHVs with the advent of mathematical models and simulation tools, experimental findings and measurement techniques and the dynamic behavior of trucks and truck combinations have become more accurately studied [2].

The importance of the issue in both business and safety considerations has caused the development of standards related to AHVs and their motions. Australian's National Road Transport Commission (NRTC) has developed a comprehensive set of performance measures for heavy vehicles which is known as Performance Based Standards (PBS) project. PBS have been developed since 1985 with the purpose of providing evaluation criteria for newly developed heavy commercial vehicles for increased vehicle safety and also the proper pavement and bridge loading aspects [2]. In addition to that, there are two other measures that are applicable to tractor-semitrailers operating in the UK [7].

In order to overcome the problems of AHVs, improve their dynamic behavior and minimize the risk of accidents and damages to roads infrastructure, there have been lots of efforts made during the last few decades. One of the most efficient ideas is the implementation of trailer steering systems for improvement of both high and low speed dynamics of AHVs. There are also other safety control systems which are implemented on AHVs that are going to be discussed in the following chapters. There are lots of considerations for use of such systems and there are many advantages and disadvantages associated with each of them. This study is concentrated on dynamic analysis of tractor-semitrailer combination as one of the most popular types of AHVs and the implementation of Active Steering Control (ASC) of semitrailers on such combinations in order to improve the dynamic behavior of the vehicle. Active safety systems have been developed thanks to the advent of electronic devices and equipment and such an application of the technology would be greatly beneficial for both the traffic and highway safety and the transportation sector.

1.2. Articulated Heavy Vehicles (AHVs)

AHVs consist of individual vehicle units which are connected to each other at the articulations point[s] by means of mechanical elements such as hitches, dollies and fifth wheels [2], [5]. Extremely long single unit vehicles with very large wheelbases are poorly maneuverable especially in tight corners; articulated vehicles, on the other hand, consist of several single unit vehicles with shorter wheelbases coupled together at articulation joints. This combination allows the articulated vehicle to be much more maneuverable compared to the single unit long wheelbase vehicle. There are two major types of trailer units: full trailers and semitrailers. Full trailers are supported by their own running gear at the front and rear of the trailer and are similar to single unit independent vehicles but with no power unit. Semitrailers, on the other hand, only possess a rear axle group and the vertical load at the front end of the semitrailer is fully supported by the tractor unit. However, there are some devices called as “converter dolly” which may be connected to the front end of the semitrailer and convert it to a full trailer unit. Nevertheless, the converter dolly mechanism requires a small portion of semitrailer’s front end vertical load to be carried by the tractor unit as well [2].

The most common type of AHVs is tractor-semitrailer combination. In tractor semi-trailers the power unit is separated from the cargo section allowing the cost effective use of the vehicle. In other words, the tractor unit is able to join many different semitrailers which are designated for specific loadings. The semitrailers are usually connected to the tractor unit by means of a mechanical coupling called the fifth wheel. Fifth wheel is located on the tractor chassis and mates the king pin of the semitrailer unit. The fifth wheel allows yaw, pitch and small amount of roll rotations of the semitrailer with respect to the tractor with probable compliances.

Truck-full trailer combinations are also relatively popular in some countries. In such vehicles, the full trailer is usually attached to the leading unit by means of a hitch and a drawbar eye which is connected to the front axle group of the full trailer. In other words, the full trailer is similar to an independent single unit vehicle with steerable front axle which is towed and steered directly by the leading truck unit. The hitch allows yaw, roll and pitch rotations of the vehicle units with respect to each other [2], [8].

There is also a third and less common type of trailers which is known as center-axle trailer. The center-axle trailer only possesses on set of axles which are located slightly aft the nominal Center of Gravity (CG). This configuration requires very small vertical load carriage by the leading unit. Center-axle trailers are mainly used in car-caravan combinations [2], [8], [9].

Usually, tractor-semitrailer combinations are preferred to truck-full trailer combinations because of being easier to load/unload, better vehicle stability and being more easily handled especially in reversing. Common AHV units and hitches used in combinations are shown in Figure 1.1.

There are also other types of articulated vehicles which are known as multiple-trailer articulated vehicles which consist of more than one trailer unit. Such vehicles are less popular due to their difficult handling characteristics and increased level of instabilities compared to double unit articulated vehicles [5]. This study is concentrated only on the most common type of conventional AHVs which is the tractor-semitrailer combination.

While the presence of articulation joint helps to the low speed maneuverability of the vehicle, it might become the main source of some vehicle instabilities such as trailer swing and jackknifing due to the extra yawing degree of freedom added to the vehicle compared to single unit ones [6].

Trailer swing happens when the trailer unit starts to oscillate with small disturbances such as side winds; it also happens when the trailer wheels are locked up due to hard braking. Under this condition, the trailer moves inwards and/or outwards of the path and might become a source of an accident [1], [8].

Jackknifing is defined as the excessive angular motion between the two vehicle units which happens as a result of large slip angles generated at tractor's rear axle in conditions like hard braking causing tractor rear wheels to lock [1], [8].

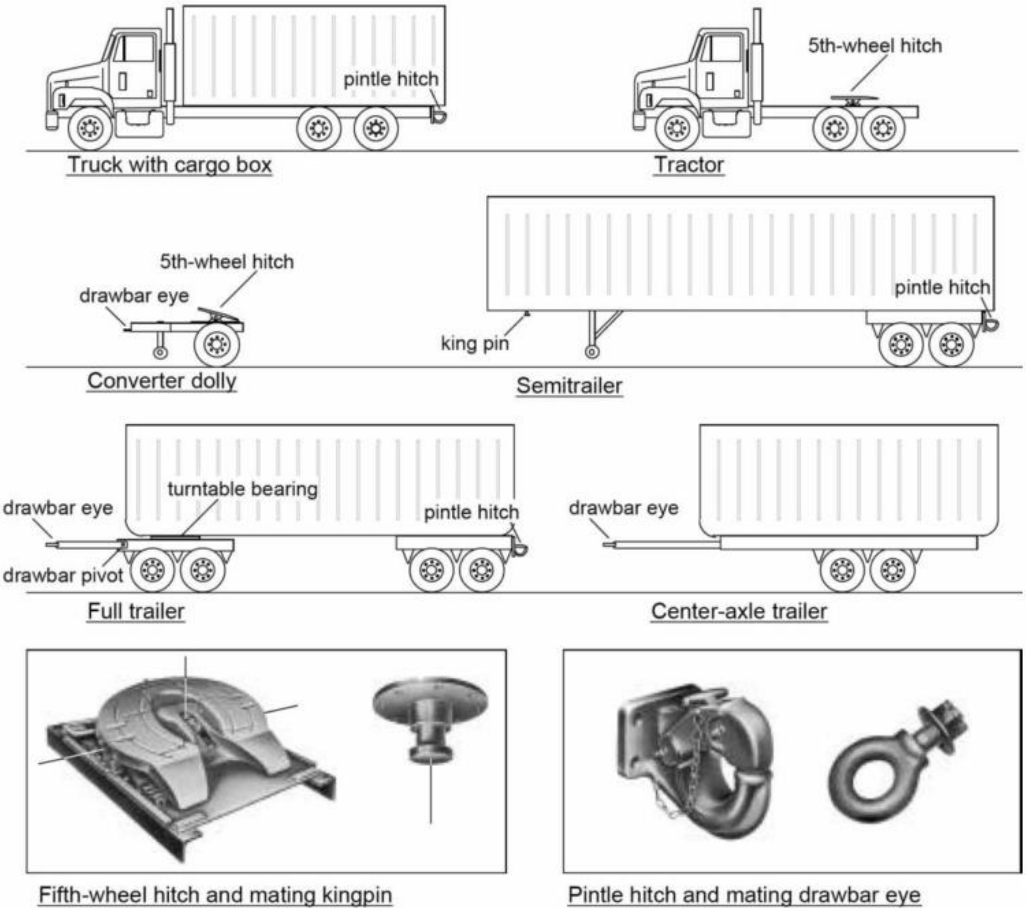

Figure 1.1 - Common Vehicle Units and Hitches [2]

Visualized description of trailer swing and jackknifing are depicted in Figure 1.2.

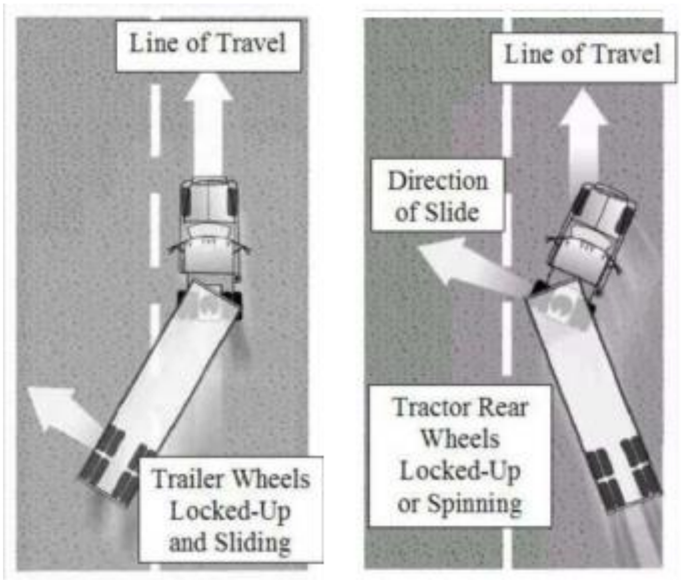

Figure 1.2 - Trailer Swing (Left), Tractor Jackknife (Right) [8]

So far, we have introduced two of the main instabilities associated with AHVs which were known as trailer swing and jackknifing. These instabilities are related to yawing motion and are known as yaw instabilities. There is also another significant instability linked to AHVs which is the rollover. Rollover happens when the lateral load transfer of one of the units exceeds the rollover threshold which is mainly caused by large lateral accelerations of the vehicle units. Usually the trailer unit is under more risk of rollover because of very large roll moments of inertia due to large height loading of the cargo and the huge amount of cargo weights. There is also a phenomenon known as Rearward Amplification (RA) which refers to the fact that at high speeds, the lateral acceleration of the trailer unit may be higher than that of the tractor unit in steady-state and transient maneuvers. RA is defined as the peak lateral acceleration of the trailer unit to that of the tractor unit, measured at the CG points during a single lane change maneuver [1], [10]. The higher RA ratio causes higher risk of trailer rollover. Studies showed that the RA can be large enough to cause the towed unit to rollover in severe driving conditions and maneuvers [11]. Generation of semitrailer's lateral tire forces happens with a time delay after the driver's steering action which causes the towed unit to experience large yawing motions and consequently, significant tracking error which is going to be referred to in the following sections. As a result of such large yaw motions, the tire slip angles are generated at the semitrailer axles causing the generation of lateral tire forces which try to stabilize the vehicle and set the towed unit back into its equilibrium position and regulate the yaw motion [12].

Above mentioned instabilities correspond to high speed maneuvers of AHVs; on the other hand, AHVs exhibit poor maneuverability at low speeds. The undesired behavior of the vehicle at low speed results from the very long wheelbase of the vehicle so that the trailer's rear end does not follow the tractor's front end path, in other words, the trailer tends to cut the corners at low speeds. The radial offset between the trajectory of tractor's front axle and that of the rearmost trailer's rear axle is called Tracking Error or Path Following Off-Tracking (PFOT) [1], [5]. The amount of this off-tracking, to the first order, is proportional to the length of each unit's wheelbase [2]. Visual definition of the PFOT is shown in Figure 1.3.

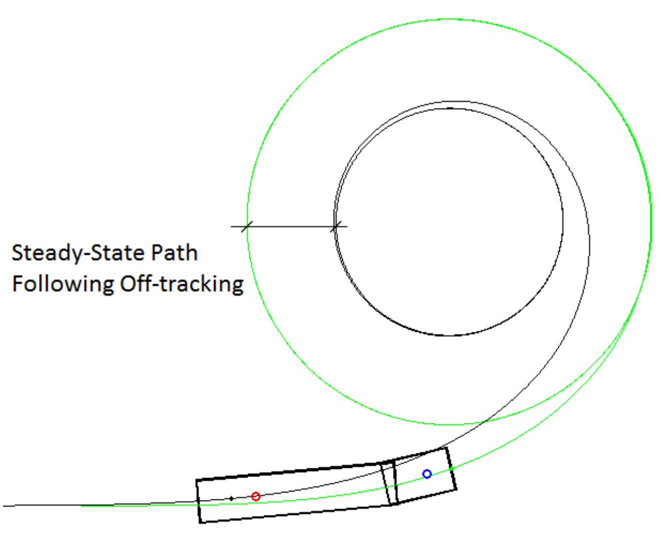

Figure 1.3 - PFOT of an AHV during a 360-degree turn

This low speed tracking error imposes a large risk of accident to the pedestrians and surrounding traffic as well as other urban and suburban properties. PFOT also makes it very difficult for the driver to turn into tight corners especially in cities and small towns. In addition, the PFOT causes a lot of tire and road surface wear in time. In some cases, very large weights and dimensions of the semitrailers require multiple-axle groups in order to reduce the vertical individual tire load. Such a geometry of the vehicle dictates the semitrailer axle groups to encounter large slip angles during low-speed tight turns which generates large tire forces and causes tire and road surface wear which is known as "scrubbing of the tires against the road surface" [6].

There is another type of off-tracking which corresponds to high-speed maneuvers. At high speeds, the side-slip of the semitrailer, and consequently the articulation angle, change sign and the semitrailer tends to go outwards during the maneuver due to its large inertia. This type of tracking error is usually known as High-Speed Transient Off-tracking (HSTO) [13], [14]. Since the high-speed maneuvers are usually obstacle avoidance or lane change maneuvers, HSTO is also an undesirable behavior of AHVs as it can degrade the high-speed maneuverability of the vehicle and cause serious damages. As it is going to be discussed in the following chapters, the HSTO is closely related to the RA ratio and they can be managed together.

The sign change of the semitrailer's side-slip between low and high-speed maneuvers makes the general dynamic behavior of the vehicle dependent to vehicle's forward speed. At low speeds, the semitrailer would cut the corners, while at high speeds the semitrailer tends to go away from the tractor unit towards out of the curve. In other words, at low-speed, the behavior of the vehicle, i.e. the generation of tire slip angles, is mostly dependent to vehicle's geometry; but at high-speed the inertia characteristic of the vehicle, as well as the magnitude of the lateral force between the units, is significant enough to define the vehicle dynamic response to a steering input; the tire slip angle at high-speed are generated after the tractor unit changes its direction while the semitrailer still tends to remain on its previous direction of motion. This is because the only interaction between the two vehicle units is caused by the forces at hitch point (fifth wheel); hence, the generation of semitrailer tire slip angles happens slightly after that of the tractor unit.

Up to this point, it seems that in order to compensate for such undesired motions of the semitrailer, there would be a need to design two different control strategies for high and low-speed operations separately. In the next sections we are going to discuss the different control approaches that have been conducted in dealing with AHVs.

As mentioned above, the low and high-speed dynamics of the AHVs are quite different in some aspects; for this reason, the developed standards must cover both high and low-speed acceptable characteristics of the AHVs. For instance, the Australian PBS covers both low and high-speed operations of the AHVs [1].

In order to improve both low and high-speed dynamics of the AHVs, several considerations have been made through years such as parameter optimizations and use of active and passive control systems including: semitrailer steering systems, differential braking of the semitrailers, roll control systems, using variable damping for the fifth wheels, etc. which are going to be covered to some extent in the literature review section. Among such control systems, the semitrailer steering control seems to make the most significant improvements. As we can see, nowadays, many AHVs including city buses and construction vehicles are utilizing this technology either in active or passive form. Although some studies such as [15] are concerned with control of both units simultaneously in order to provide easier steering conditions for the driver, for the sake of simplicity and better analysis quality, the main concentration of this study is to investigate the effects of Active Steering Control (ASC) for the semitrailer on tractor-semitrailer combinations; hence, in the next section, an introduction to common trailer steering systems is given to clarify the content.

1.3. Semitrailer Steering Systems

Semitrailer steering systems are designed and implemented for two main reasons: improving maneuverability and tire and road damage at low speeds and to increase vehicle stability, response and handling at high speeds [6].

Trailer steering systems fall into two main groups: passive and active steering systems. Although the systems are somehow similar for full-trailers and semitrailers, the concentration of this study will be on the semitrailer steering systems when we discuss the details.

As mentioned in the previous section, the high speed and low-speed characteristics of the AHVs are quite different requiring almost independent control algorithms for regulation of motion. According to Fancher and Winkler [2], attempts to improve low-speed performance of the AHVs are likely to adversely affect the high-speed performance and vice versa. As a result, the designers need to apply different control methods for high and low-speed maneuvers. The first step is to decide on the mechanism and the algorithm by which the semitrailer wheels are going to be steered.

1.3.1. Passive Steering Systems

Passive steering mechanisms have been developed in earlier stages of the researches conducted on the stability and maneuverability of AHVs. These systems are intended to steer each axle of the semitrailer based on a simple geometrical relationship or a force balance. The most common passive steering systems are self-steering axle, command steering system, and pivotal bogie system [1], [7].

1.3.1.1. Self-Steering

Self-steering mechanism is based on the moment balance about the kingpins. A pre-designed Break Point (BP) is defined which corresponds to the amount of above moment which the semitrailer axle is able to steer with very small resistance. When the moments caused by tire cornering forces reach the break point, the self-steering axle starts to steer rapidly with very small increase in the lateral tire force. Overall performance of the mechanism leads to decreased lateral tire forces and tire and road surface damage as well as providing easier turning ability and decreased PFOT of the towed unit. The characteristic of the system is depicted in Figure 1.4 [2]. As shown in the figure, the amount of axle steering depends on the amount of moment (lateral tire forces) about its kingpin.

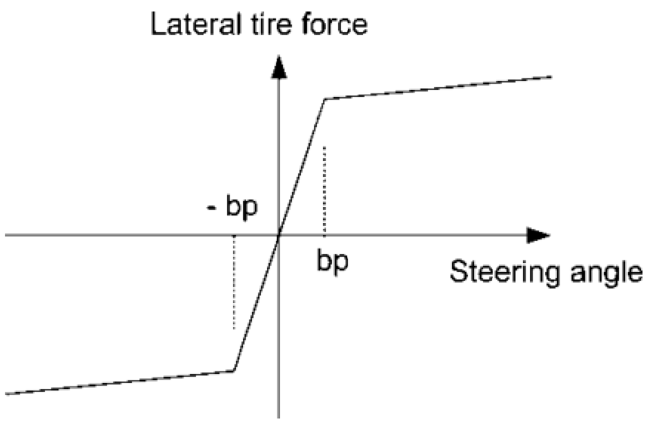

Figure 1.4 - Self-steering Characteristics [2]

Self-steering system is known as the most popular passive steering system because of the simplicity of the system and less expensive equipment [7].

The mechanism of the self-steering axle is similar to a conventional steering axle, but the steering action is not controlled by a steering box; instead, the whole system is located with positive mechanical trail making the steering system more stable. The trailing arm is equipped with springs and dampers in order to provide required stiffness for small steering angles and to control the break point at which the steering axle starts to experience large steering angles with larger compliance. Self-steering system is equipped with proper mechanisms to lock the axle for reverse motion of the vehicle [7].

Self-steering axle reduces the effective wheelbase of the vehicle in low-speed turning maneuvers improving maneuverability and tracking ability by reducing the lateral tire forces. Tire and road wear amount is also decreased as another byproduct of the system [7].

1.3.1.2. Command Steering

In command steering systems, the amount of steering depends on the articulation angle between two vehicle units. The relationship is based on the geometry of the vehicle at low-speed turns (i.e. speeds very close to zero). Objective of this geometrical relationship is to have zero slip angles at semitrailer rear axles. This will eliminate lateral tire forces under ideal conditions [2].

In typical command steering systems the two rear axles of the conventional tri-axle semitrailer are replaced with steerable axles which are steered according to the articulation angle. The articulation angle can be measured by mechanical means, using electronic sensors or other measurement methods and the steering command is then transferred to the axles by means of either electronic devices or mechanical and hydraulic linkages [2], [7].

Command steering system, similar to the self-steering mechanism, reduces the effective wheelbase of the vehicle and eliminates lateral tire forces in rear two axles at nearly zero speeds. Reduction of the effective wheelbase is up to that of a semitrailer unit with fixed single axle [7].

A schematic view of the command steer system and corresponding effective wheelbase are given in Figure 1.5.

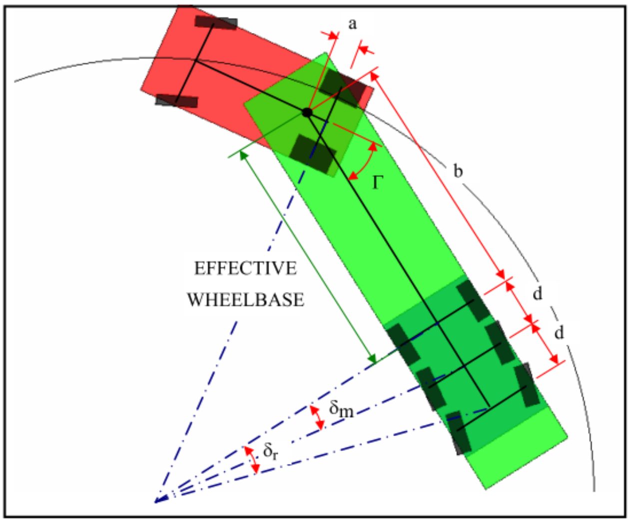

Figure 1.5 - Effective Wheelbase of a Vehicle Equipped with Command Steering System [7]

1.3.1.3. Pivotal Bogie System

Pivotal bogie assembly usually consists of a set of three axles (tri-axle) located in a housing (bogie) which itself is able to rotate freely about the semitrailer chassis by means of a ball race connection. The bogie includes one fixed front axle and two steerable rear axles which can be steered with respect to the bogie assembly. Amount of steering of these two axles with respect to bogie body depends on the angle between the bogie and the semitrailer chassis [2], [7].

Bogie's rear axles steering increase when the bogie itself rotates more about the semitrailer chassis. This mechanism enables the system to get back to its normal equilibrium, in line with semitrailer chassis, during normal conditions. Thus the pivotal bogie system steers the semitrailer wheels when the cornering forces are large. By use of pivotal bogie assembly, the effective wheelbase decreases up to about half of the distance from fifth wheel to the bogie ball race which is the normal wheelbase of the conventional tri-axle semitrailer [7].

The schematic view of the pivotal bogie assembly and corresponding effective wheelbase are presented in Figure 1.6.

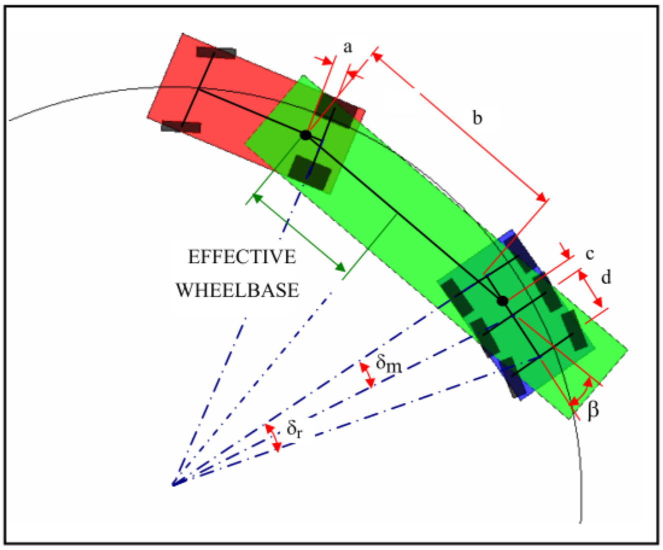

Figure 1.6 - Effective Wheelbase of a Vehicle Equipped with Pivotal Bogie System [7]

1.3.1.4. Comparison between Passive Steering Systems

There have been many attempts to make a comparison between different passive semitrailer steering systems and to find out the most proper system to be used in vehicle structure. Studies have shown that the passive steering systems are helpful for low-speed maneuverability, but they usually exhibit poor high-speed performance and stability [1], [6], [16]. For this reason, the passive semitrailer steering systems need to be compared not only at low-speed maneuverability tests, but also at high-speed maneuvers.

At low-speed maneuverability tests, the pivotal bogie system will gain the most positive points because of the largest minimization in effective wheelbase [2], [7]. The value of steady-state low-speed PFOT is proportional to the square of vehicle's wheelbase to the first order; so, the smaller the effective wheelbase, the smaller the amount of off-tracking [2]. Command steering falls between the other two mechanisms from low-speed performance point of view [7].

Some high-speed RA ratio test maneuvers have reported that the self-steering system performs much better than the other two mechanisms. Command steering takes the second place and the pivotal bogie system shows the worst RA characteristics. HSTO characteristic of the pivotal bogie system is also reported to be quite poor with respect to other two mechanisms. In some test maneuvers, the pivotal bogie system has shown double amount of HSTO compared to that of the conventional vehicle [2].

There is another conventional solution for making reduction in the PFOT and tire-road wear which is not listed here which is called liftable axle group. This system is widely used in many heavy vehicles enabling some of the rear axles of the vehicle to lift off when unnecessary, from loading perspective, or during tight turning maneuvers [17], [18]. Such a system could also reduce the effective wheelbase of the vehicle and consequently improve low-speed maneuverability and tire-road wear, but it is not a very effective and efficient solution to the problem; on the other hand, the increased amount of road surface pressure could be a potential source of damage. For these reasons, we are not going to include the liftable axle group mechanism in this study and we try to concentrate on much more effective mechanisms namely the passive and active steering systems.

Since the passive steering systems reduce the effective amount of wheelbase, the PFOT characteristic of the vehicle is improved, but the smaller effective wheelbase comes with the expense of larger tail-swing at medium to high-speed maneuvers. For pivotal bogie mechanism has the minimum effective wheelbase among other aforementioned passive mechanisms, the medium and high-speed characteristics of this system seem to be less desirable [2], [19].

In another viewpoint, since the passive steering systems mostly depend on the vehicle geometry, they might be effective at low speeds minimizing the tire lateral forces and increasing maneuverability, whilst their stability performance for high-speed maneuvers is reported to be diminished [1], [6], [20].

Although the passive steering systems do not generally exhibit great transient maneuver performances even in low-speed maneuvers [7], the use of such systems would improve the maneuverability of the vehicle and widen the scope of vehicle usage for low-speed case.

Such comparisons and analyses show the weakness of passive semitrailer steering systems in providing both low and high-speed desirable behaviors. For this reason, some studies suggest that the passive steering systems could be used in the vehicles but it would be better, and sometimes necessary, to lock the steering mechanism for medium and high speeds [7], [19].

1.3.2. Active Steering Systems

With the advent of electronic measurement, control and actuation devices, the ability to improve high-speed performance of the AHVs also increased. Active control algorithms have enabled the designers to apply more comprehensive management to the dynamic systems and the deficiencies of the systems can be eliminated more easily. The same strategy can be applied to the semitrailer steering systems in order to make them perform much better regardless of the vehicle forward speed by means of active semitrailer steering systems.

As mentioned previously, the generation of tire lateral forces at high-speed maneuvers occurs after the large yaw motion of the semitrailer unit. Active steering strategy can reduce this time lag significantly and imposes the semitrailer steering in such a way that it does not need to experience excessive yaw motion to regulate itself [12]. Depending on the control algorithm, the response time of the active steering systems may vary.

For an active semitrailer steering system, the performance of the system is mainly dependent to the control algorithm and its objective. The potential control objectives include the minimization of PFOT, RA, HSTO, roll angle, yaw rate, etc. Different control approaches are going to be covered in the chapter 4.

Implementation of active steering systems alleviates the need for complicated mechanical linkages and devices for the steering mechanism. Each of the semitrailer axles could be steered separately or in relation to each other depending on the control algorithm. In most cases they are steered in such a way that the slip angles of the axles will be equal [15] in order to distribute the lateral force and tire wear evenly between the axles.

CHAPTER 2

LITERATURE REVIEW

We have covered an introduction to AHVs, their definitions, types and the concerns about their dynamic behavior. This chapter will discuss the important researches considered as significant contributions to the literature in field of vehicle dynamics and control. At the end of this chapter, the objective of this research according to the findings in literature and the general considerations about the subject is presented.

2.1. Contributions by the Researchers

Many studies have been made on the dynamic behavior of AHVs and the challenges associated with them. There are a number of relatively old references to the subject that were conducted when modern computers and simulation tools did not exist. Those researches were mostly concentrated on the design criteria of the AHVs and their layout, parameters, etc. New researches, on the other hand, are focused mostly on the methods and ideas to make improvements on the AHVs dynamics such as active and passive control systems.

Different dynamic improvement methods such as roll control systems, yaw regulation systems, braking control strategies, stability alert systems, etc. have been studied and developed in order to make improvements in dynamic response of the AHVs and increase their level of safety and stability. In what follows, the most relevant modern researches are presented in summary.

One of the early studies concerning the passive dynamic improvement systems for AHVs is the study by Vallurupalli [21] in which damped articulation and its effect of dynamic behavior of the AHV, especially in yaw oscillation behavior, are the subjects of research. The system does not use any electronic measurement device or actuation. This study can be treated as a preliminary example of the efforts that have been made in the past few decades to compensate for the undesired AHV behaviors.

Dahlberg and Wideberg [22] conducted a principal research on the influence of the longitudinal location of the fifth wheel lateral dynamic behavior of AHVs as a design variable in order to increase stability and handling.

Luijten [8] performed a comparative stability analysis of conventional articulated vehicle combinations. The RA ratio is also measured for different types of combinations as a measure of performance and stability. The effect of using multiple axles on lateral dynamics of different combinations is investigated as well.

In a study by Luo [17] three conventional axle groups, namely typical multiple axle group, liftable axle and self-steering axle are modelled and compared from the low-speed performance perspective.

In another research, Moon et al. [23] proposed a method for control of an all-wheel steering articulated vehicle for low-speed turning maneuvers based on geometrical relationships.

Bortoni et al. [24] designed a Fuzzy Logic Controller (FLC) in order to steer the wheels of a semitrailer with objective of PFOT minimization at low-speed turns. They presented a steering equation and applied the FLC to follow the desired variables. The relation between PFOT and system variables is derived by means of field test results and the equation is then used in the design of control system.

Rangavajhula and Tsao [25] developed an LQR active trailer steering algorithm for a truck-full-trailer combination based on a conventional passive command steering system. The geometrical relation is used to obtain a typical command steering strategy and then an active controller based on LQR technique compensates the deficiencies of the command steering system by taking into account the effect of vehicle forward speed. Selection of LQR weighting factors for this combined system is performed manually. The system showed much reliable performance at high speeds with respect to the older and conventional command steering mechanism.

In the work by Odhams et al. [26] which is followed by Jujnovich and Cebon [6], the path-following ability of the trailer unit was the control objective for all vehicle speeds. They accounted for the path-following ability as the measure of desirable and stable vehicle response. Two points are defined on the vehicle body as “lead point”, on the fifth wheel, and the “follow point” on the center of semitrailer’s rear end. The distance travelled by the lead point and its heading angle is stored in a lookup table memory in order for the path of lead point to be determined. This procedure is done by means of a simple bicycle model of the vehicle. Then the desired heading angle of the semitrailer is generated; a pendulum model of the AHV is then used to calculate the required side-slip of the semitrailer and the corresponding steering angle of the rear wheels. This reference value is then fed into the real trailer along with an additional correcting steering angle generated by a PID controller to compensate for the error between the real and the reference yaw angle of the semitrailer. They also considered additional active steering for the tractor’s drive axle and compared the results of both control layouts. All of these comprehensive studies are developed and implemented on a test vehicle as reported in [27] and [28]. The hardware-software interaction and the practical implications of the proposed active steering systems are discussed in a study by Roebuck et al. [29].

A dual objective control strategy is presented by Cheng and Cebon [20] and implemented on a test vehicle in another study by Cheng et al. [10]. The vehicle is modelled using a 5 DOF linear model. Active semitrailer steering is applied in order to minimize the tracking error at high-speed maneuvers (HSTO) and improve the roll stability of the vehicle as a byproduct of this method by minimizing the RA in the meantime. The method is that a virtual driver is assumed to sit at the rear end of the semitrailer and follow the path of the fifth wheel at high-speed. System states are gathered using measurement devices or predicted using a Kalman filter as an alternative. In a method similar to [6] and [26], a series of information about the location of fifth wheel is generated and stored in a memory. Current trajectory of the semitrailer's rear end is then compared to the previous corresponding position of the fifth wheel at an earlier time and the virtual driver which works based on LQR control steers the semitrailer wheels to compensate for the path deviation. The linear system is discretized in order for the method to be applicable. The vehicle states and the trajectory data are constructed as an augmented system of equations in a discrete form. LQR is tuned in such a way that a specific cost function consisting of path deviation error, lateral acceleration and the control input are minimized. Since the main source for vehicle roll motion is the centrifugal force acting on the vehicle, roll stability control is achieved by minimization of trailer lateral acceleration. It has been seen that there is a compromise between highly improved roll stability and the path deviation consideration; for this reason, the amount of path deviation has become acceptable to some extent and the roll stability is improved further. This is done by adjusting the weighting factors in the cost function to give more priority to one of the control objectives. Finally, the performance of the system is assessed using TruckSim® software. The results have shown improved roll stability (reduction in lateral load transfer ratio) with acceptable path-tracking deviation.

In order to be able to use data required for vehicle control, in case the measurement systems does not fit, the study by Cheng and Cebon [30] presents a Kalman filter to be used in the control system in order to estimate the vehicle states for control purpose.

Wu and Lin [31] considered the truck-full trailer combination with the front axle of the trailer being actively steerable. A 3 DOF linear yaw-plane model is used for vehicle modeling. The steering angle of the trailer is proportional to that of the driver's steering input to the truck. Control objective is to minimize the side-slip of the trailer in steady-state maneuvers in order to follow the leading unit more precisely. The steering ratio is not constant and varies according to vehicle forward speed. They have reported that the vehicle's high-speed performance is improved as well as its stability limit. At low-speed, however, the control strategy does necessarily improve the PFOT as much.

Tabatabaei Oreh et al. [32] used a 14 DOF nonlinear full-car model for simulation of a tractor-semitrailer combination. They took the articulation angle as their performance measure and tried to regulate it by means of a fuzzy-logic controller. The method is based on the objective that the semitrailer's rear end must track the path of fifth wheel similar to [26], but in their approach, the location of the fifth wheel is predicted using Taylor Series Expansion method. Assuming the vehicle speed to be constant, they predicted the location of the fifth wheel as their reference point and the corresponding articulation angle for best tracking condition based on kinematic relationships. Then the fuzzy logic controller applies proper steering angle to the semitrailer wheels in order to follow the desired articulation angle. The rule base of the fuzzy controller is developed based on several simulations and expert knowledge. Two other control objectives are also introduced and compared to the original one and the results have shown their suggested method to be more effective at both low and high-speed cases. The other two methods are to assume steady-state steering and equating the heading angles of the fifth wheel and the follow point for the matter of tracking.

In another study by Tabatabaei Oreh et al. [15] a 14 DOF nonlinear full car model using Dugoff tire model is developed for simulation purpose of a tractor-semitrailer combination. Active semitrailer steering strategy is applied on the vehicle for high-speed lane change and low-speed turning maneuvers. In this research, the rear axle of the tractor unit is also steerable for further improvement. The controller uses a model-based strategy with a feed-forward part and a feedback based on the LQR technique. A linear 3 DOF model is used for the controller design purpose. Desired yaw rate response, lateral velocities of the tractor, and articulation angle are defined to be used as control criteria. Steady-state yaw rate response of a linear model, zero lateral velocity of the tractor and a geometry-based desired articulation angle are considered as the control objectives. Three different feedback designs are developed, implemented and compared for the control system based on optimization of the three variables as discussed above. The selection of LQR weighting factors is done manually for different control algorithms. Finally, the simulations are performed for low and high-speed maneuvers; for high-speed maneuvers all of the three control algorithms are implemented and compared while for low-speed case, only the articulation angle regulation is applied. It is concluded that regulation of the introduced desired articulation angle performs much better than considering the steady-state turning articulation angle as the reference value.

Tabatabaei Oreh et al. [33] also applied a sliding mode control strategy to the AHVs in order to follow the previously derived desired articulation angle. Design of the controller is based on a 3 DOF nonlinear model of the AHV and implemented on a 14 DOF complicated full-car model for simulation purpose.

The complex model is validated using real test data available from other researcher's works.

Islam [1] and Islam et al. [34] studied a truck-full trailer combination with different active trailer steering strategies. 3 Degree of Freedom (DOF) models are used to predict the yaw-plane behavior of the vehicle. For the control purpose, LQR technique is taken into consideration for both high and low-speed cases. Along with LQR parameters, some of the vehicle parameters are optimized for the best possible lateral stability at high-speed and better maneuverability at low-speed. The control objectives correspond to regulation of RA at high-speed and minimization of the PFOT at steady-state low-speed maneuvers. Two types of design loop methods are introduced one of which includes a driver model to follow a prescribed vehicle path. Standard steady-state circle and single lane change maneuvers are simulated for low-speed and high-speed cases respectively during which the PFOT is reduced by 35.16% and the RA ratio by 30.01%.

In another comprehensive study by Islam [5], tractor-semitrailer combination is studied and the similar approach in [1] is implemented with more considerations. The linear model used in the study is validated using TruckSim® and three different control approaches are applied separately and combined which are Active Trailer Steering Control, Anti-Roll Control, and Differential Braking Control. All of the control systems are based on LQR method. Since the multi-trailer vehicles are also included in the study, two reference vehicles are of interest: a tractor-semitrailer and a tractor-double-semitrailer. A driver model based on a PID controller is developed to follow a prescribed path in a closed-loop steering. Since the control system is applied for both low and high-speed maneuvers, it is suggested that the transition between PFOT control and RA control may be done at speed of 40 km/h.

Jujnovich and Cebon [7] compared the most popular existing passive semitrailer steering systems to the date: self-steering, command steering, and pivotal bogie system. A nonlinear yaw-roll model is developed in Simulink with ability to account for lateral load transfer and nonlinear tire model. A proportional controller is introduced as a driver model in order to follow a prescribed path. The three semitrailer steering systems are implemented on the model and six different maneuvers are simulated according to Australian PBS and UK standards. The performances are then measured in both low and high-speed maneuvers and the results are given as presented in the previous chapter of this thesis in the comparison section.

In a study by Kharrazi et al. [13], an active steering strategy is applied on 9 different vehicle combinations. The concentration of the controller design is on moderate to high-speed lane change maneuvers. The controller, utilizing a model-based control algorithm, consists of a feed-forward part as well as a feedback section in order to compensate for un-modelled dynamics, parameter uncertainty and external disturbances. The feed-forward part of the controller is based on a linear vehicle model with negligible roll dynamics and lateral load transfer. Objective of the control is to minimize the yaw rate rearward amplification (the amplification ratio of tractor's yaw rate to that of the towed unit) as well as HSTO minimization as a consequence. Simulations are then performed on a nonlinear two-track vehicle model including roll dynamics and nonlinear Magic Formula tire model in the Simulink environment. Results are interpreted as favorable for high-speed lateral performance of the combinations. The controller is then implemented and tested on a real truck-dolly-semitrailer test vehicle as reported in [9], [12].

Ding et al. [35] applied LQR optimal feedback controller for active steering of multi-trailer AHVs. The design objectives of the system are to minimize the PFOT at low-speed and RA at high-speed maneuvers. The roll motion of the vehicle units is neglected in the modeling. Rest of the study is concentrated on implementation of the designed system on LabVIEW an applying the controller for a double-trailer AHV modelled and simulated using TruckSim® commercial software.

In a comparative research by Ding et al. [36], two control algorithms are presented for active semitrailer steering control based on LQR and FLC. FLC input variables are tractor's lateral velocity, yaw rate and articulation angle; the the LQR is based on a 3 DOF yaw-plane model. The control objective is to minimize the PFOT at low-speed and to reduce the RA at high-speed maneuvers. Controllers are implemented in Simulink and the vehicle model is configured in TruckSim® commercial software. Simulations are performed under the collaboration of the two computer environments and the results extracted by each algorithm are compared. Maneuvers of interest are standard 90-degree intersection turn and single lane change for low and high-speed cases respectively. It is concluded that LQR turns out to be a better performing algorithm compared to FLC, but in expense of higher energy consumption.

In order to put all those control algorithms into practice, many safety considerations must be applied; Odhams et al. [19] investigated the safety and practical challenges of a potential active semitrailer steering system from stability and safety perspectives.

Prem et al. [37] investigated the performance quality of a commercial semitrailer steering system called Trackaxle® on multi-combination heavy vehicles by means of simulations using ADAMS. The system comprises a command steering control combined with bogie mechanism so that it could be more effective with respect to conventional command steering. The performance of the Trackaxle® is assessed based on the PBS framework and it is concluded that the use of the system substantially improves safety and level of damage to road infrastructure. The research is based on a previously published material by Prem and Ramsey [38] which included the evaluation of Trackaxle® by practical applications of the PBS.

In another study, Prem et al. investigated the effectiveness of a developed steerable axle group using ADAMS. The Steerable Wheel System (SWS) consisting of multiple-axle setup is able to reduce low-speed PFOT as well as high-speed RA and HSTO. SWS uses fully active control strategy which can be classified in steer-by-wire systems.

Other than active steering, other control approaches are studied for improvement in lateral behavior of the AHVs. Fancher et al. [11] introduced a Differential Braking (DB) algorithm over a decade ago which enables a yaw moment generation by the different right and left wheels' braking pressure regulation. The yaw moment meant to regulate the plane-motion of the vehicle. For the design of control system, a linear model is developed, but the simulation results are obtained using a nonlinear model based on multi-body dynamic approach.

Nonlinear control method is used in a study by Mobini et al. [39] to optimally distribute the braking force between the wheels of an articulated vehicle in order to improve yaw stability of the AHV in a similar manner to the DB system in a fully nonlinear tire characteristic area. The DB algorithm is investigated in many other references such as the study by Mai et al. [40], and Tianjun and Changfu [41].

As one of the simple and yet practical anti-roll strategies, a roll control system is introduced by Lu et al. [42] which gathers the information from roll rate sensor and the conventional electronic stability control system. Then the roll condition is evaluated and compared to a rollover threshold and proper braking control and engine torque limitation are applied accordingly in order to eliminate the rollover risk.

In a more direct roll control approach, active anti-roll bar mechanisms have been designed in a number of studies [43-51] in order to directly control, using active or semi-active suspensions, the roll motion of heavy vehicles regardless of the number of vehicle units.

More general researches are also made such as the study by Edgar [52] which contributes to the development of regulatory Performance Based Standards (PBS) by developing performance criteria for heavy vehicles in order to increase road safety and the level of eligibility for modern heavy vehicles.

In a very recent work by Dilberoğlu [53], an active semitrailer steering control strategy is introduced by applying some modifications on an existing strategy, and it has been shown that the proposed LQR semitrailer steering is able to improve both low and high-speed performance of the combination compared to other existing methods. The control approach is based on mathematical relationship between the lateral accelerations of the vehicle units and the control objective is to reduce the difference between those values at both low and high-speed cases. The advantage of such a control strategy is that it uses the similar strategy for both low and high-speeds with slight changes in details.

2.2. Research Objectives

In this section, as a conclusion of the literature review, the objectives of the current research are defined as follows:

The first step in the study of vehicle dynamics, regardless of the vehicle type, is to develop, or choose and modify the proper vehicle model according to the research goals. Issues such as the model being linear or nonlinear, degrees of freedom of the system, assumptions in writing the equations of motion, etc. are of great importance for the validity of the vehicle model. Since many models of AHVs are developed and used in the literature, it would be more rational to study the existing models and the objectives they have been used for, rather that developing a completely new model from the beginning; therefore, the first objective of this thesis is to identify the most suitable model among the existing models presented in the literature for this study.

After construction of the vehicle model, it should be implemented in a proper environment. MATLAB is the most commonly used program for implementation of the models which are of reasonable complexity. In this research, MATLAB is used to implement the mathematical model of the vehicle as well. Since the number of degrees of freedom considered for this research is limited, manual MATLAB code generation is preferred over Simulink structure for the sake of getting faster results and easier handling of the code.

After preparation and implementation of the model, as a professional study, a validation of the model is required in order to apply the methods including the control to make sure that the study is verified in the real life. Since the field test facilities are limited and in order to act more time and cost efficiently, many researches are based on the powerful multi-body dynamic software packs which are generally used for commercial purposes. One of the best suited simulation tools, as far as the scope of this work is concerned, is the TruckSim® commercial software which is a special package of the more general CarSim® software. The multi-body dynamic simulator has been found reliable enough to be used instead of real vehicle field testing in the preliminary phases of the design. As a result, the next step is to validate the mathematical model using TruckSim®.

An analysis of the baseline vehicle response to standard inputs is necessary to observe and understand the areas of weakness of the AHVs. As mentioned previously, the main concerns associated with AHVs are PFOT at low-speed, RA and HSTO at high-speed maneuvers. These issues are going to be observed and recalled in the next step.

Selection of a control approach for dynamic improvement of the tractor-semitrailers is going to be another important objective of the current study. As observed in the literature review, some control approaches have been considered so far each of which has its own advantages over other methods. At this stage, a decision making is required to continue the study.

Many control methods are available including fuzzy logic control, optimal control, nonlinear control, model-based control, etc. among which, the most suited strategy needs to be adopted and implemented as the next stage of the study. The objective of the control system should also be defined: what function is going to be minimized as a measure of better control?

Existing control strategies may bring some difficulties in aspects of complexity, inaccuracy and limitation in application. Based on the study of the baseline uncontrolled vehicle, it can be concluded that there is the opportunity to apply a relatively simple control strategy to the vehicle in order to make the implementation more feasible, while precise.

Regarding the existing literature, the first significant objective of this study is to make a more detailed analysis of the vehicle behavior to understand how tracking ability, lateral accelerations and roll motion are correlated. The second objective is to provide a simpler control strategy with respect to the more complicated methods with intensive mathematical calculations and hardware requirements such as the methods of vehicle future position prediction, virtual driver tracking method, nonlinear control, fuzzy control, and model-based control, without sacrificing the accuracy. The third target of this study is to develop the proposed control strategy for a list of vehicle speeds over the whole speed range and to obtain a table of feedback coefficients for this purpose, while the current studies have concentrated on a specific vehicle speed each for the low and high speed operation. Providing a few different variants of the controller is the fourth objective of this research so that it would enable the designer to choose between the proposed control methods according to the application and the vehicle of interest. Finally, the last objective is to apply a relatively recent optimization method for adjustment of weighting factors of a linear quadratic regulator, which has not been previously tried on vehicle dynamics control studies.

In other words, simplicity, generality, and accuracy are the most emphasized points throughout this thesis to fill the void in this area and contribute to the literature in the above aspects which seem to have received less concentration in the existing literature.

At the end, the results are going to be presented as well as the outcome of the study. A brief explanation of the potential future studies is also going to be given as the motivation for other researchers.

CHAPTER 3

VEHICLE MODELING AND SIMULATION

3.1. Modeling

In the following section, a brief look on the existing models in the literature and selection of the best suited model for the application of interest will be given.

3.1.1. Conventional AHV Models

3.1.1.1. Pendulum Model

The simplest available model according to the literature review section is a 1-DOF pendulum model. This model assumes constant forward speed and direction of motion for the vehicle and leaves only one variable free which is the yaw angle of the semitrailer unit with respect to the tractor unit. This model is used for a part of control system design in [6].

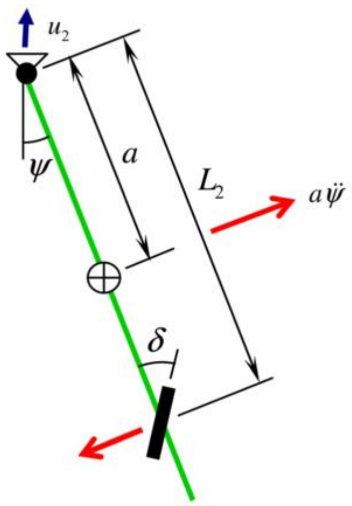

Figure 3.1 - Pendulum Model of the Semitrailer [6]

Because of the highly simplified characteristic of the model and large number of assumptions, this would definitely not be the proper model for the current study; besides, the pendulum model is only used for a single part of the controller design and the other parts are based on more complicated models to complete the whole design for practical applications.

3.1.1.2. Roll-Plane Model

This model only considers the roll motion of each vehicle unit separately by a single (or multiple) DOF model which accounts for the roll motions of the sprung, and sometimes unsprung, mass. The model is usually used for design of anti-roll controllers and the analysis of roll motion of conventional vehicles [48].

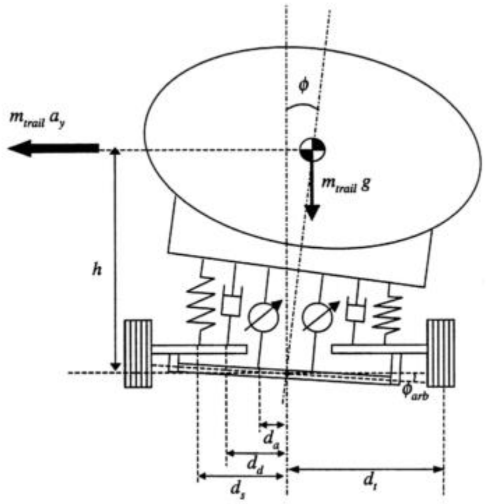

Figure 3.2 - Roll-Plane Model [48]

3.1.1.3. Linear Bicycle Model

The third simple model is the bicycle model of the vehicle which enables the use of linear and simplified equations of motion including linear tire models. Bicycle models are generally used for the analysis of the plane-motion for studies related to yaw regulation, handling, etc. at low lateral accelerations. The model disregards the width of the vehicle enabling the designer to assume each axle as one equivalent wheel instead of considering left and right wheels separately which consequently eliminates the lateral load transfer consideration. For this reason, the accuracy of the model may degrade under severe maneuvers. This issue is referred to as a limiting lateral acceleration below which the use of bicycle model is allowable [16]. 3-DOF linear bicycle models are used in [1], [15].

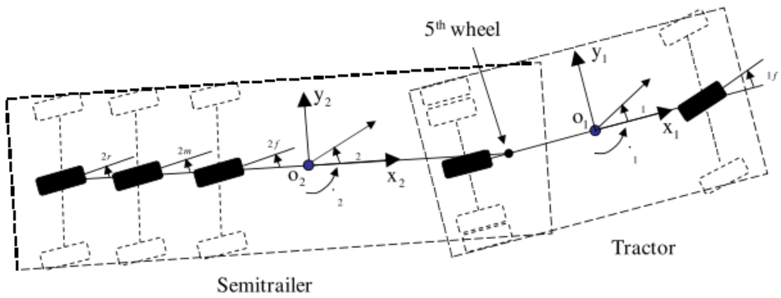

Figure 3.3 - Bicycle Model of Tractor-semitrailer [10]

3.1.1.4. Nonlinear Bicycle Model

Another alternative for the bicycle model is to limit the assumptions and not to consider small angles for the system. This could be done when the vehicle experiences large articulation and/or steering angles while being in the low lateral acceleration region alleviating the need to consider roll motion as well as the lateral load transfer. This model can be treated as an extension to the linear bicycle model. An example of usage of such a model could be found in [6].

3.1.1.5. Bicycle Model Including Roll Motion

The next model in the level of complexity is an extended version of the bicycle model which also includes the roll motion in another separate roll-plane model. This model can be treated as a combination of a conventional bicycle model combined with a roll-plane model which acts independently. This extension allows the researcher to observe the roll motion as well as plane motion of the vehicle, but still is only valid for a limited range of lateral accelerations so that the effect of lateral load transfer could be neglected. This model keeps the ability to use linear equations of motion while adding the roll degree of freedom, increasing the total DOF of the system to 5. The best examples of using such a model could be found in [10] and [54]. Investigation of the frame flexibility can also be done by making small modifications on the model [49].

3.1.1.6. 8-DOF Model

In the study by Chen and Tomizuka [55-57], an 8-DOF nonlinear vehicle model is derived based on Lagrangian mechanics. The derivation of the model is somewhat a complicated task, but ones the model is ready, it could be used for different purposes of vehicle dynamics analyses. The model includes the pitch motions of the units as well as longitudinal velocity change which adds 3 additional degrees of freedom compared to the previous model.

3.1.1.7. 14-DOF Full-Car Model

Another nonlinear modeling approach is followed in [15], [32] which presents a full-car model including the rotational motions of the wheels as additional degrees of freedom. Such a model is used as a simulation basis of the study and is assumed to have a response very close to real vehicle. Such models are also sometimes used to validate simpler models.

3.1.1.8. Multi-Body Software Modeling

The most complicated modeling approach is to use the multi-body dynamic analysis software packs which are design for this purpose. The comprehensive models created by such software are usually so highly accredited that they are assumed to produce the real vehicle response, if modelled appropriately. The most well-known programs are ADAMS, CarSim®/TruckSim® and ArcSim® [3], [58]. Models created by using such software packs are widely used to validate simpler models developed for specific studies.

Figure 3.4 - Example of a TruckSim® Model

The method of tire modeling is an additional consideration; depending on the vehicle performance limits and the maneuvers of interest, the proper tire model must be selected and used to predict the lateral and/or longitudinal forces affecting the vehicle in its ground contacts. If the slip angles are small enough, a linear tire model would be sufficient for prediction of tire cornering forces. If the slip angles are large, other models such as the Magic Formula and Dugoff models could be used for simulations.

3.1.2. Assumptions

In order to develop, implement and make use of an efficient vehicle model, any designer must be aware of the assumptions being made in design and simulation procedure. Any unheeded or over-introduced assumption may lead to unnecessarily complex or invalid modeling which can divert the results from real field tests.

Since in this study the lateral dynamic behavior of the AHVs is of interest, we can get rid of the longitudinal forces acting on the vehicle and assume the vehicle forward speed to be constant, reducing one degree of freedom.

Pitch motions of the vehicle units are also not of interest, so they are assumed to be small enabling us to reduce another degree of freedom of the system.

On the other hand, such a stability and handling analysis eliminates the need for consideration of tire and unsprung mass dynamics.

As the simulation results confirm, the assumption of small angles for high-speed case is totally reasonable while at low-speed, the articulation and steer angles are large.

Vehicle parameters are assumed to be constant, i.e. the loading characteristic of the vehicle does not change during the maneuvers and the cargo is assumed to be fixed within the semitrailer sprung mass. Any other vehicle parameter is also assumed to be unchanged.

The fifth-wheel connection or the articulation point is assumed to allow free yawing of the units with respect to each other and there is no stiffness considered for it.

The sprung masses of both vehicle units are assumed to be rigid in roll motion and there is no flexibility considered for them. In highly severe maneuvers and very large loading of the semitrailer, this assumption may not be appropriate [49]. However, in the scope of this research and according to the simulated maneuvers, the assumption of solid sprung masses is quite rational.

The effect of lateral load transfer is neglected due to small lateral accelerations. As a consequence, the steering and slip angles of each axle, on right and left wheels, are assumed to be equal so that all of the wheels on each axle could be replaced by a single wheel with equivalent cornering stiffness (single-track model).

The final assumption is to consider the semitrailer axle group as a single equivalent axle located in on the middle axle position in order to simplify the model further. By this assumption, the control input for the active steering system would be a single value reducing the number of inputs to the system. As mentioned previously, in case of multi-axle vehicle, the determination of steering angles for other axles is done by equating the slip angles for each of the axles.

3.1.3. Model Selection

Regarding the purpose of this research, we need a model which is both efficient and as simple as possible to predict the behavior of the tractor-semitrailer combination with proper level of complexity.

It would be better to consider low and high-speed cases separately. For low-speeds, the model must be capable of providing yaw-plane information as well as the ability to account for large (larger than 4 degree which cannot be approximated using conventional linearization) articulation and steer angles. On the other hand, the slip angles are small enough at low-speed maneuver of interest and can approximated using conventional linearization methods. Other motions including roll, pitch and bounce motions are not of interest for low-speed turning maneuvers. Lateral load transfer is also negligible at speeds of below 10 km/h as mentioned in our reference low-speed standard maneuvers which are going to be discussed later.

As a result, we may not choose a fully linear model in order not to lose the accuracy for large articulation and steer angles. On the other hand it is much better to opt for a minimal model for the sake of simulation convenience. A single-track yaw-plane model with no small angle approximation, except for the slip angles, with linear simple tire model would be the best possible choice. Among the proposed models, the nonlinear bicycle model meets our needs perfectly. It also enables us to choose any type of tire model which in our case would be linear model. Equations are nonlinear yet simple as presented in Appendix A. It is worth to mention that proper modifications need to be made on the model according to specific needs of this research. The nonlinear bicycle model is referred to as model 1 in the rest of the thesis.

At high-speeds though, requirements are somewhat different; one of the major variables to be studied at high-speed maneuvers is the roll motion. Inclusion of roll degree of freedom requires a more complicated model. On the other hand, the slip angles increase as well as the lateral acceleration requiring more considerations for simulation purpose. As we are going to see in the next sections, the standard maneuver defined by SAE J2179 does not make the vehicle exceed the limiting lateral acceleration of about 0.3g which is designated to validate the use of linear vehicle model [16]. The slip angles also take values smaller than 4 degrees in the scope of this study. On the other hand, the articulation and steer angles are very small such that they could be easily approximated by conventional linearization as referred to in the next sections. Regarding the assumptions given, the linear bicycle model including roll motion would be the best possible choice by being both efficient and simple and there would be no need to work with nonlinear complex models.

The best organized linear bicycle model including the roll motion is proposed by Sampson [54] which enables the consideration of not only single-unit vehicles, but also multiple-articulate vehicles regardless of the number of units. This model is later used in several other AHV studies for both roll control approaches and active trailer steering designs in references such as [1], [5], [20], [49]. As a result, this linear model is going to be considered as the reference model for high-speed simulations and called as model 2.

The dynamic equations of motion, as expressed in Appendix A, are quite understandable and easy to use even when the researcher needs to apply modifications. As far as this study is concerned, the modified equations of motion and the state-space arrangement of the system are also provided in Appendix-A. The vehicle model is presented in Figure 3.5.

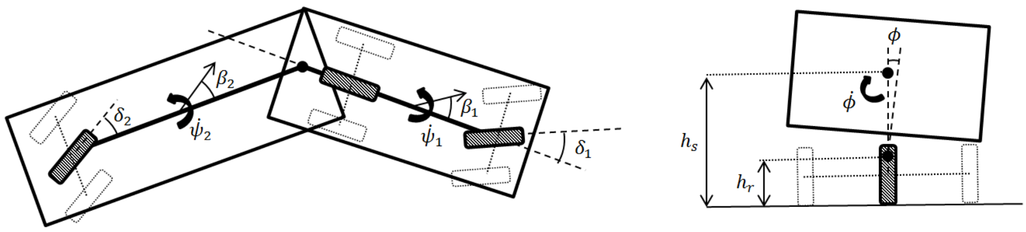

Figure 3.5 - AHV Bicycle Model Including Roll Motion

3.2. Simulation

3.2.1. Simulation Tool

Both of the nonlinear and linear models corresponding to low and high-speed simulations are implemented using MATLAB's ODE solvers as the main simulation tool. A Graphical User Interface (GUI) is also created to ease the simulation by enabling the user to load or save vehicle parameters, view or save simulation results, turn on and off the controller, adjust the control parameters, choose the maneuver characteristic, select from existing simulation models and get a simple animation of the vehicle motion during the maneuver to observe the behavior graphically.

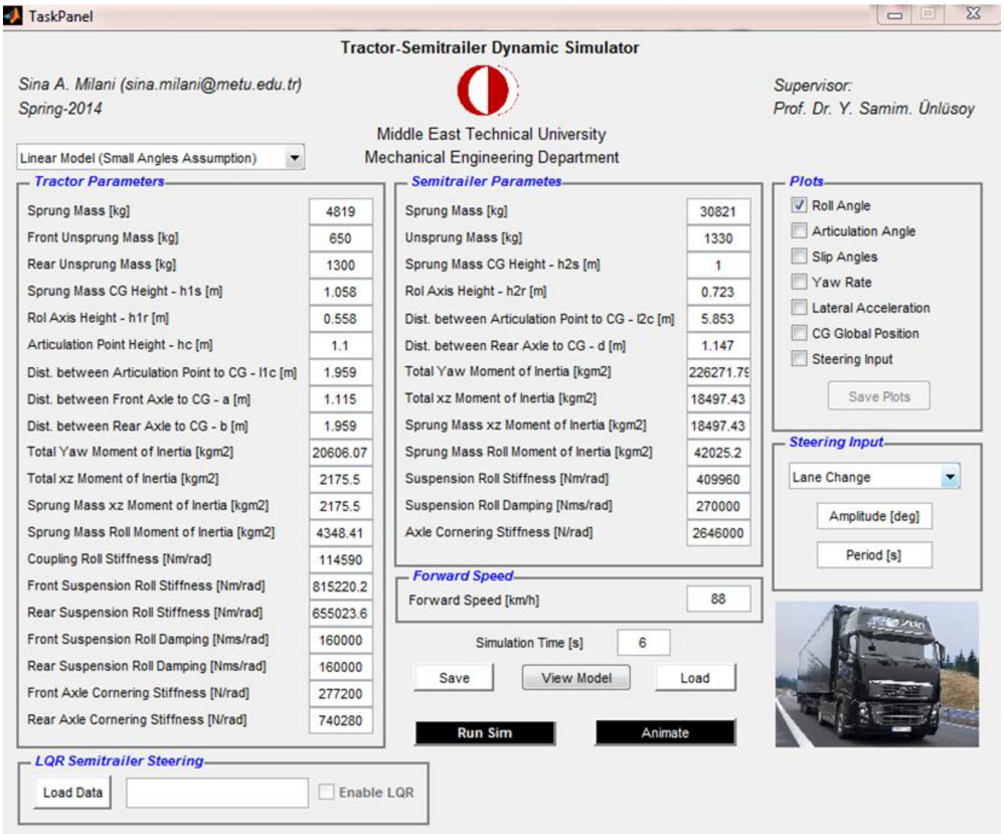

Figure 3.6 - MATLAB GUI Environment

The other simulation environment used in this thesis is the TruckSim® commercial software. This model is developed to validate the results obtained from the simple models namely model 1 and model 2.

3.2.2. Test Maneuvers

There are a couple of commonly used tests throughout the literature corresponding to low and high-speed maneuvers. For the low-speed case, the most widely used standard maneuvers are based on the PBS and UK regulations for Swept Path Width (SPW) measures of low speed turning maneuvers [5], [7], [52], [59]. The vehicle forward speed for low-speed maneuvers are reported differently in various references according to the standard they use, but since the objective of this study is to compare the baseline vehicle with the vehicle equipped with active controllers, we may choose any of the maneuvers as our reference. We assume the vehicle to travel at speed of 10 km/h entering a circular path of 11.25 m radius for the roundabout turning maneuver [7] based on the Australian PBS which is known as SPW_{360} test. Another maneuver is also considered for low-speed performance evaluation which is a similar 90-degree low-speed turn (SPW_{90}). It is worth to note again that the selected maneuvers are only valid to make a comparison between the baseline and controlled vehicles and any attempt for evaluation of a specific vehicle based on the regulatory standards requires exact considerations according to the application region and the conditions presented in the standards.

As far as the high-speed simulation is concerned, majority of the references including [1], [7], [10], [16], [26], [30], [36], [60] have used the very popular SAE J2179 test maneuver which considers a single lane change maneuver at speed of 88 km/h causing lateral displacement of 1.46 m within 61 m of vehicle's forward travel (2.5 sec) [61].

3.2.3. Model Verification

So far, the suitable models have been selected according to the goals of this study. The next step is to validate the proposed model which consequently validates our simulation and results. The validation procedure is performed for the vehicle variables which are important for this study including the lateral acceleration, yaw rate response, and the roll angle response in order to verify the vehicle behavior in the lateral, yaw, and roll motions. A good match between the high-speed maneuver results derived using model 1 and model 2 and the TruckSim® results would imply the reliability of both models due to the fact that vehicle states change more severely at high-speed maneuver. As a result, the standard SAE J2179 maneuver is considered as the basis for model validation.

The vehicle for which the simulations are performed, is a conventional tractor-semitrailer combination based mainly on the vehicle parameters provided in [5]. These vehicle parameters are retained throughout the thesis as the description of the reference vehicle. Details of the parameters are given in Appendix C.

Since the simulation results from model 1 and model 2 are quite similar to each other and almost overlap such that the identification of them would be practically impossible, we avoid including them both; instead, the legend MATLAB model is used to indicate the results from both model 1 and model 2, except the roll response figure, when compared to the TruckSim® results.

In order to make a valid comparison between the models, it is necessary to match the vehicle path so that both models follow the desired path described in the standard. Equal steering inputs may not be considered as the simulation basis due to the probable deviation in the results. This is because of the fact that vehicle models are quite different in the level of details such as the steering mechanism and also the lack of some parameters for such complex vehicle modeling. On the other hand, equal vehicle paths would guarantee the similar conditions under which the vehicles are manipulated.

The common vehicle trajectory following the SAE J2179 is as indicated in Figure 3.7.

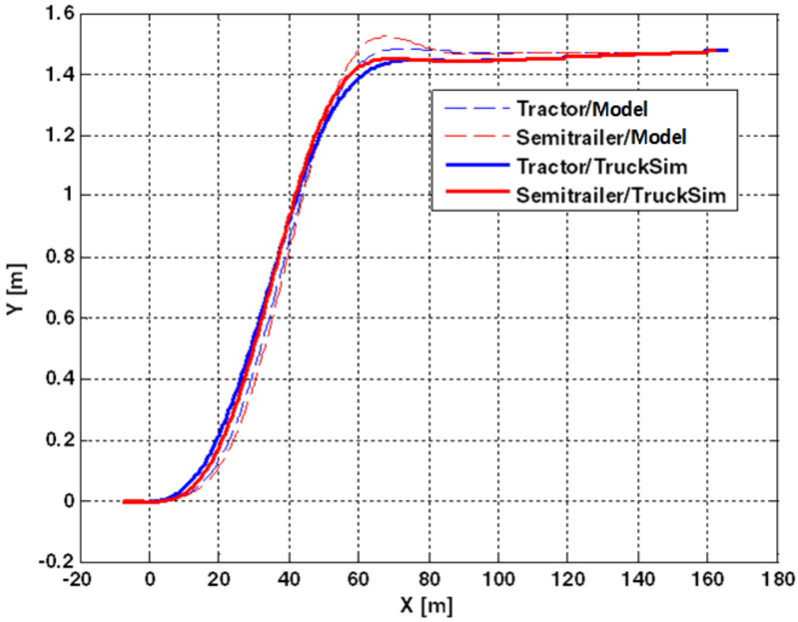

Figure 3.7 - Lane Change Vehicle Trajectories for Model Validation (SAE J2179)

As indicated in the figure, the trajectories are not exactly the same due to the differences in the details of models. It is also possible to use a virtual driver model to follow the prescribed path in order to get more exactly matching trajectories, but it would also introduce oscillations in the steering input to the vehicle affecting the outputs as well which, as a consequence, brings up difficulties to decide whether or not the vehicle model is responding properly. As a result, considering the almost similar vehicle trajectories, we would expect also similar output results from both models.

The lateral acceleration results are given in Figure 3.8.

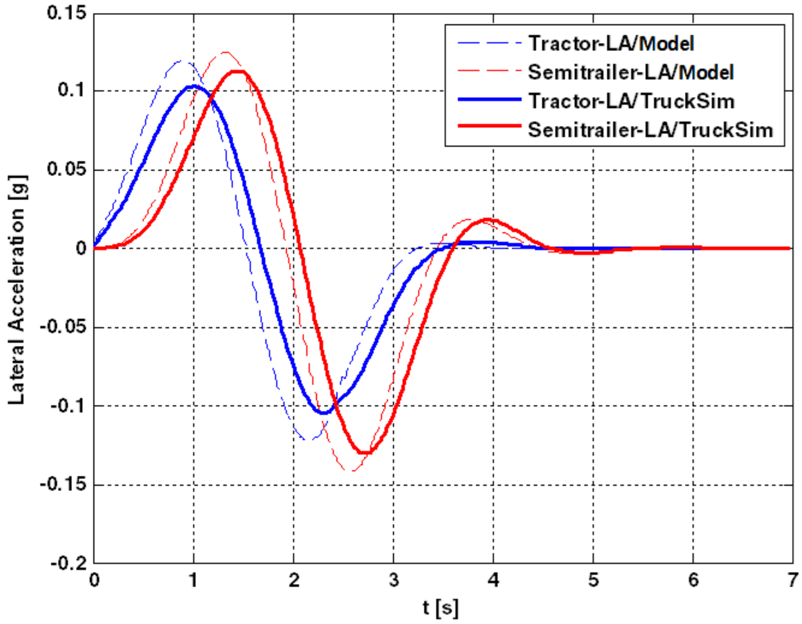

Figure 3.8 - Lateral Accelerations of MATLAB/TruckSim Models (SAE J2179)

From the lateral acceleration results it can be concluded that both models behave similarly in a combination of lateral and yaw motions as defined by the lateral acceleration. Small deviations are also detectable which are expected due to the small differences in initial trajectories and vehicle modeling details including the simplifications made on number of semitrailer axles, neglecting the lateral load transfer, linear tire dynamics, etc.. It is also worth to mention that the deviations are in the direction of higher peak lateral accelerations making the design work to be more conservative rather than unreliable. Overall vehicle model's dynamic behavior is evaluated as acceptable and similar to that of the TruckSim® as far as the lateral acceleration is concerned.

Figure 3.9 gives the yaw rate responses of the vehicle models.

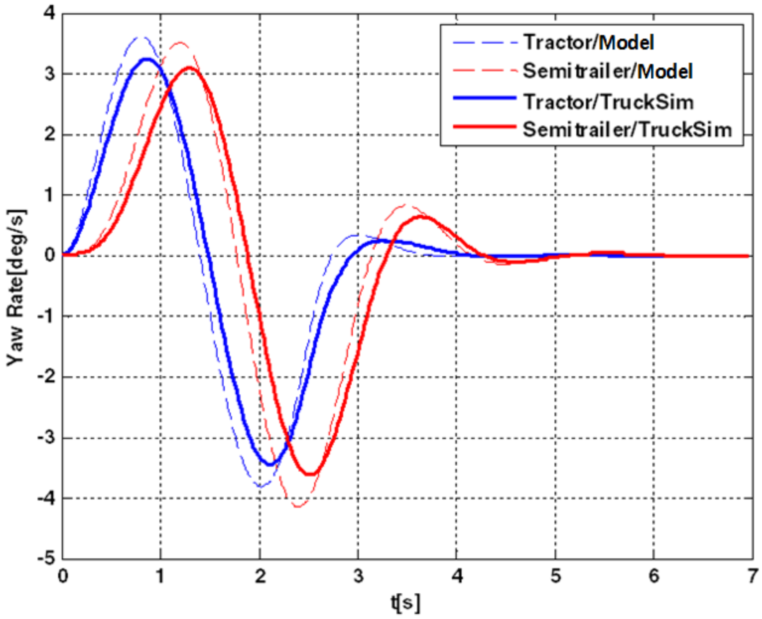

Figure 3.9 -Yaw Rate Response of the MATLAB/TruckSim Models (SAE J2179)

Similar to the lateral acceleration response, the yaw rate of the MATLAB model follows that of the TruckSim® response in its general behavior. The deviations are small and acceptable as stated previously on lateral accelerations.

And finally, the roll behaviors of the vehicle models are as indicated in Figure 3.10.

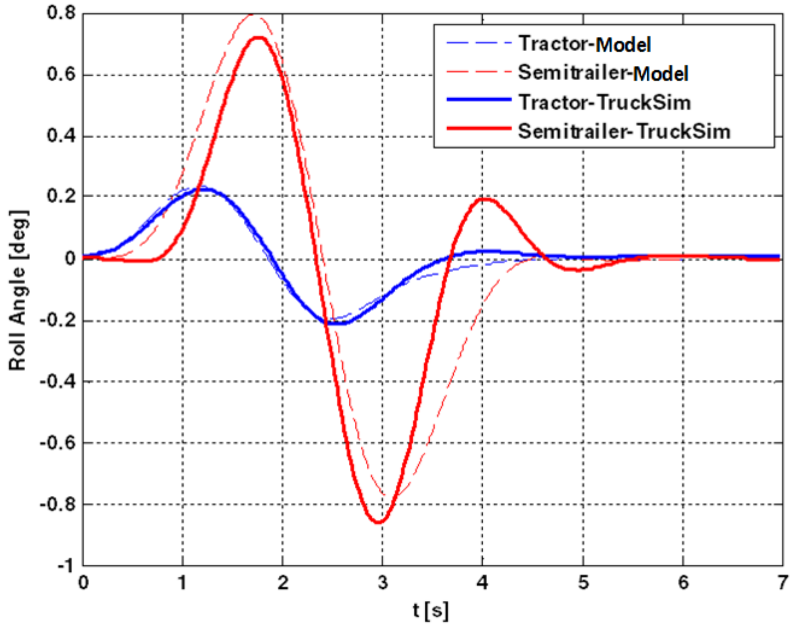

Figure 3.10 - Roll Angles of the MATLAB/TruckSim Models (SAE J2179)

The roll behavior of the tractor unit closely follows that of the reference TruckSim® model, but there are some variations in the roll responses of the semitrailer units. Although the general behavior of the models seems to be consistent, the oscillations of the MATLAB model (model 2) are more likely to be damped compared to the TruckSim® model. This could be as a result of the modeling details and simplifications we made in modeling the roll dynamics of the linear model, e.g. neglecting the unsprung mass roll dynamics and the lateral load transfer are potential sources of such discrepancies. This shows that the simple linear roll-plane model may not be an exact model for detailed roll analyses, but as far as this study is concerned, the roll-plane model is able to predict peak roll angles and general behavior of the vehicle in roll motion with acceptable accuracy during a lane change high-speed maneuver.

Regarding above analyses and comparisons, it can be concluded that the simplified models namely model 1 and model 2 show reasonably reliable dynamic behaviors in the scope of this study and could be used as our reference models for the controller design and simulation purposes. Hence, instead of dealing with complicated vehicle models, we are allowed to apply the simplified models and generate fast, simple and user-friendly mini-software by utilizing MATLAB’s GUI environment for the purpose of general design and study of the tractor-semitrailer combinations with active safety controllers under low lateral acceleration maneuvers.

3.2.4. Low-Speed and High-Speed Challenges

At this stage, it would be beneficial to observe the behavior of the tractor-semitrailer combination at low and high-speed maneuvers based on simulations of our reference models. Such observations give insight to the designer to deeply understand the problem and develop physical explanations for them which finally help him to apply appropriate control algorithms.

3.2.4.1. Low-Speed

As mentioned previously, the most undesirable behavior of the AHVs at low-speed maneuvers is the PFOT. In order to understand the severity of this phenomenon, it would be beneficial to run a low-speed 360-degree turning maneuver for a conventional tractor-semitrailer with the prescribed parameters.

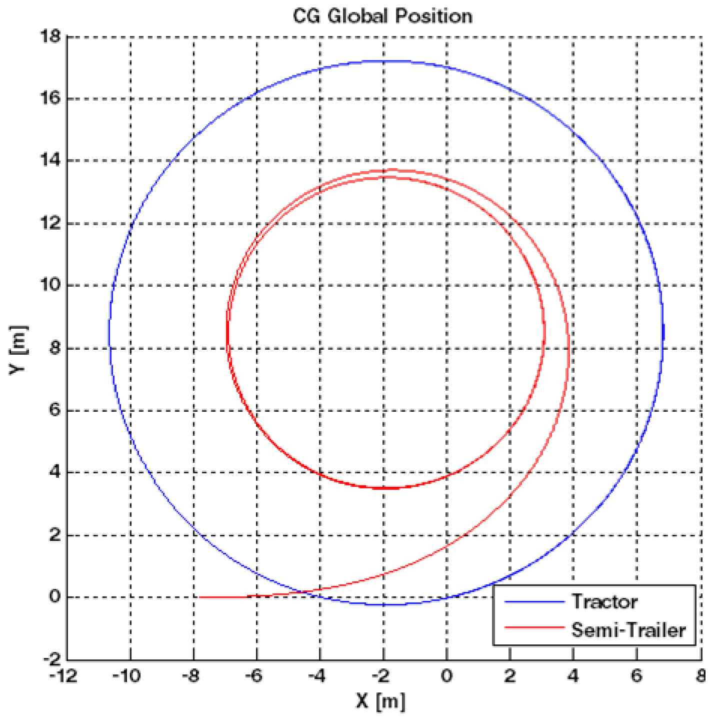

Figure 3.11 - Low-Speed PFOT of Conventional Combination at 5 km/h

As depicted in Figure 3.11, in a 360 degree turning maneuver at speed of 5 km/h, the tractor's front axle follows a circular path of about 9 m radius while the semitrailer unit creates a much smaller circle with radius of less than 5 m leading to an off-tracking value of larger than 4 meters which is definitely quite undesirable for such a tight maneuver. This is clear that in order to compensate for such an undesirable behavior, the active steering applied to the semitrailer wheels must steer them in opposition to the driver's steering direction in order to reduce the tracking offset.

3.2.4.2. High-Speed

At high-speed, RA is the cause of increased risk of rollover as well as HSTO both of which raise the accident risk and impose danger to the surrounding traffic. But would it be desirable to reduce the RA as much as possible? In order to respond to that question, one needs to further understand the RA and its physical explanation.

As stated briefly, the semitrailer's tire lateral forces are developed with a time delay with respect to that of the tractor [12]. This causes the tractor unit to stay a step ahead of the semitrailer in changing direction and causing the semitrailer to go outwards, to some extent, from the intended path. Under such circumstances, the semitrailer follows a larger circle while the tractor experiences smaller radius turn assuming a steady-state turn. At this point, the forward speeds of the vehicle units are not exactly the same due to the different heading angles they possess; on the other hand, if we assume both units to be in steady-state motion, the rate at which they are turning would be constant, i.e. the yaw rates being equal. The equality in the yaw velocity of both vehicle units enables us to do a simple mathematical analysis; the centripetal acceleration of the units is directly proportional to the radius of curvature through the following relationship:

$$a_C = r\omega^2 \tag{3.1}$$

Since the angular velocities of both units are equal, the centripetal acceleration of the semitrailer would be slightly higher than that of the tractor unit due the larger radius of the circular path. Lateral acceleration, on the other hand, would be approximately equal to the centripetal acceleration of the vehicle units implying the amplification in lateral acceleration response of the semitrailer with respect to tractor. During a transient maneuver, larger semitrailer tire forces, and consequently larger lateral acceleration are developed to put the towed unit back into its steady-state position from which it has deviated outwards.

Therefore, reduction in RA ratio corresponds to the less outwards motion of the semitrailer. RA ratio of 1.0 corresponds to the zero HSTO in steady-state meaning that the semitrailer follows the path of tractor exactly. Further reduction in RA ratio causes the semitrailer to run inwards similar to low-speed maneuvers. Some conventional studies aimed to reduce the RA ratio as much as possible for high-speed maneuvers in order to minimize the rollover risk [25], [35]; but there have been some arguments about the issue due to the fact that minimizing the rearward amplification further than a limited value, which appears to be 1.0, degrades the following capability of the towed unit as increased HSTO [1], [26].

Simulation results are presented here in order to understand the phenomenon more easily by looking at the diagrams. Figure 3.12 shows the high-speed lane change behavior for a conventional tractor-semitrailer experiencing certain amount of RA.

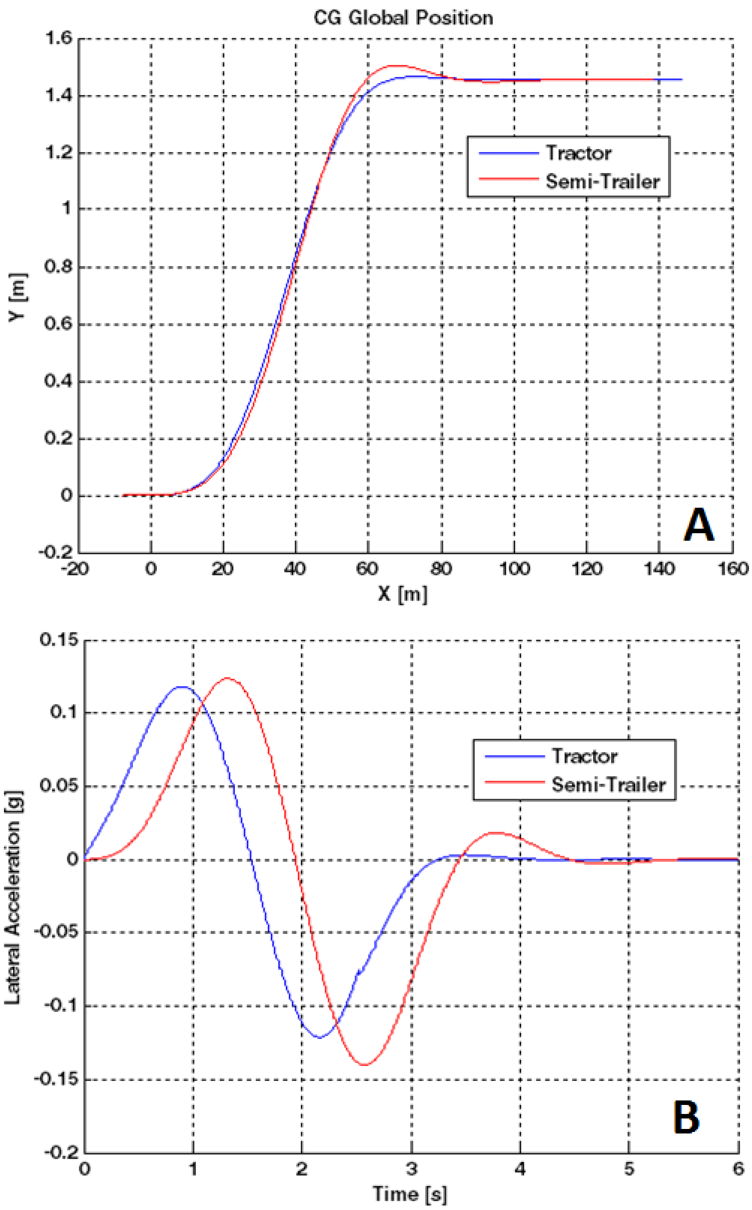

Figure 3.12 - SAE J2179 Lane Change for a Tractor-Semitrailer with RA>1.0 (A: Vehicle Trajectory, B: Lateral Acceleration)

It can be seen that the semitrailer tends to stay outwards, specifically at the end of the maneuver, with respect to the tractor which corresponds to the RA ratio of greater than 1.0 as shown in the lateral acceleration plot.

If a semitrailer steering control algorithm is applied and tuned such that the RA ratio is reduced to values less than 1.0, the results would be similar to Figure 3.13 with the semitrailer moving inwards.

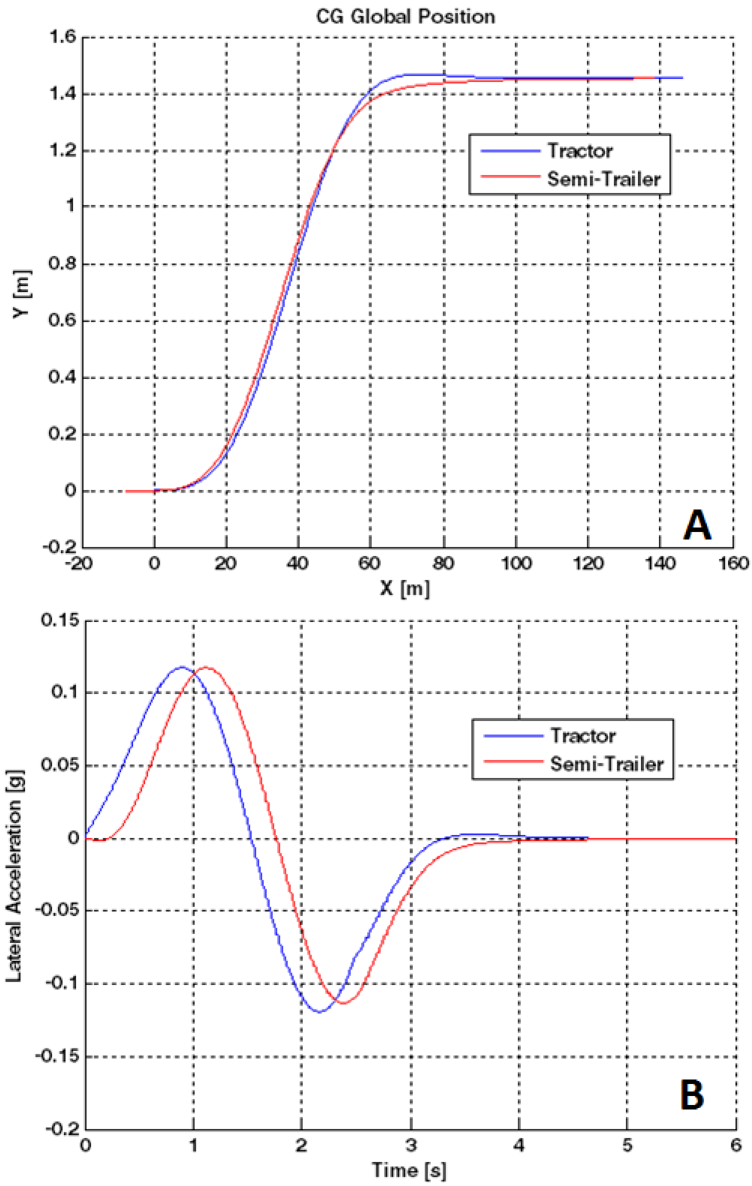

Figure 3.13 - SAE J2179 Lane Change for a Tractor-Semitrailer with RA<1.0 (A: Vehicle Trajectory, B: Lateral Acceleration)

Figure 3.12 and Figure 3.13 are clear demonstrations of the stated argument about the criteria to be taken when treating the RA ratio. In the next chapters, there will be some other simulations verifying that the desired RA ratio of 1.0 would cause the control system to result in the best circumstances from the HSTO viewpoint.

At this point, it is the designer's criteria to make this compromise between increased HSTO and risk of rollover. In other words, it seems that the active steering strategy of the towed unit is not fully favorable and may imply the need to use additional controls in order to compensate for this conflict.

CHAPTER 4

ACTIVE STEERING CONTROL

4.1. General

The results obtained from the previous chapters imply that AHVs exhibit undesired characteristics that require some control approaches to overcome the dynamic weaknesses. Among the most common AHV controllers, semitrailer steering control seems to be the most effective approach due to its favorable effects for low and high-speed cases and also imposing large effects with relatively small energy consumption. The lateral forces generated by semitrailer's slightly steered axles generate very large yawing moment, due to the much larger lever-arm, compared to a differential braking system which consumes a lot of energy to generate braking forces to control the yaw motion. Besides, the semitrailer steering system has the advantage that enables it to be favorable for both low and high-speeds by means of active controllers while the DB system is only applicable for high-speed yaw control approaches. On the other hand, active anti-roll bars, when used as the sole controller, also consume a lot of energy in order to generate very large roll moments, specifically for heavy vehicles with massive sprung masses. However, active semitrailer steering systems are also able to reduce the roll motion, to some extent, at high-speeds by regulating the RA ratio.

As a result, it would be worthwhile to consider the application of active semitrailer steering with a focus on eliminating its weak spots in order to come up with an optimal design solution being able to meet all of our design objectives, namely PFOT reduction, roll reduction, HSTO minimization and RA regulation at the same time.

Based on the results obtained from literature, one of the best suited control algorithms in dealing with AHVs is the LQR approach. Nowadays, with the advent of measurement and control electronic devices, application of feedback controllers has been greatly influenced positively resulting in quite a large number of active electronically controlled mechanical systems. The LQR would be a favorable approach when the design problem is coupled with minimization of certain variables or functions while the linear controller design is applicable, e.g. not dealing with a highly nonlinear system. All of these conditions apply to our system of interest emphasizing on the use of LQR being appropriate.

To put it in a nutshell, the LQR is a very popular approach for dealing with lateral control of AHVs because of its simplicity, feasibility and favorability compared to other control methods such as fuzzy logic and sliding mode controls. In the following sections, design of an LQR active semitrailer steering control is going to be discussed and then the necessity of an additional anti-roll system is going to be explained.

4.2. LQR Technique

The linear quadratic regulator problem is generally defined as the problem of generating an optimal control vector consisting of state feedback so as to minimize a certain performance index or cost function [62].

Suppose the linear system as:

$$\{\dot{x}\} = [A]\{x\} + [B]\{u\} \tag{4.1}$$

with the control input vector as:

$$\{u\} = -[K]\{x\} \tag{4.2}$$

which minimizes the following cost function:

$$J = \int_0^\infty (\{x\}^T[Q]\{x\} + \{u\}^T[R]\{u\})\, dt \tag{4.3}$$

in which $[Q]\ and\ [R]$ are positive semidefinite and positive definite matrices, respectively.

The feedback matrix $[K]$ is determined using the following relation:

$$[K] = [R]^{-1}[B]^T[P] \tag{4.4}$$

in which, $[P]$ is the solution to algebraic Riccati equation:

$$[A]^T[P] + [P][A] - [P][B][R]^{-1}[B]^T[P] + [Q] = 0 \tag{4.5}$$

Recalling the state-space representation of our linear system of interest, the state matrix has the form:

$$\{x\} = [\phi_1 \quad \dot{\phi}_1 \quad \beta_1 \quad \dot{\psi}_1 \quad \phi_2 \quad \dot{\phi}_2 \quad \beta_2 \quad \dot{\psi}_2]^T \tag{4.6}$$

and the input matrix takes the form:

$$\{u\} = [\delta_1 \quad \delta_2]^T = [u_d \quad u_t]^T \tag{4.7}$$

in which u_d is the driver's steering as the reference input and u_t is the semitrailer's steering angle as the control input, leading to the following modified state-space representation in which the two different input types are separated:

$$\{\dot{x}\} = [A]\{x\} + [B_d]u_d + [B_t]u_t \tag{4.8}$$

$$u_t = \delta_2 = -[K]\{x\} \rightarrow \{\dot{x}\} = ([A] - [B_t][K])\{x\} + [B_d]u_d \tag{4.9}$$

and the performance index for the system is:

$$J = \int_0^\infty (\{x\}^T [Q]\{x\} + u_t.R.\ u_t)\ dt \tag{4.10}$$

Note that the input weighting matrix $[R]$ reduces to a single scalar for this case.

4.3. Hardware Requirements

In order to be able to apply LQR technique for the tractor-semitrailer combination, certain measurement devices are required for making system states available.

Regarding the definition of the states, two angular displacement sensors are required for measurement of roll angles of two vehicle units as well as two roll rate sensors.

Two side-slip angle measurement devices are needed which could be based on either GPS signals or optical sensors [6], [20], [30]. In case the conditions does not allow such sensors to be available, other techniques such as specific vehicle modeling approaches [63] and use of Kalman filter [26]. In the current study, it has been assumed that all vehicle states are readily available with proper accuracy and signal quality and there is no need for filters to estimate the vehicle states.

Finally, two yaw rate sensors are required to measure the yaw velocity of vehicle units which are easily available and widely used in the automotive industry.

In order to apply the steering action, an Electronic Control Unit (ECU) is required along with some actuators which might be hydraulic or electric to steer the semitrailer wheels within the predesigned steering mechanism according to the developed control algorithm. The reliability of the steering mechanism must be certain in order to avoid unwanted steering of the wheels in case of any unexpected vehicle condition, movement and also during reversing.

Such electronic devices enable the designers to implement advanced steering algorithms, which are usually known as steer-by-wire systems, without the need for complicated mechanical linkages.

4.4. LQR Weighting Factors

LQR Weighting factors are, in fact, parameters that determine the states, or a combination of them, which are going to be penalized in order for the controller to optimally meet a minimization objective. Weighting factors constitute the elements of weighting matrices $[Q]$ and $[R]$ acting on the states and inputs in the definition of the cost function.

The next step in design of the controller is to translate the desired physical control objectives into mathematical expression as our performance index or cost function to be minimized. According to the LQR theory, cost function must consist of quadratic terms in order for the $[Q]$ and $[R]$ matrices to be derivable.

However, it would be very difficult in this case to define such quadratic cost functions expressing the PFOT at low-speeds and HSTO or RA at high-speeds. Specifically at high-speed, there is a phase difference between generations of lateral accelerations for the vehicle units and defining a single mathematical relationship for calculation of RA ratio would require dependency of the cost function to parameters such as vehicle forward speed. An alternative approach is to define the lateral accelerations of the vehicle units separately or simultaneously as our cost function which also increases the complexity in mathematical derivations as well as decreasing the accuracy level. At low speed, on the other hand, it would be difficult to define a quadratic function of PFOT also being applicable to LQR technique.

In order to eliminate such complexities and also increase the efficiency level of the LQR design, it is decided to make use of an optimization method in determining the weighting factors which are, in fact, the elements of $[Q]$ and $[R]$ matrices. In this approach, the designer assumes the general form of the weighting matrices and tries to find applicable values for their elements based on an iterative optimization method. During the optimization, another cost function called the fitness function is going to be calculated repeatedly in each loop in order to evaluate the overall performance of the controller and the level of fitness for each control parameter set. This time, the designer is free to choose any type of cost function even by looking at the overall performance of the system during its operation and/or looking at some specific times.

4.4.1. Low-Speed

At this step, we need to define specific configurations for weighting matrices and determine the number of parameters which are going to be optimized. It is usually customary to assume $[Q]$ and $[R]$ to be diagonal matrices for the sake of simplicity. In this case, however, it is suggested that we could assume matrix $[Q]$ to be non-diagonal, but with lots of zeros in it while the matrix $[R]$ is automatically reduced to a single scalar. Regarding our system of interest, it can be claimed that, at low-speed, most of the system states experience very small values as well as very small change rates such as roll angles and their rates of change as well as the side-slip angles. This, in fact, makes them negligible to consider as vital system states at low-speed run.

Yaw rates, on the other hand, are of great importance as far as the low-speed run is concerned, because they are directly related to the location of the center of curvature about which the vehicle units are turning. In other words, if there would be an improvement in low-speed turning ability of the tractor-semitrailer by use of LQR technique, it would be based on a cost function generated by a combination of these substantially relevant states. For this reason, it is suggested to consider weighting matrices as:

$$[Q] = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & q_1 & 0 & 0 & 0 & q_2 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & q_2 & 0 & 0 & 0 & q_3 \end{bmatrix} \tag{4.11}$$

$$R = r_1 \tag{4.12}$$

The nonzero elements of $[Q]$ will collect the significant state variables, namely $\dot{\psi}_1$ and $\dot{\psi}_2$in a quadratic combination leading to the following LQR cost function:

$$J = \int_0^\infty \left(q_1\dot{\psi}_1^2 + 2q_2\dot{\psi}_1\dot{\psi}_2 + q_3\dot{\psi}_2^2 + r_1\delta_2^2 \right) \, dt \tag{4.13}$$

The problem is now reduced to determination of q_1, q_2, q_3, r_1 by means of an optimization method to meet our design needs.

4.4.2. High-Speed

At high-speed operation of the vehicle, tractor's lateral behavior is mainly determined by the driver's steering input. The only interaction between two vehicle units is due to the forces at the hitch point. These forces are not significantly affected by applying the active semitrailer steering. In other words, active semitrailer steering has more significant effect on the yawing behavior of the towed unit rather than affecting the tractor unit. As a result, it is suggested to simplify the weighting matrix $[Q]$ by neglecting the element corresponding to the tractor unit. It means that we concentrate on minimizing a combination of states related to semitrailer unit by means of LQR active steering.

Furthermore, in order to avoid additional complications, the matrix $[Q]$ is assumed to be diagonal meaning that the states are going to be minimized one by one which is quite reasonable in sense of stability improvement. The ratio of the weighting factors determines the relative reduction in the state variables according to their significance.

$$[Q] = diag(0,0,0,0,q_1,q_2,q_3,q_4) = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & q_1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & q_2 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & q_3 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & q_4 \end{bmatrix} \tag{4.14}$$

$$R = r_1 \tag{4.15}$$

In a manner similar to the low-speed case, the cost function of the LQR controller is determined as:

$$J = \int_0^\infty \left(q_1\phi_2^2 + q_2\dot{\phi}_2^2 + q_3\beta_2^2 + q_4\dot{\psi}_2^2 + r_1\delta_2^2 \right)\, dt \tag{4.16}$$

So far, the requirements for LQR technique are provided with very favorable and yet reasonable simplifications which are claimed to result in desirable control outcomes. These simplifications eliminate the need for complicated calculations while avoiding manual adjustment of the weighting matrices in order to come up with the most cost-effective controller design procedure with intuitive analysis of the vehicle dynamic behavior based on designer’s insight. Other considerations regarding the procedure of optimization and the corresponding fitness functions are going to be discussed in the next section.

4.5. Quantum Particle Swarm Optimization (QPSO)

LQR technique as a handy optimal control theory requires the determination of its weighting matrices in order to perform properly. Good operation of the controller could be achieved by properly selecting the elements of the weighting matrices. This can be done by manual trial and error procedure in which the designer adjust the weighting factors intuitively and observes their effect on the system and tries to reach the optimum solution. But in in order to follow a more systematic approach, several optimization methods could be applicable including Bryson’s method [64], pole placement method [65], and genetic algorithm. There is also another rather recent technique called the Particle Swarm Optimization (PSO) introduced by Eberhart and Kennedy [66] which is based on movement of a swarm of particles obeying the Newtonian mechanics. It is concluded that the PSO method has advantages over other methods such as the fewer need for iteration and faster convergence which makes it more suitable for application in LQR weighting factor optimization. There has been an extension to the PSO presented recently which assumes the particles to follow the rules of quantum mechanics [67]. This method is later called as Quantum Particle Swarm Optimization (QPSO) which has the advantage over PSO that it is less likely to get stuck in the local optima. There has been also concluded that the QPSO-based LQR has shown more desirable results in terms of overshoot, steady-state error, etc. compared to LQR which is tuned by PSO, genetic algorithm, and manual adjustment techniques [68].

In the current study, QPSO is selected as the optimization method as one of the most recent approaches for tuning of LQR parameters. In the following, the details of QPSO method are going to be discussed to some extent.

4.5.1. Method

Starting with PSO as a more general consideration, the method is inspired by the social behavior of flocking birds. The algorithm deals with a population of particles which are treated as the candidate solutions to the optimization problem. A set of particles constitutes a population which is called a potential solution. Initially, the particles are given a position and velocity randomly; at each time step, the swarm is given a fitness value regarding the positions of particles and the velocities of the particles are altered based on the laws of Newtonian mechanics forcing the particles to follow a trajectory towards a global optimum point. The definition of the fitness function is left to the designer to insert all of the design criteria with proper weighting factor according to his intention. The behavior of the particles is also affected by application of two positive parameters called personal and social cognitive factors as well as an inertia parameter for convergence enhancement [66], [68].

In order to develop the PSO method, Sun et al. [67] introduced new properties for the PSO method in which the particles' motion follows the quantum mechanics approach known as QPSO. With this property, the positions of the particles are actually estimated rather than being known; this imposes an inherent randomness in the method which reduces the probability of getting stuck in local optima. The details of the mathematical considerations for the method could be found in [67] and are beyond the scope of this material.

Formulations of optimization required for the computer programming of a QPSO algorithm are given in the following [67]:

Initialize the population: random x_i

do

for I=1 to M

$if\ f(x_i) < f(p_i)\ then\ p_i = x_i$

$if\ f(p_i) < f(p_g)\ then\ p_g = p_i$

for d=1 to D

$\Phi_1 = rand(0,1)$

$\Phi_2 = rand(0,1)$

$p = \left(\Phi_1 \times p_{id} + \Phi_2 \times p_{gd}\right)/\Phi_1 + \Phi_2$

$u = rand(0,1)$

$L = \left({}^{1}/_{G}\right) \times abs(x_{id} - p)$

$if\ rand(0,1) > 0.5$

$x_{id} = p - L \times \ln\left({}^{1}/_{u}\right)$

else

$x_{id} = p + L \times \ln\left({}^{1}/_{u}\right)$

end

end

until the termination criterion is met

If the upper limits of optimization parameters Φ_{1k}, Φ_{2k} are selected properly, the convergence of the method is guaranteed. Enhancement in the convergence time could be achieved by adjusting the parameter G, the population size, number of iterations and/or other termination criteria.

4.5.2 Application

At this stage, the design objective needs to be converted into a mathematical form called the fitness function to be used in the optimization procedure. Optimization parameters as described in the previous section are required to be tuned as well;

finally the computer programming code must be generated using MATLAB to implement the QPSO for both low and high-speed cases according to the suitable vehicle models.

At low-speed, the only design criterion is to minimize the PFOT. Since the calculation of exact PFOT for the low-speed case in a computer code could complicate the programming, another function is defined to be used as a measure for the value of PFOT:

$$E = \sqrt{(X_1' - X_2)^2 + (Y_1' - Y_2)^2} \tag{4.17}$$

The value of E function is, in fact, the distance between current position of semitrailer's rear axle denoted by X_2, Y_2 and that of the tractor's front axle at the corresponding previous time denoted by X_1', Y_1' calculated according to vehicle forward speed. The value of error E is calculated at the points of interest during the maneuver as well as the summation of it in order to constitute the fitness function for low-speed optimization. The value of the steering input could also be added in case the control energy is to be limited for the general form of fitness function:

$$\begin{aligned} f_{p,lowspeed} = c_1|E_1| + c_2|E_2| + \cdots + c_n|E_n| + d_1 sum(|E|) \\ + d_2 \max(|\delta_2|) \end{aligned} \tag{4.18}$$

The c_i's and d_j's are weighting factors to give proper importance to each term.

It is worth to mention that the performance of the ASC system would be more limiting during the 90-degree turn maneuver than 360-degree maneuver. Since the steady-state condition is not achieved during 90-degree maneuver, some tracking errors are detected in the beginning of the maneuver as an outward motion of the semitrailer; for this reason, the 90-degree turn is used as the basic maneuver for LQR optimization to make sure that the ASC works well for wider range of low-speed maneuvers, regardless of the steady-state or transient low-speed maneuvers.

At high-speed, the design criteria are to regulate RA and yaw rate response as well as HSTO improvement and roll motion reduction. Regarding the previous explanations, regulation of the RA ratio improves the HSTO and regulates the yaw rate response as well due to the fact that the yaw velocity actually contributes in generation of the lateral acceleration. On the other hand, reduction in RA ratio calms the roll motion as well by reducing the centripetal acceleration. Hence, dealing with RA for high-speed control seems to be a comprehensive approach. In the Simulation Test Results section, there will be more concentration and discussion on the effects of RA regulation on other vehicle responses.

For the high-speed case, the calculations are relatively easier and the designer only needs to compare the peak lateral accelerations of the vehicle units during the lane-change maneuver. This observation then results in the following general fitness function:

$$f_{p,highspeed} = c'_1 \left|\max(a_{y2}) - \max(a_{y1})\right| + c'_2 \left|\min(a_{y2}) - \min(a_{y1})\right| + c'_3 \max(|\delta_2|) \tag{4.19}$$

which is calculated based on the results of each simulation at every iteration.

4.6. Lookup Table

As a result of LQR optimization, the optimal feedback gain matrix is obtained for the prescribed vehicle forward speeds as well as maneuver types. It is worth to mention that this controller design process is only valid for the two vehicle speeds at which the optimization algorithm is applied. In order to provide valid optimal state-feedback control for a wide range of vehicle speeds, we need to develop a lookup table consisting of feedback gains corresponding to various speeds.

For this reason, the same optimization procedure is applied to a set of vehicle speeds namely at 10, 20, …, 120 km/h. It is concluded experimentally that at speeds between 40 and 70 km/h there is no need for steering control neither for PFOT enhancement nor RA reduction. RA phenomenon actually starts to appear at speeds above 70 km/h for the prescribed vehicle parameters. At speeds between these values, the elements of feedback matrix are calculated using interpolation due to the consistency in the mechanical system and for speeds below 10 km/h and over 120 km/h the border values are going to be considered. It is worth to mention that the maneuvers for which the feedback values are optimized are the same as we concentrated on in the previous sections. So there is no claim that the feedback values work for the entire probable vehicle maneuvers optimally, but the maneuvers of interest, especially at high-speed, are very basic ones which include the main behaviors of the vehicle at other maneuvers to some extent. The performance of the control system on maneuvers other than the basic ones is also evaluated in the Simulation Results section.

The optimal values for feedback matrices and corresponding weighting matrices are given in Appendix B.

4.7. Suggested Additional Control Strategies

So far, the basic control strategy for active semitrailer steering is introduced. In this section, there are two alternative control methods suggested to improve the high-speed roll limitation characteristic of the control system, based on the designer’s preference. These suggestions are only given for making a comparison between the original ASC controller and the modified ones and are not developed for a wide range of vehicle speeds.

Further demand for roll limitation brings up the question: what if we reduce the RA ratio further in order to achieve better roll limitation? This will happen obviously at the expense of larger tracking errors known as HSTO as described in the previous sections. The overall performance of such a controller must be evaluated by looking at the level of reduction in roll angle as well as the HSTO degradation as a compromise. This could easily be handled by adjusting the optimization criteria by adding an additional term to equation (4.19) to account for roll angle reduction:

$$\begin{aligned} f_{p,highspeed} = {} & c'_1 \left|\max\left(a_{y2}\right) - \max\left(a_{y1}\right)\right| \\ & + c'_2 \left|\min\left(a_{y2}\right) - \min\left(a_{y1}\right)\right| + c'_3 \max(|\delta_2|) \\ & + c'_4 \max(|\phi_2|) \end{aligned} \tag{4.20}$$

Another solution is to add some other control inputs to the system such as a roll control torque which may be applied by means of active anti-roll bars [46], [54], [69-71]. Since the roll motion has very little effect on lateral behavior of the vehicle in the scope of interest, adding the roll control torque would have the advantage that the ASC system would take the responsibility of HSTO minimization while the Active Roll Control (ARC) torque takes care of further roll motion limitation. For implementation of such a control strategy, besides the input matrix $[B]$, equations (4.7)-(4.10) must be altered respectively as:

$$\{u\} = [\delta_1 \quad \delta_2 \quad T_{AR2}]^T = [u_d \quad \{u_t\}^T]^T \tag{4.21}$$

$$\{\dot{x}\} = [A]\{x\} + [B_d]u_d + [B_t]\{u_t\} \tag{4.22}$$

$$\begin{aligned} \{u_t\} = [\delta_2 \quad M_{AR2}]^T = -[K]\{x\} \rightarrow \{\dot{x}\} \\ = ([A] - [B_t][K])\{x\} + [B_d]u_d \end{aligned} \tag{4.23}$$

$$J = \int_0^\infty (\{x\}^T[Q]\{x\} + u_t[R]\,u_t)\,dt \tag{4.24}$$

And the matrix $[R]$ becomes:

$$[R] = \begin{bmatrix} r_1 & 0 \\ 0 & r_2 \end{bmatrix} \tag{4.25}$$

The additional control input weighting factor r_2 is selected as small as possible in order to have the maximum possible roll minimization.

CHAPTER 5

SIMULATION TEST RESULTS

5.1. General

In the following chapter, the LQR steering control is applied on the vehicle models 1 and 2 for low and high-speed maneuvers which were previously discussed. The objective of this chapter is to evaluate the effectiveness and the performance of the proposed controller and to enhance its overall performance by some additional options.

The chapter is divided into several simulation tests for evaluation of the actively steered tractor semitrailer. Each simulation is performed for a specific purpose which is discussed in the corresponding section. For every test, simulation results such as vehicle trajectory, control input, tire slip angles and articulation angle are presented as well as some specific results such as the PFOT measure, lateral acceleration and roll angles for specific maneuvers.

5.2. 360-Degree Turn at 10 km/h (SPW_{360})

The basic low-speed maneuver according to PBS, as described in the previous sections, is the low speed 360-degree turning which is explained in [7]. The vehicle response is presented in both baseline vehicle and the vehicle equipped with ASC of semitrailer.

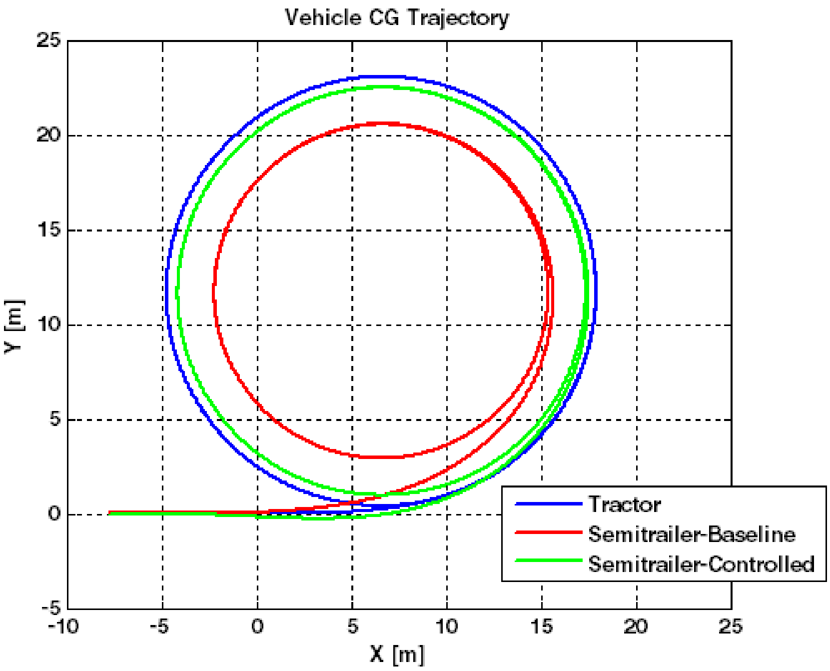

Figure 5.1 - Vehicle CG Trajectory for SPW$_{360}$ Maneuver – Baseline vs. ASC

Figure 5.1 shows the vehicle CG trajectory for both units in baseline and actively steered cases. It can be concluded that the ASC could dramatically improve the tracking ability of the semitrailer. There is however a small degradation in the tracking ability at the beginning of the maneuver where the semitrailer's rear end exhibits a little bit of outwards movement, but its negative effect is quite small compared to the improvement in steady-state response.

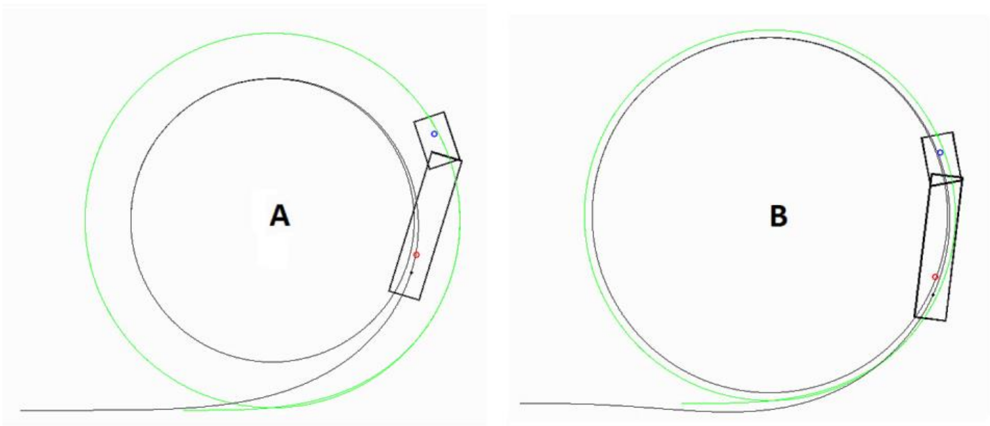

Figure 5.2 - Top-View of the Vehicle in SPW_{360} Maneuver (A: Baseline Vehicle, B: Vehicle Equipped with ASC of Semitrailer)

The tracking improvement is depicted more explicitly in Figure 5.2 with the green line indicating the path of tractor's front axle and the black line indicating that of the semitrailer's axle.

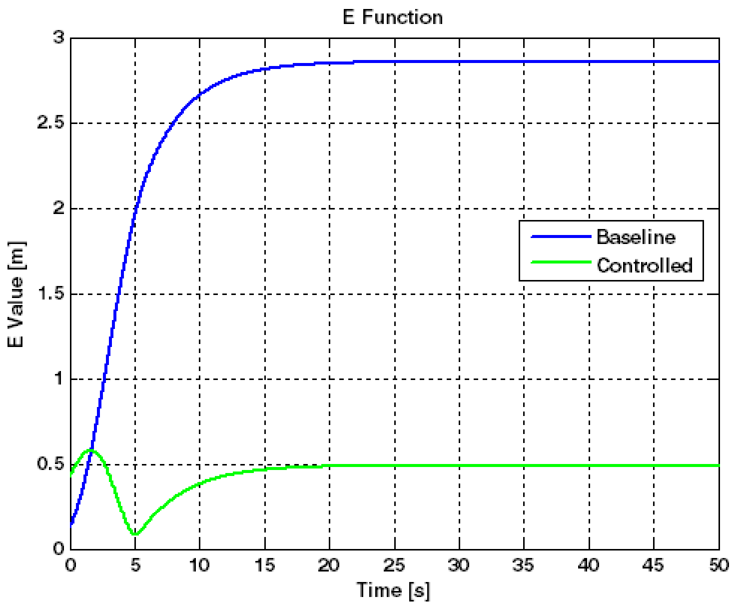

Figure 5.3 - The E Function Value as a Measure of PFOT for SPW_{360} Maneuver – Baseline vs. ASC

Figure 5.3 shows the value of E function as described in the controller design section. It can be seen that the approximate value of PFOT has been reduced from about 2.8 m to 0.5 m, which is quite desirable.

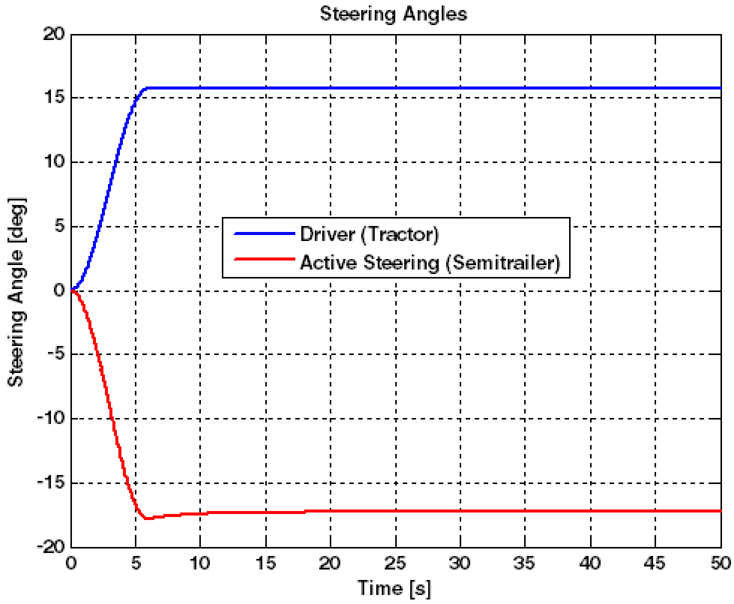

Figure 5.4 - Steering Angles for SPW_{360} Maneuver

As depicted in Figure 5.4, the driver's steering input is considered as a ramped-step input of 15.7 degree amplitude. As a result of ASC, the semitrailer wheels are steered accordingly with a slightly larger value compared to the drivers input. This can be explained regarding the fact that in order for the vehicle units to share a common center of curvature, according to the Ackerman steering geometry, the unit with longer wheelbase requires sharper steering with respect to the shorter vehicle. The condition is partially achieved by the application of semitrailer ASC resulted in such steering angles.

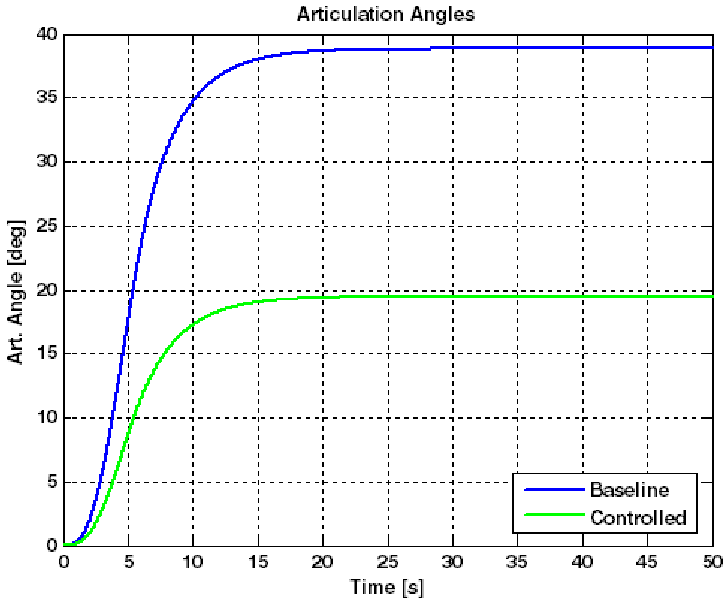

Figure 5.5 - Articulation Angles in SPW_{360} Maneuver – Baseline vs. ASC

Articulation angle of the vehicle is also reduced by about 50% as a result of ASC, but it can be seen that the values are quite large to be approximated by small angle assumption validating the use of nonlinear model for low-speed maneuvers.

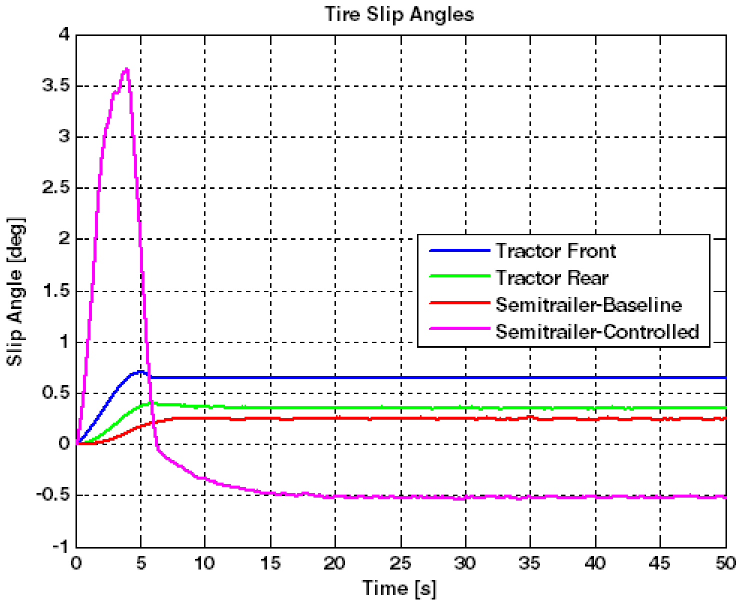

Figure 5.6 - Tire Slip Angles for SPW_{360} - Baseline vs. ASC

Tire slip angles are shown in Figure 5.6 for both baseline and the vehicle equipped with ASC. It is worth to mention that the ASC has almost no effect on the slip angle response for the tractor unit as mentioned previously. In case of ASC, the semitrailer slip angle rises at the beginning of the maneuver due to the sudden steering control applied on the semitrailer wheels as well as the simulation accuracy; it is then greatly reduced to a small level which validates the use of linear tire model assumption with small tire slip angles.

5.3. 90-Degree Turn at 10 km/h (SPW_{90})

The 90-degree turning maneuver at 10 km/h is another important PBS measure explained in [7] (SPW_{90}) which is taken as the basis of LQR optimization for low-speed maneuvers. The following simulation results explain the reason for this selection:

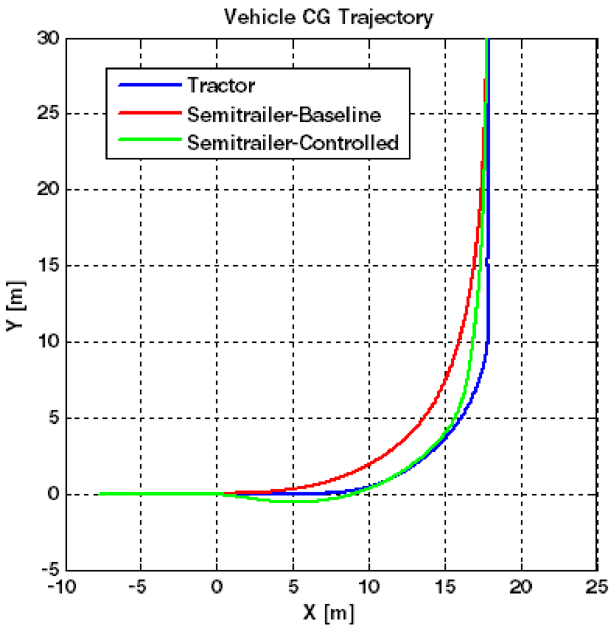

Figure 5.7 - Vehicle CG Trajectory for SPW90 Maneuver - Baseline vs. ASC

It can be seen in Figure 5.7 that the ASC could effectively reduce the PFOT for the SPW_{90} maneuver as well. As shown in Figure 5.7, the semitrailer moves a little bit outwards in the beginning of the maneuver and then gradually turns back into the curve at the end of maneuver. These happen at the times in which driver steering input is changing; such a behavior introduces more complexity for the use of ASC compared to SPW_{360} which necessitates taking this limiting maneuver as the basis of controller design. As presented in the previous section, the ASC also give quite desirable results for SPW_{360} as well.

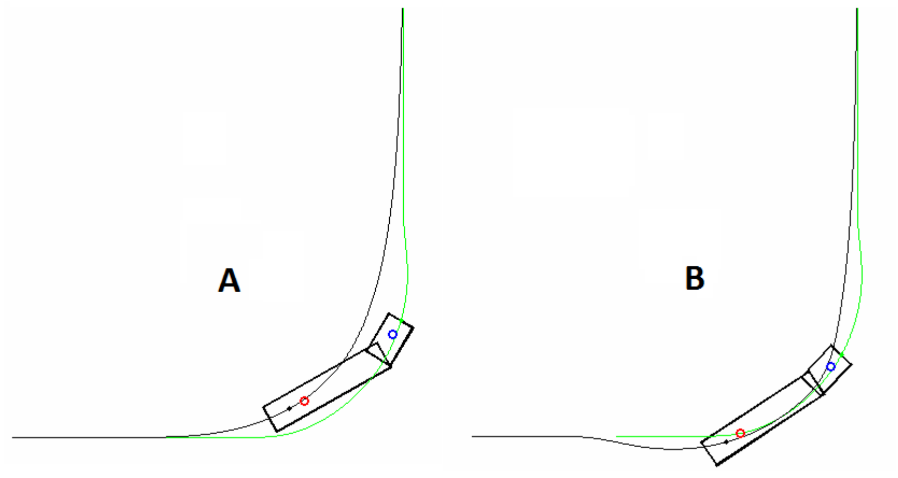

Figure 5.8 - Top-View of the Vehicle in SPW$_{90}$ Maneuver (A: Baseline Vehicle, B: Vehicle Equipped with ASC of Semitrailer)

The differences that ASC makes in SPW$_{90}$ maneuver are depicted in Figure 5.8 as a top-view of the vehicle motion. The green line indicates the path of tractors front axle while the black one corresponds to the semitrailer's axle.

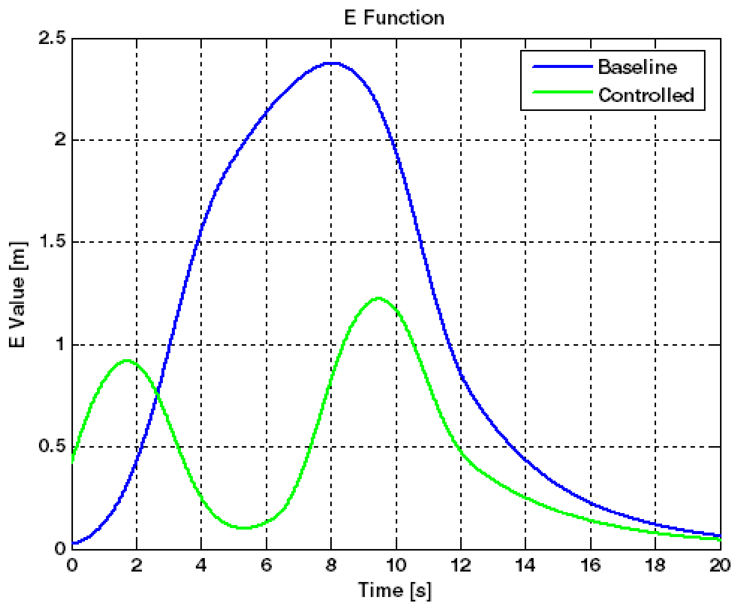

Figure 5.9 - The E Function Value as a Measure of PFOT for SPW$_{90}$ Maneuver – Baseline vs. ASC

Figure 5.9 shows how the ASC is able to reduce the E function value as a measure of PFOT. It can be seen that the peak value is reduced significantly and PFOT behavior is also change exhibiting two peaks.

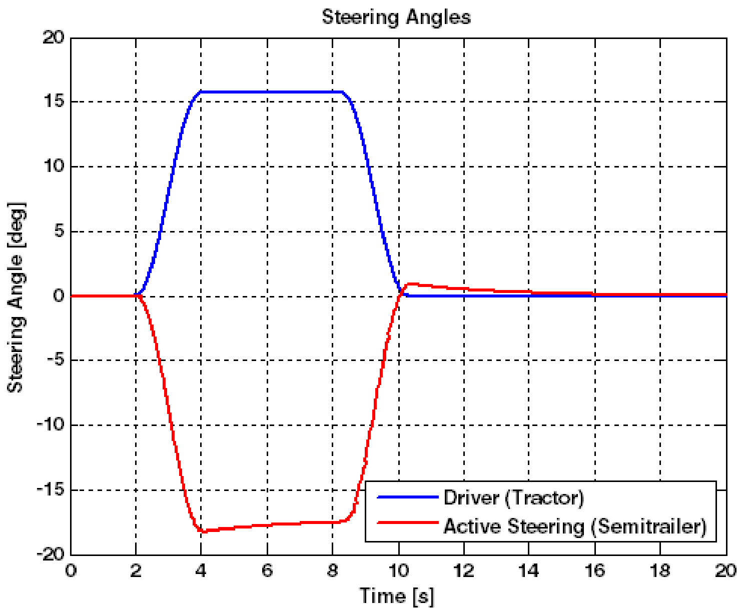

Figure 5.10 - Steering Angles for SPW_{90} Maneuver

Driver's steering input and the ASC input are applied to the vehicle as presented in Figure 5.10. It can be concluded that for both SPW_{360} and SPW_{90} maneuvers that the ASC results in a kind of mirrored steering of the semitrailer compared to driver's input.

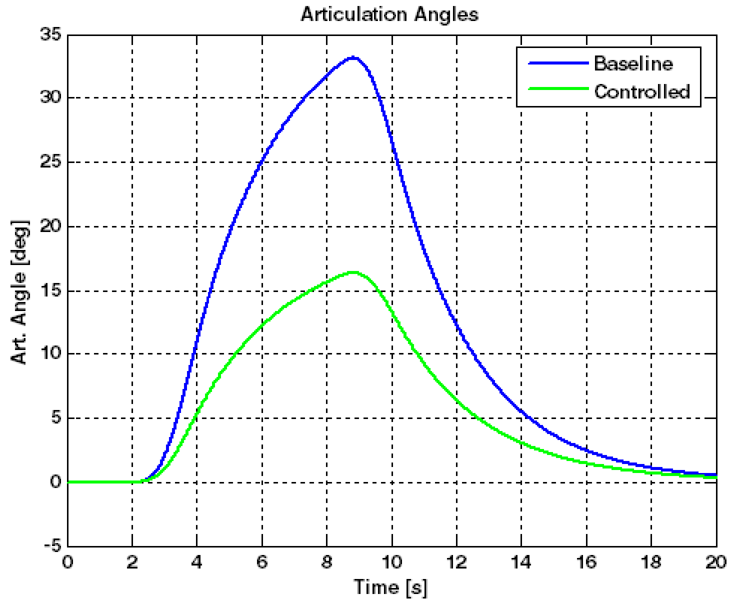

Figure 5.11 - Articulation Angles in SPW_{90} Maneuver – Baseline vs. ASC

Figure 5.11 indicates the reduction in articulation angle by about 50% with application of ASC. Again, the articulation angle values are too large to be assumed small and disprove the proper use of a linear vehicle model.

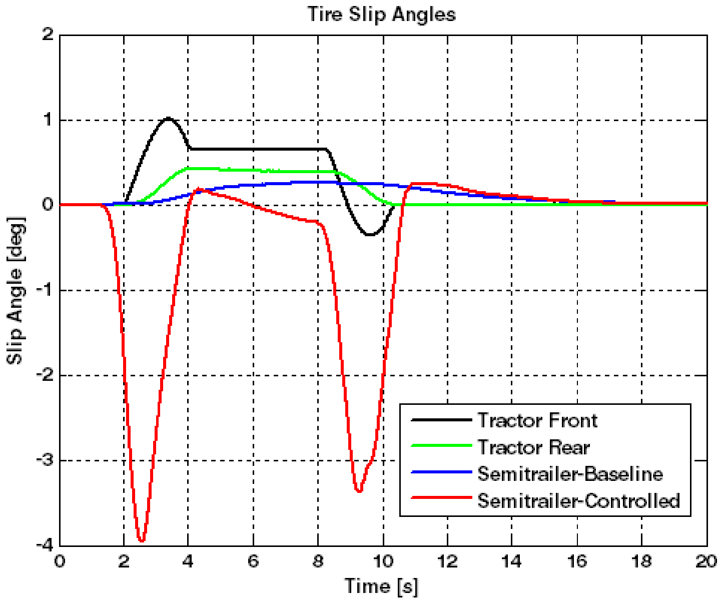

Figure 5.12 - Tire Slip Angles for SPW_{90} - Baseline vs. ASC

Tire slip angles, as depicted in Figure 5.12, are small enough to be approximated by linear formulations validating our tire model for the current maneuver study. The only points at which the semitrailer wheels experience large slip angles correspond to the change in driver's steering input due to sudden change in active semitrailer steering and simulation accuracy.

5.4. Lane Change Maneuver at 60 km/h

As mentioned previously, the ASC is going to be inactive at speeds between 40 and 70 km/h. Obviously the 360-degree and 90-degree turning maneuvers are not practically applicable to vehicle speeds higher than 40 km/h and the ASC effect will become smaller and smaller due to the inherent reduction in PFOT at that speeds. For this reason, in order to evaluate the performance of the vehicle without application of ASC, the response of the vehicle to a maneuver at 60 km/h is presented in this section. Since the turning maneuvers are not practically expected at such a vehicle speed, we shall consider a single lane change maneuver for evaluation. The model used in the following simulations is the linear model (model 2).

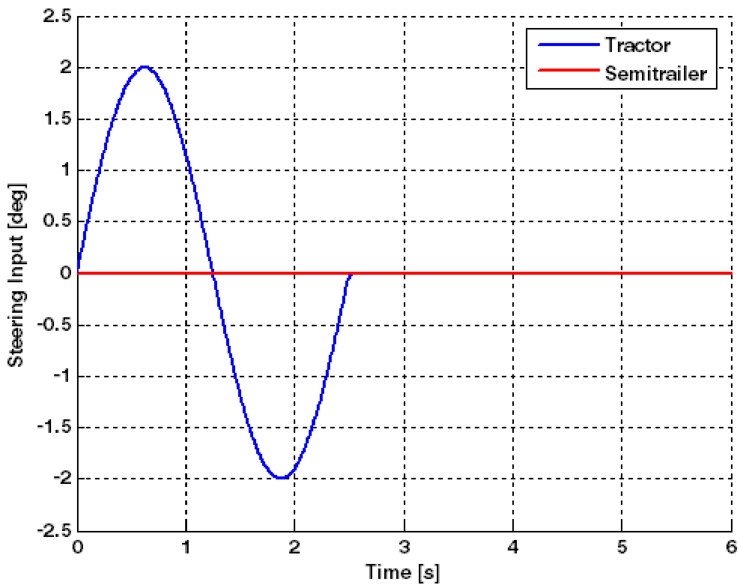

Figure 5.13 - Steering Angle Input for Lane Change Maneuver at 60 km/h

Figure 5.13 shows the presumed steering input from the driver trying to perform a single lane change maneuver.

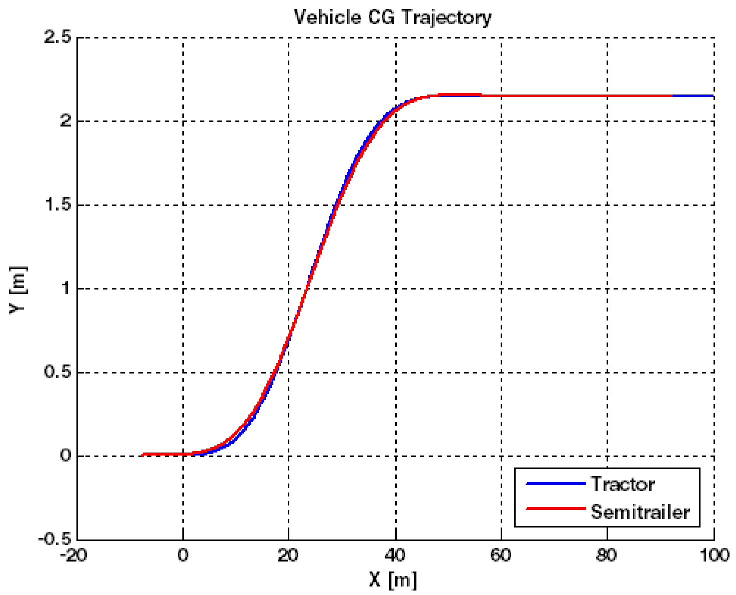

Figure 5.14 - Vehicle CG Trajectory for Lane Change Maneuver at 60 km/h

The vehicle trajectory is obtained as shown in Figure 5.14 in which there is almost no tracking error requiring additional controllers to compensate for.

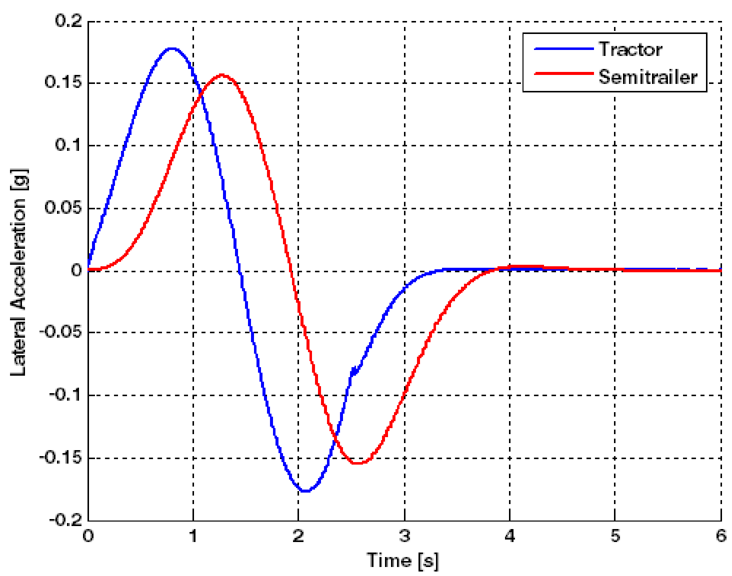

Figure 5.15 - Lateral Accelerations for Lane Change Maneuver at 60 km/h

Lateral accelerations for tractor and semitrailer are obtained as shown in Figure 5.15. There is no sign of RA and the peak lateral acceleration of the semitrailer is actually smaller than that of the tractor unit. The level of lateral accelerations being below 0.3 g also validates the use of linear model.

It should be noted that at such vehicle speeds, the effect of lateral acceleration on the tracking ability is less of concern compared to higher speeds due to the larger radii of curvature at those speeds. At the current maneuver, the difference in lateral accelerations causes very small transient off-tracking as depicted in Figure 5.14; the tracking error is such that it could be easily neglected.

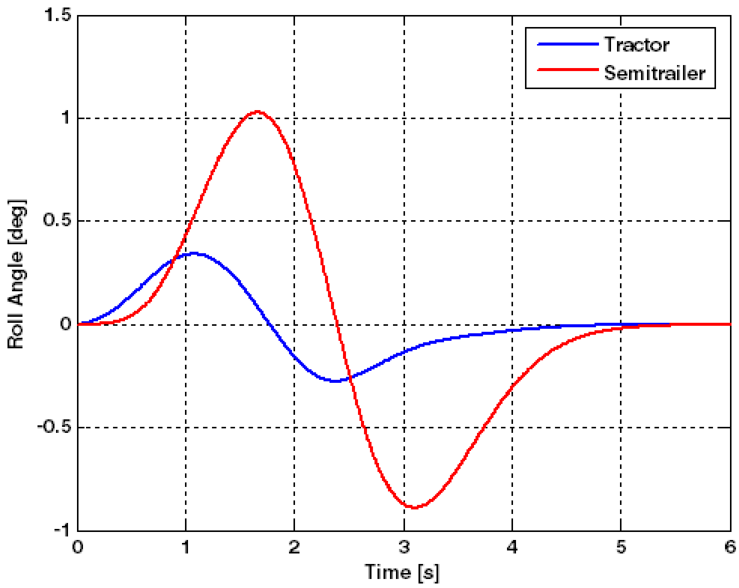

Figure 5.16 - Roll Angles for Lane Change Maneuver at 60 km/h

Figure 5.16 shows the values of the roll angles for tractor and semitrailer. It can be seen that the semitrailer experiences relatively large roll angles compared to the tractor unit due to its large weight and roll moment of inertia. However, the values are small enough to be approximated by linear formulations validating the use of linear model further.

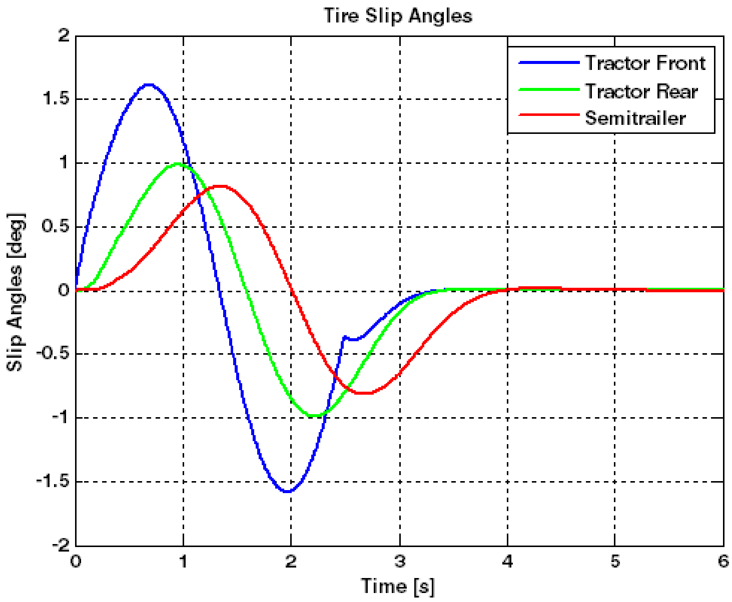

Figure 5.17 - Tire Slip Angles for Lane Change Maneuver at 60 km/h

Tire slip angles are shown in Figure 5.17. The sudden change in slip angle of the tractor's front axle is occurred due to sudden change in the driver's steering input at the end of steering process regarding the Figure 5.13.

The small values of the tire slip angles validate the use of linear model once more.

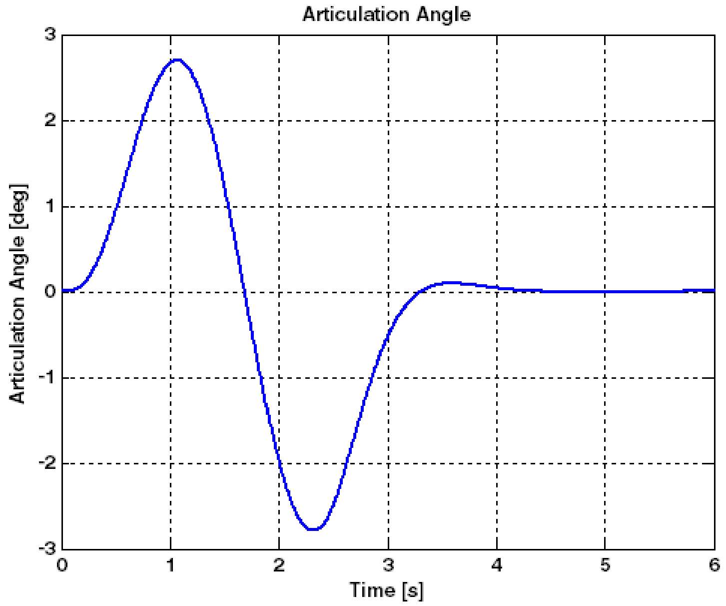

Figure 5.18 - Articulation Angle for Lane Change Maneuver at 60 km/h

Figure 5.18 demonstrates the value of articulation angle for the lane change maneuver; as shown in the figure, the values are smaller than 4 degrees which justify the use of linear vehicle model.

As a result, overall response of the vehicle for lane change maneuver at 60 km/h indicates the proper performance of the vehicle and does not bring up the need for application of ASC. The validity of the simulation is also proved as a byproduct.

5.5. Lane Change at 88 km/h (SAE J2179)

At this stage, the standard high-speed maneuver is studied for which the ASC is supposed to regulate RA ration as well as minimization of HSTO.

Figure 5.19 demonstrates the effect of ASC on the vehicle trajectory and compares it to the original baseline vehicle response. Vehicles lateral displacement is 1.46 m according to the SAE J2179 standard. The baseline semitrailer exhibits a certain level of tracking error specifically at the end of the maneuver. The tendency of the semitrailer unit to travel outside the curve is depicted in the figure. On the other hand, the vehicle equipped with ASC of semitrailer behaves quite desirable from the tracking point of view. The tractor's and semitrailer's centers of gravity almost follow a common path with zero HSTO.

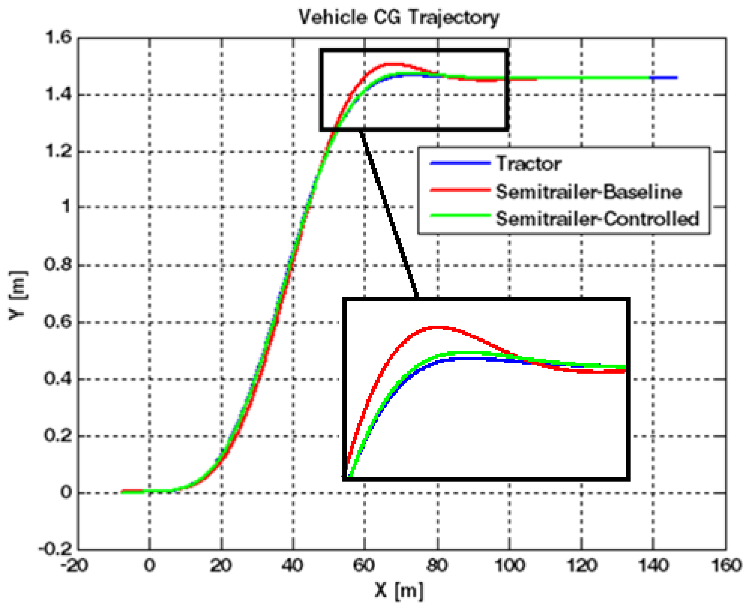

Figure 5.19 - Vehicle CG Trajectory for SAE J2179 Maneuver - Baseline vs. ASC

Figure 5.20 show the lateral acceleration response of the vehicle. It can be seen that the lateral acceleration of the semitrailer has been reduced and damped by means of ASC compare to the baseline vehicle. The RA ratio is reduced to 1.0 as intended by the LQR design. Values of lateral accelerations are also in the expected region for the bicycle model to be valid.

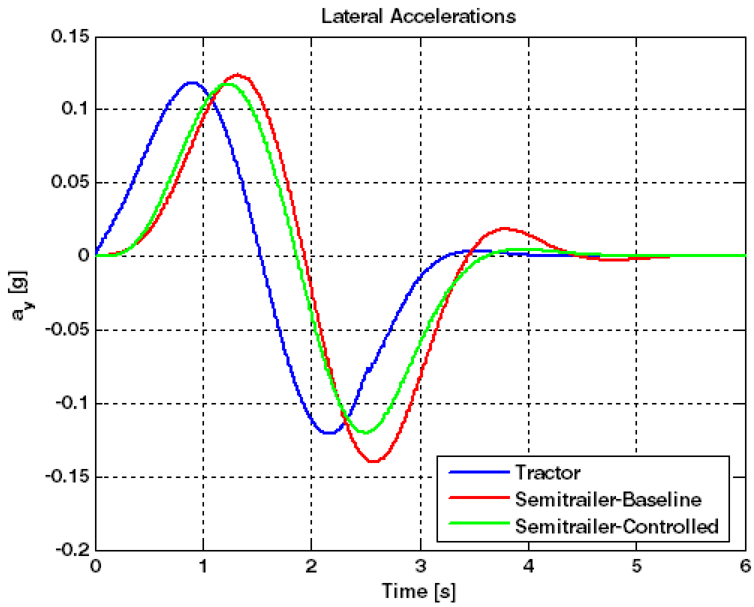

Figure 5.20 - Lateral Accelerations for SAE J2179 Maneuver - Baseline vs. ASC

As shown in Figure 5.21, the roll motion of the semitrailer unit is regulated a little as a result of lateral force reduction which is also seen in the reduction of lateral acceleration of the semitrailer. But the reduction in roll angle is not significant and seems to require more consideration if the designer intends to reduce the roll motion effectively.

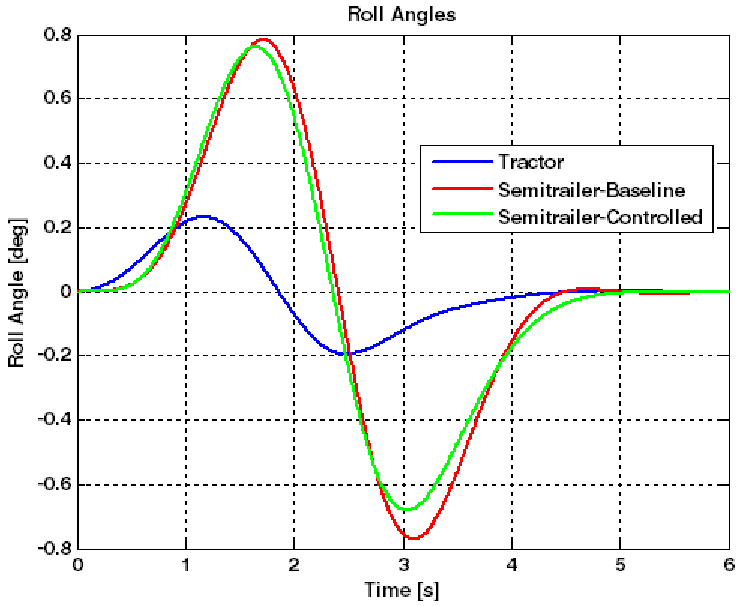

Figure 5.21 - Roll Angles for SAE J2179 Maneuver - Baseline vs. ASC

The sine-wave steering input is assumed for the driver to negotiate the lane change maneuver as shown in Figure 5.22. The ASC input is applied with a phase difference with respect to the driver's input due to the inertia of the vehicle and the inherent delayed response of the semitrailer which could also be noted in the lateral acceleration plot. The small angles assumption is also met in the results.

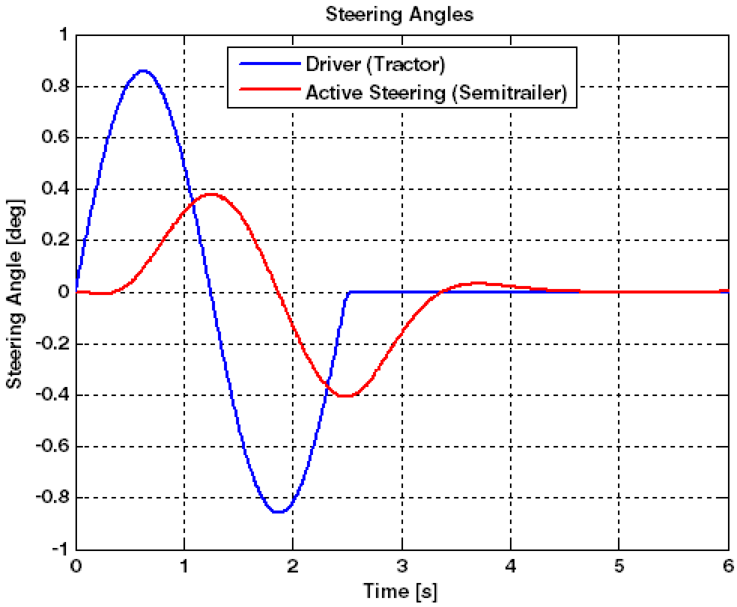

Figure 5.22 - Steering Angles for SAE J2179 Maneuver

The tire slip angle values are demonstrated in Figure 5.23. The ASC has reduced the slip angle for semitrailer' axle which, in fact, results in decreased lateral tire forces and lateral acceleration of the semitrailer unit. The range of slip angles is below 4 degrees, thus the use of linear tire model is justified.

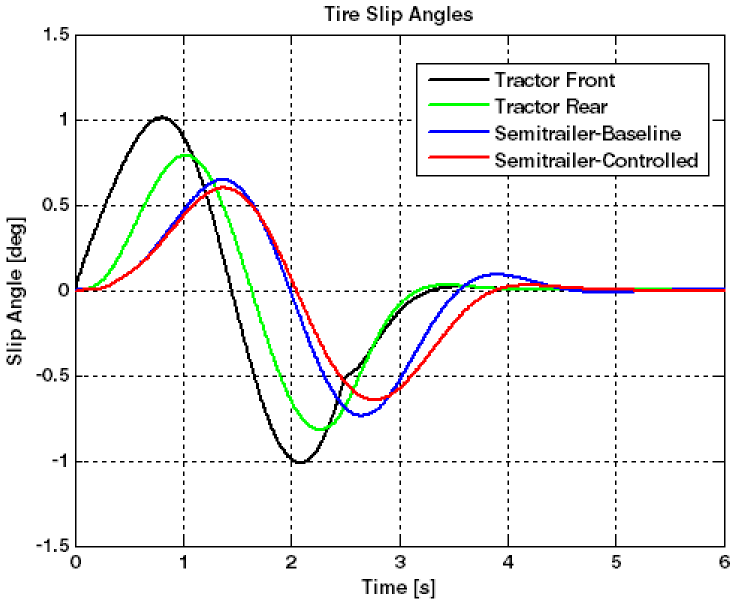

Figure 5.23 - Tire Slip Angles for SAE J2179 Maneuver, Baseline vs. ASC

Figure 5.24 represents the articulation angle value for vehicles under the SAE J2179 maneuver. It can be seen that the ASC slightly damped the oscillation in articulation angle response which corresponds to better yaw stability of the semitrailer. The values of articulation angle also are small enough to validate the use of linear vehicle model.

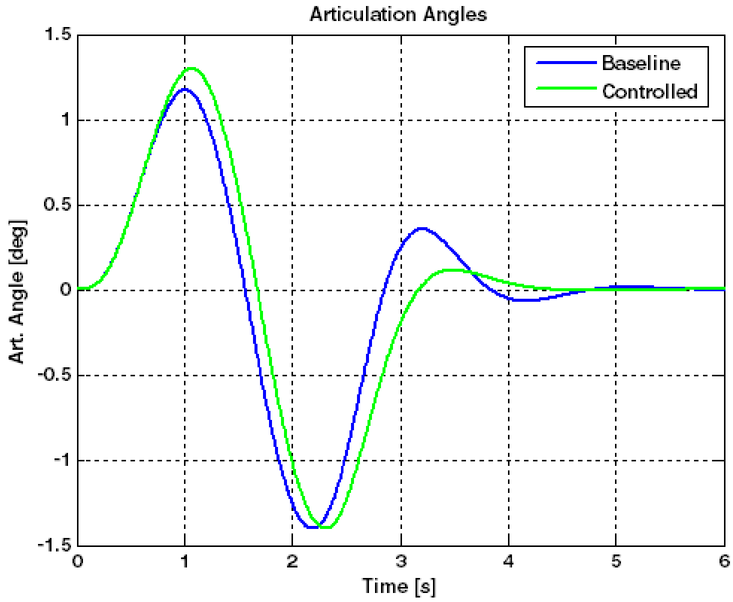

Figure 5.24 - Articulation Angle for SAE J2179 MAneuver, Baseline vs. ASC

5.6. Sinusoidal Steering at 88 km/h

In order to evaluate the performance of the ASC system on other potential vehicle maneuvers, a rather severe maneuver is selected for simulation in which the driver applies a sinusoidal steering input to the vehicle as if to simulate a severe driving condition in which the driver is close to lose control of the vehicle. The simulation results will demonstrate the performance of the proposed control method in a more general situation.

Obviously, in such a maneuver the driver does not tend to follow a specific path and his/her target is only to stabilize the vehicle. For this reason, only some of the vehicle responses which are of concern are presented in this section.

Figure 5.25 shows the lateral acceleration response of the vehicle units during the maneuver. The lateral acceleration of semitrailer is reduced compared to baseline vehicle decreasing the risk of rollover under such severe condition. The values of lateral acceleration are also under 0.3 g which evaluates the use of bicycle model.

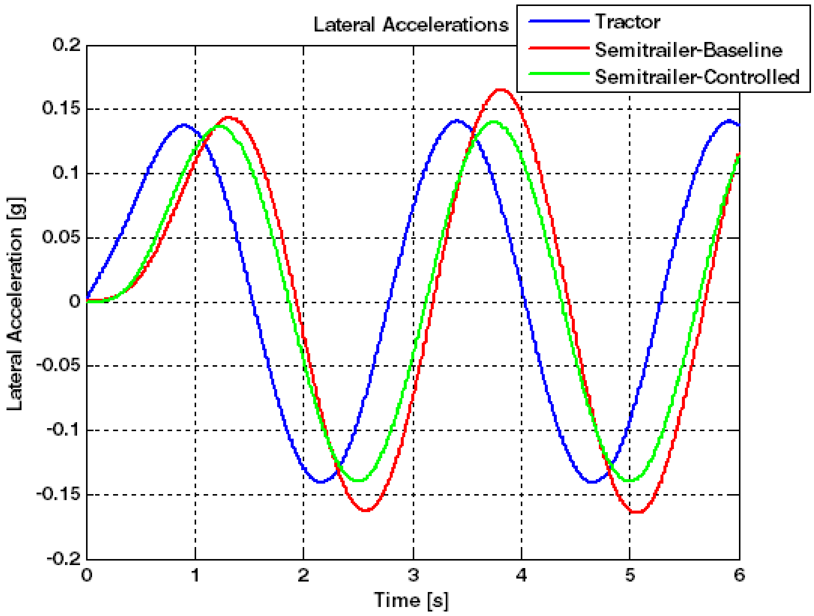

Figure 5.25 - Lateral Accelerations for Sinusoidal Steering Input, Baseline vs. ASC

As a result of lateral acceleration reduction for the semitrailer, the roll angle for this unit is also lowered to some extent as shown in Figure 5.26, but the effectiveness of the ASC on roll motion reduction does not seem to be significant. Small roll angle assumption is also validated in this figure.

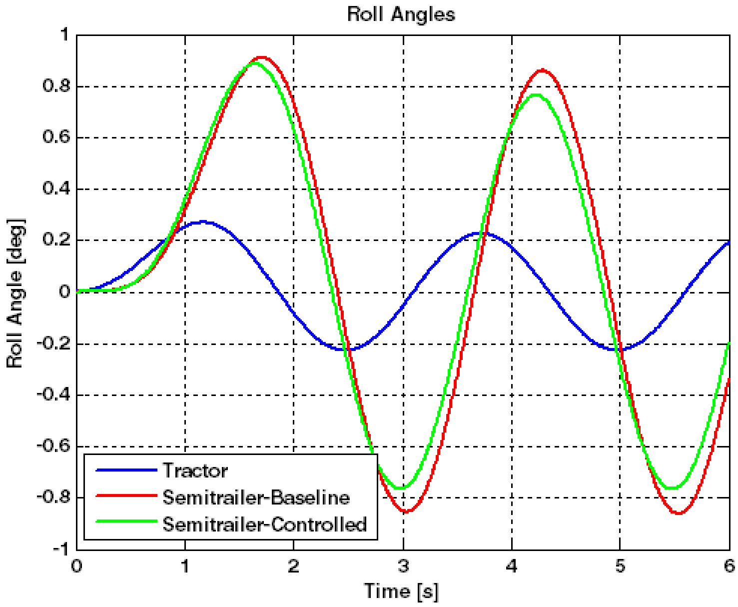

Figure 5.26 - Roll Angles for Sinusoidal Steering Input, Baseline vs. ASC

Steering input of the driver and that of the semitrailer ASC system are shown in Figure 5.27. The controller shows smooth response although the severe maneuver conditions exist. The small angles assumption is also met in the results.

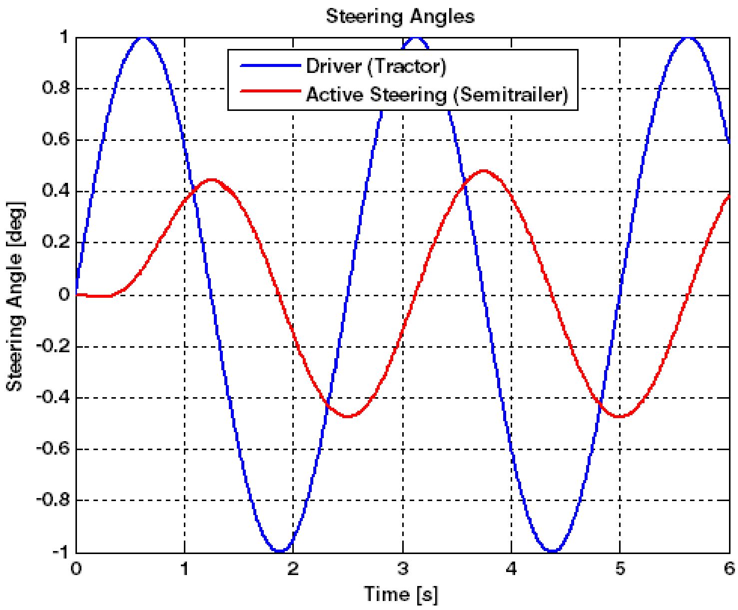

Figure 5.27 - Steering Angles for Sinusoidal Steering Input

Tire slip angles values are presented in Figure 5.28. The results are similar to the previous observations from the ASC system, which result in the reduction of lateral forces and lateral accelerations, consequently. The values do not deviate from the assumed limits validating the use of linear tire model.

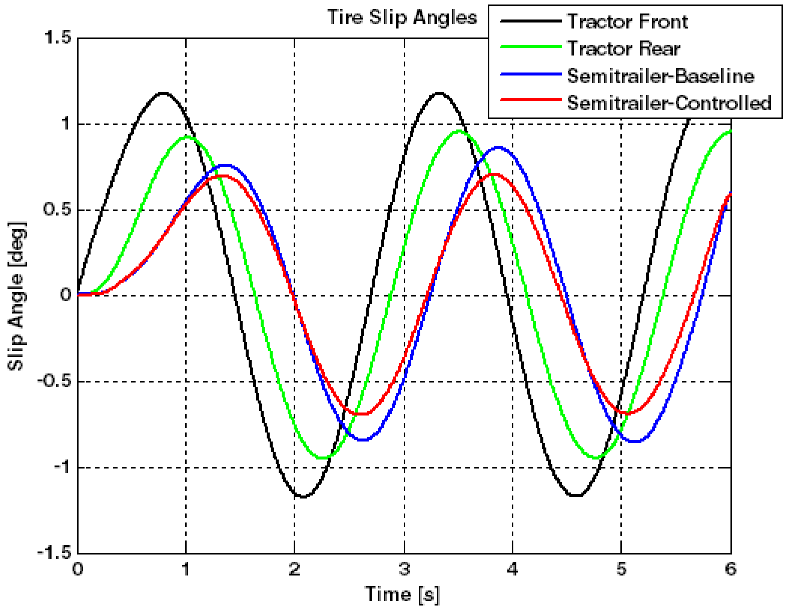

Figure 5.28 - Tire Slip Angles for Sinusoidal Steering Input, Baseline vs. ASC

5.7. Lane Change at 120 km/h

Up to this point, the performance of the semitrailer ASC system has been evaluated for the low-speed, intermediate-speed and standard high-speed maneuvers, but it is still not clear if the ASC would perform desirable at even higher vehicle speeds. In order to evaluate the control system for the whole range of speeds for which the proposed feedback matrix is developed, the results of another high-speed lane change maneuver is presented in this section corresponding to vehicle speed of 120 km/h as the highest vehicle speed for which the feedback gains are optimized.

The trajectory of the vehicle units' CGs is presented in Figure 5.29. It can be seen that in such a rather sever maneuver, the semitrailer exhibits more oscillatory response compared to lower speeds. The value of HSTO is also seen to be larger which brings up many safety issues. On the other hand, the use of ASC for semitrailer has been able to improve the HSTO significantly such that the vehicle units almost share a common trajectory with almost no oscillations in the controlled vehicle.

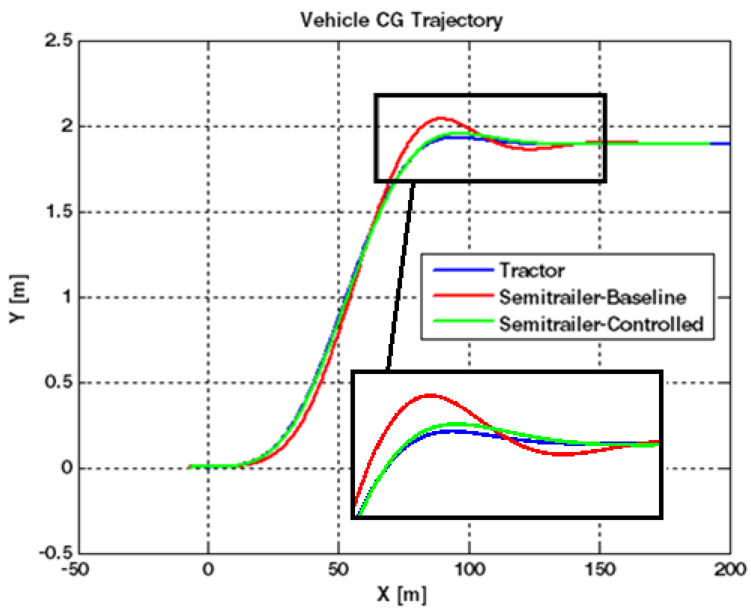

Figure 5.29 - Vehicle CG Trajectory for 120 km/h Lane Change, Baseline vs. ASC

Figure 5.30 shows how dramatically the ASC has been able to reduce the peak lateral accelerations of the semitrailer unit during the maneuver. The lateral acceleration response of the semitrailer has also been significantly damped by use of ASC. The RA ratio of 1.0 is almost achieved with acceptable level of accuracy as shown in the figure. The values of lateral acceleration are also below the threshold designated for validity of bicycle model usage.

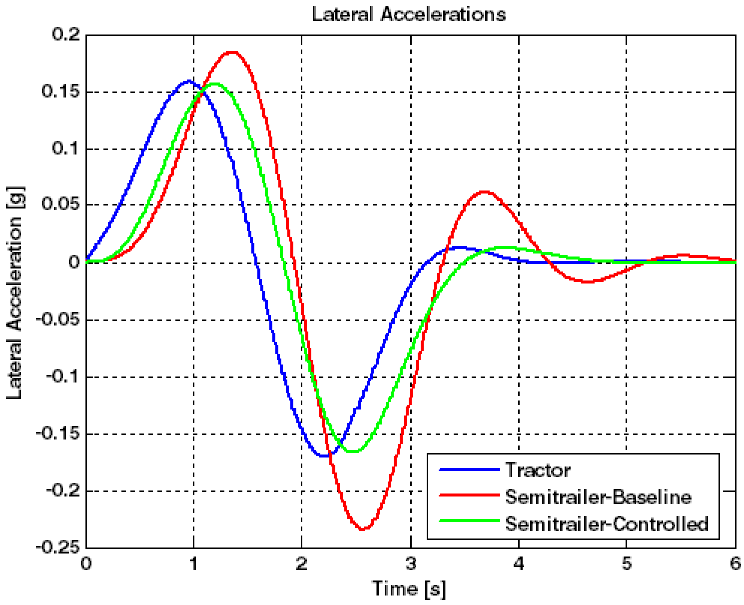

Figure 5.30 - Lateral Accelerations for 120 km/h Lane Chnage, Baseline vs. ASC

Figure 5.31 shows the roll angles generated during the maneuver. It can be seen that the ASC has been able not only to reduce the peak roll angle of semitrailer, but also settles the oscillations in the semitrailer roll response. It can be concluded that the reduction is roll angle is more explicit at this vehicle speed compared to standard SAE J2179 maneuver. The values of roll angles also comply with the assumption of small roll motions.

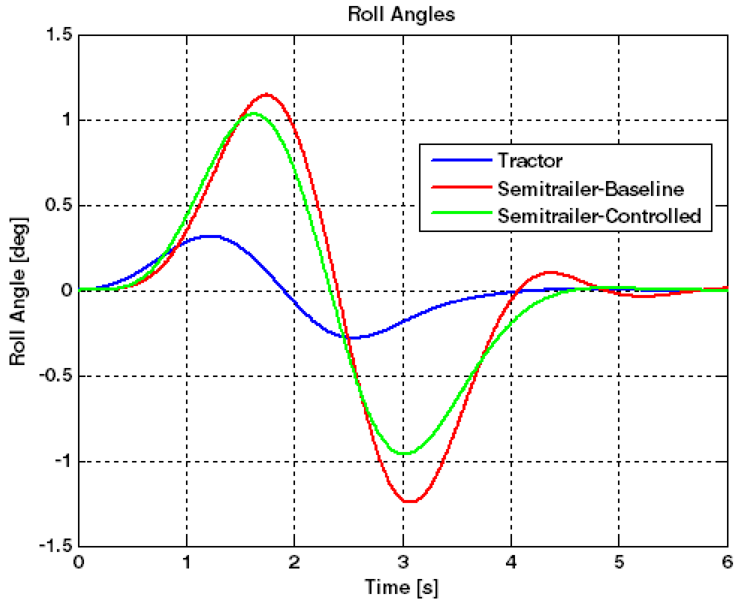

Figure 5.31 - Roll Angles for 120 km/h Lane Change, Baseline vs. ASC

As shown in Figure 5.32, a relatively large control input is applied to the vehicle to regulate the motion. An additional peak at the end of ASC input is also observed which corresponds to the end of the maneuver in which the semitrailer tends to deviate from the path due to its lateral inertia. The small angle assumption is met though.

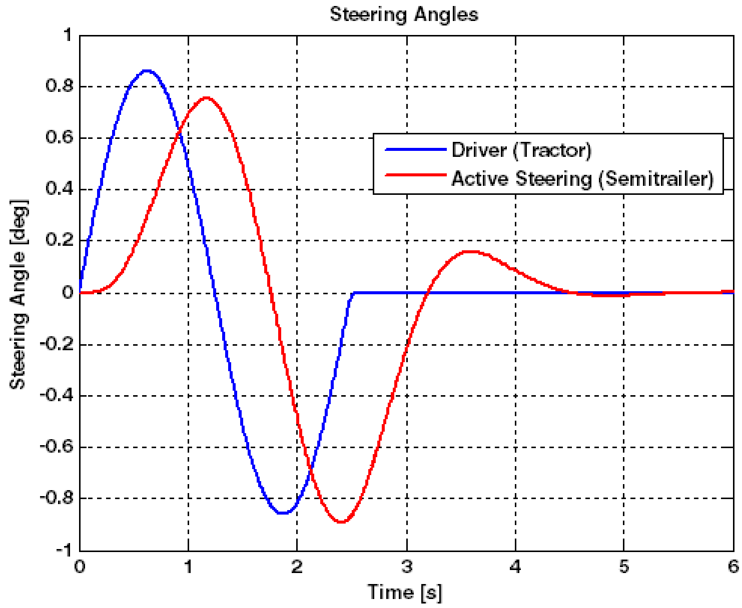

Figure 5.32 - Steering Angles for 120 km/h Lane Change, Baseline vs. ASC

The tire slip angles are shown in Figure 5.33. The results show more level of effectiveness for the 120 km/h lane change maneuver. The range of angles validates the use of linear tire model again.

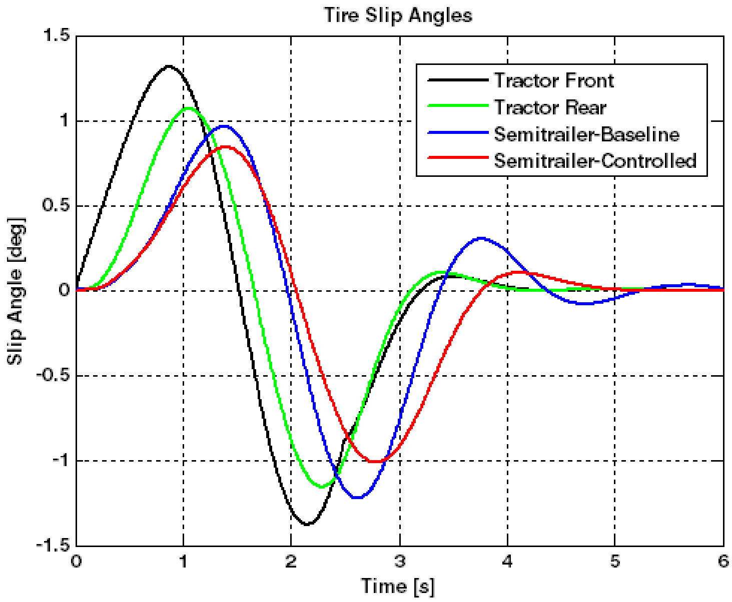

Figure 5.33 - Tire Slip Angles for 120 km/h Lane Change, Baseline vs. ASC

Increase of articulation angle peaks as a result of ASC application, as shown in Figure 5.34, are due to the prevention of semitrailer from travelling outwards by application of the ASC. The oscillations in articulation angle are also damped to some extent resulting in better vehicle response. The small angle assumption is also met for the articulation angle.

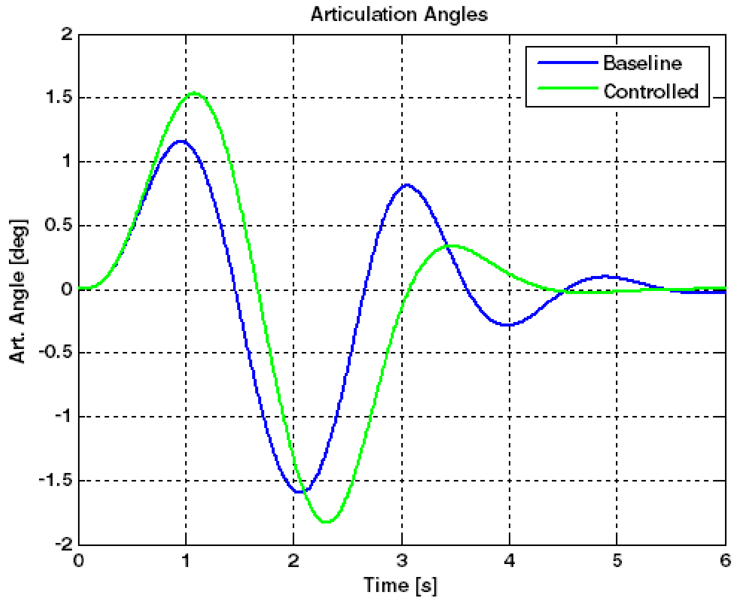

Figure 5.34 - Articulation Angle for 120 km/h Lane Change, Baseline vs. ASC

5.8. Lane Change at 88 km/h (SAE J2179), Comparing Additional Suggested Control Strategies

As concluded from the results of previous simulations, the ASC system is able to regulate the yaw and lateral motion of the semitrailer at high-speeds, but the effectiveness of the system on roll minimization is not that significant. In the following section, the results obtained by the application of those suggested strategies are going to be compared.

In the following, results from application of modified control strategies are presented and they are compared against each other for the SAE J2179 standard maneuver. Note that ASC1 refers to the initially developed control strategy, ASC2 refers to the second active steering control with modified optimization criterion, and ASC+ARC refers to the active steering control

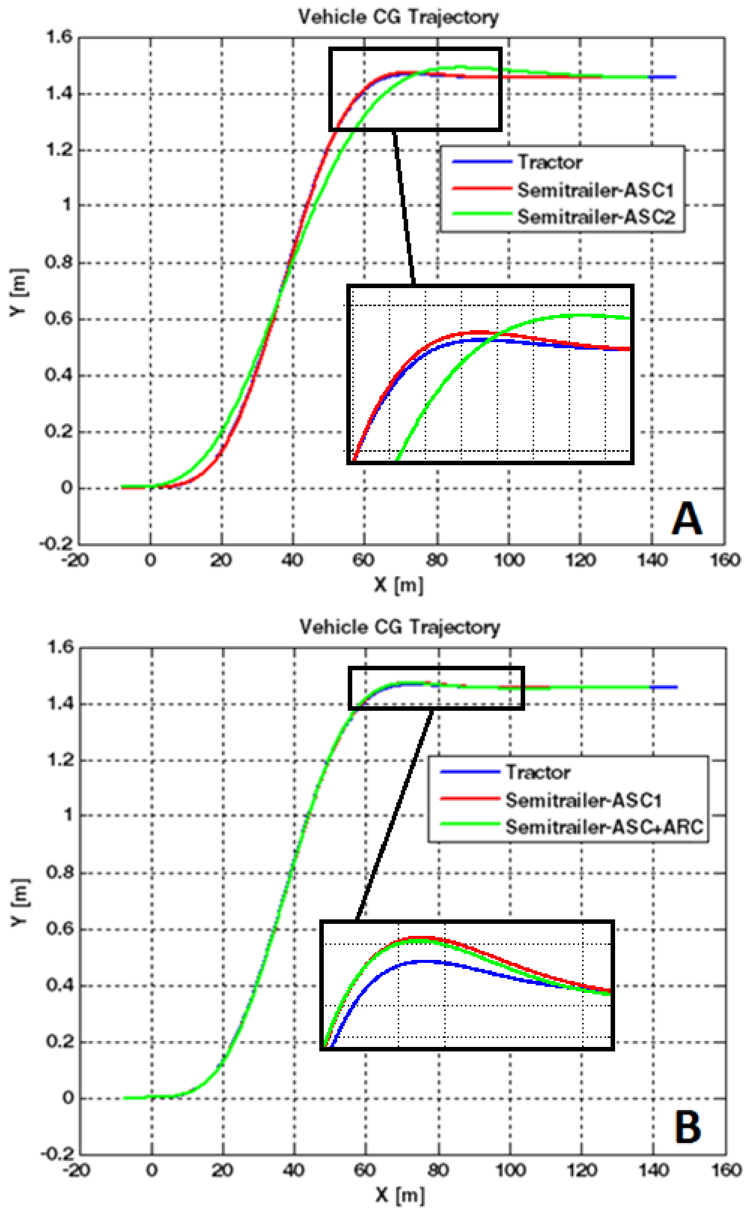

Figure 5.35 – Vehicle CG Trajectory for SAE J2179 Maneuver – A: ASC1 vs. ASC2, B: ASC1 vs. ASC+ARC

Trajectories of the vehicles are shown in Figure 5.35 with all three control methods. It can be seen that the semitrailer travels inwards, except the end of maneuver, with respect to the tractor for ASC2 due to RA ratio being less than

1.0 and the ASC+ARC system does not show any significant difference from the ASC1 system.

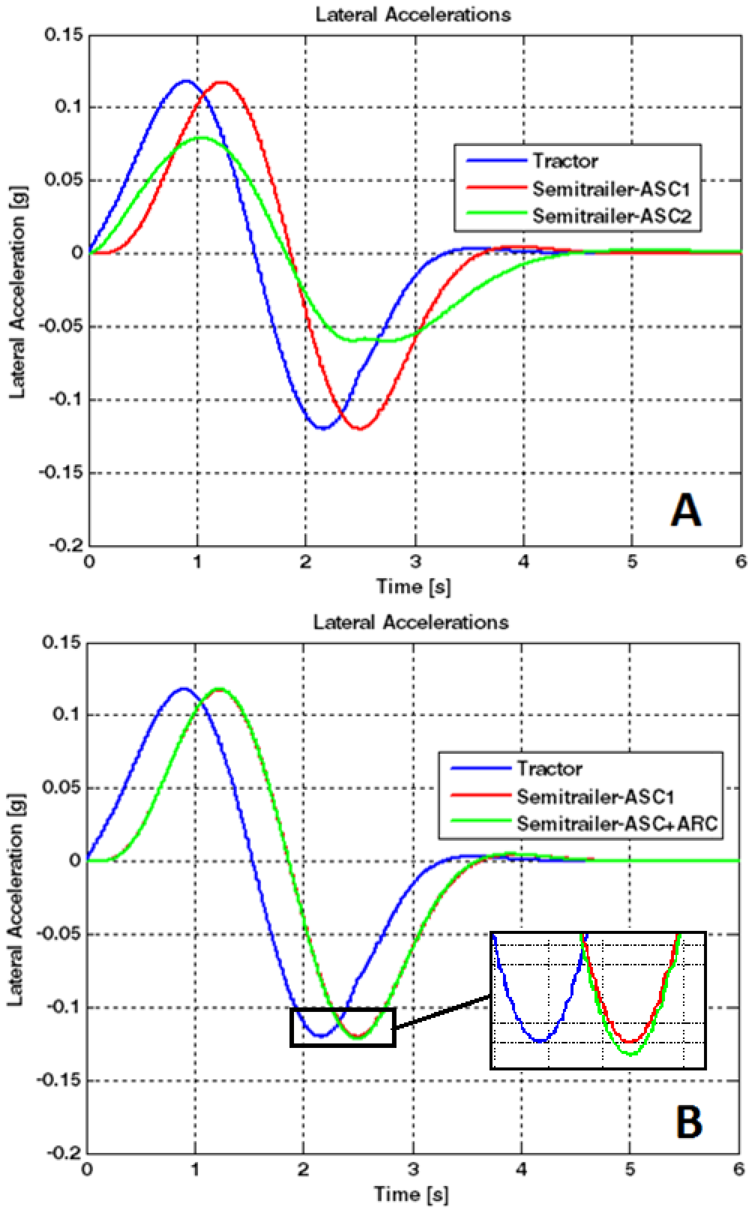

Figure 5.36 - Lateral Accelerations for SAE J2179 Maneuver – A: ASC1 vs. ASC2, B: ASC1 vs. ASC+ARC

Lateral acceleration response of the vehicle is shown in Figure 5.36 for all types of the control system. The reduction in RA ratio with application of ASC2 is seen to be quite significant while the ASC+ARC system almost does not change the lateral acceleration behavior of the vehicle as expected.

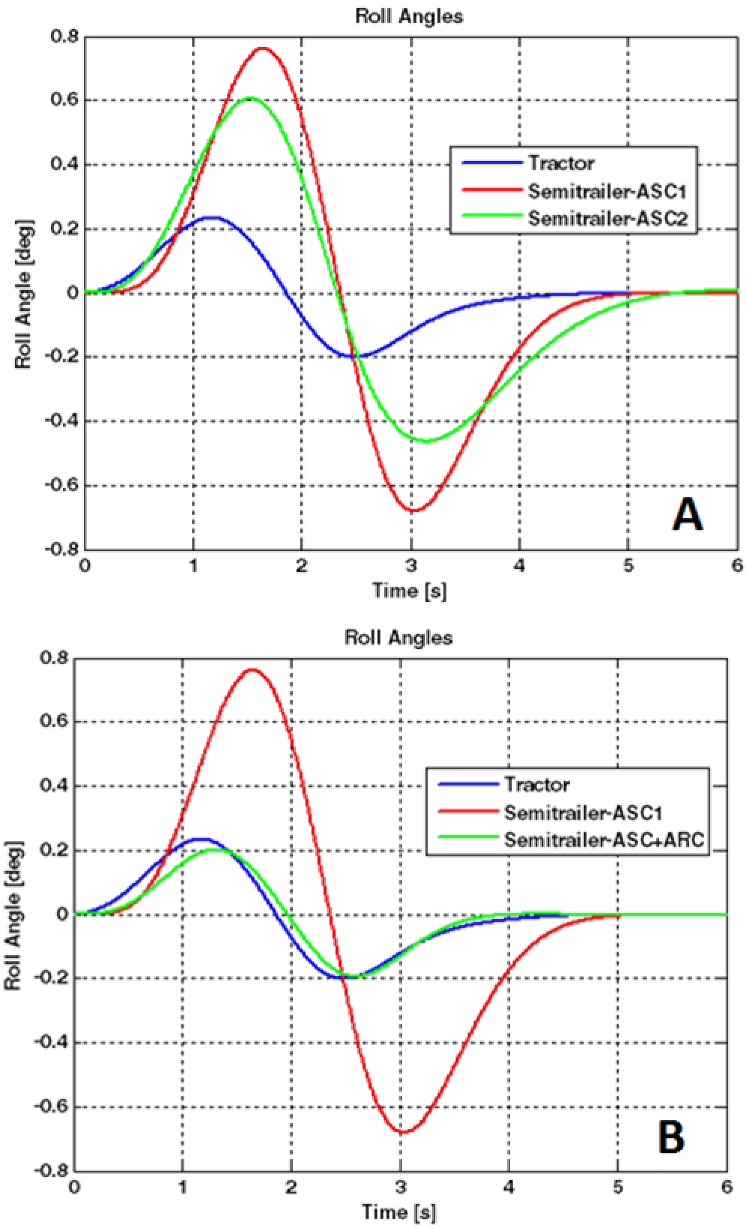

Figure 5.37 - Roll Angles for SAE J2179 Maneuver – A: ASC1 vs. ASC2, B: ASC1 vs. ASC+ARC

Figure 5.37 shows the roll motion response of the vehicles equipped with all three control systems. It can be seen that the ASC2 algorithm has been able to reduce the roll angle of semitrailer further compared to ASC1 as expected. The ASC+ARC system has been able to dramatically reduce the semitrailer roll angle much more than the ASC2.

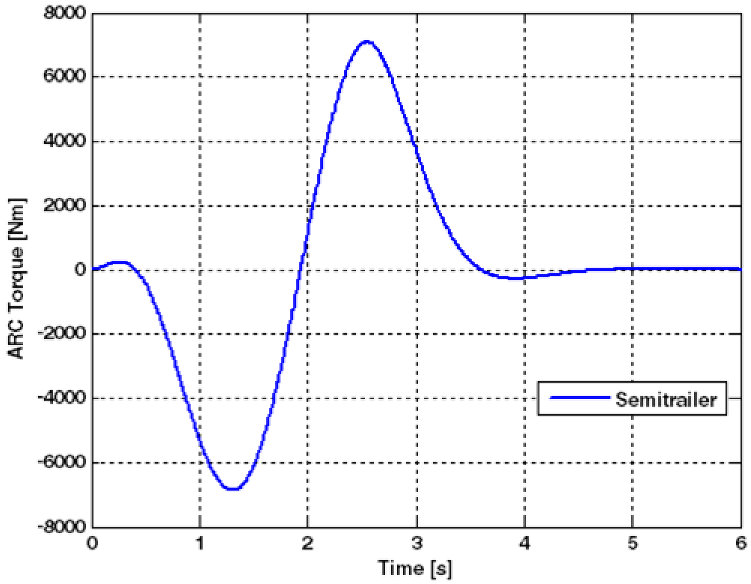

Figure 5.38 - ARC Torque for SAE J2179 Maneuver with Application of ASC+ARC

Figure 5.38 shows the value of the ARC torque as the additional control input to the system. The values obtained actually fit into an acceptable range of such conventional torques which are usually generated by hydraulic actuators [46], [70]. Generation of such values, as our criterion, eliminates the need for applying any further limitation for this control input. In fact, the structure of the LQR does not allow larger ARC torques to be applied on the vehicle inherently.

CHAPTER 6

CONCLUSIONS

6.1. Results and Discussions

Tractor-semitrailers, as the most common type of AHVs in use, exhibit poor maneuverability at low-speed maneuvers with large PFOT values while having undesirable high-speed characteristics such as HSTO and RA the latter being a main cause of increased semitrailer rollover risk.

In order to provide solutions to such instabilities and unfavorable behaviors of the AHVs, a number of passive semitrailer steering systems are developed and studied in that past including command steer, self-steer, and pivotal bogie systems each of which has advantages over the other ones, but none of them has been accepted as a common steering system for both low and high-speed maneuvers; thus the need for active steering controllers arise as detailed in chapter 1.

Previous research on the topic include introduction of active semitrailer steering systems along with other control approaches such as differential braking and active roll control of AHVs among which the ASC for semitrailer is arguably the most effective one in the yaw-plane motion as described in chapter 4.

Among existing vehicle mathematical models, the best suited models are selected and altered accordingly with the needs and assumptions of this research. Two versions of the bicycle model are considered one with small angles assumption leading to linear equation of motion for high-speed simulation and the other one without small angles assumption, except for the tire slip angles, which is used for low-speed simulations. The choice of proper reference model is of great importance to get reliable results which has been the motivation to make difference between low-speed and high-speed cases in this study.

A MATLAB user-friendly simulation environment has been developed for this study which eases the simulation procedure, enabling the designer to load/save data, set the steering input and vehicle speed, opt for the desired vehicle model, etc. Such simple simulation environments provide the designer with readily available simulation results, as well as potential parameter selection which eliminate the need for complicated multi-body dynamic software to run over and over, saving time and energy for the designer to manipulate data easily and obtain proper solutions more efficiently in both academic and industrial applications.

It is greatly important to accredit the vehicle models being used to make sure if they are giving reliable results. This can be done by field-tests which are extremely expensive along with the availability of the facilities also being another issue. The complications in data measurements and analysis also bring up additional work load for the designers. An alternative approach for vehicle model validation is the use of highly trusted industrial vehicle dynamics simulation software such as TruckSim® which can help saving the designers time while eliminating the concerns about validity of the models being used. Obviously, this decision depends on the scope of the work and the level of details included in the vehicle models. As a result, another outcome of this study is to validate some simplified vehicle models with corresponding necessary assumptions.

Different control algorithms such as fuzzy logic control, sliding mode control and linear quadratic regulator techniques has been introduced for the semitrailer steering systems in the literature. The LQR method introduced in this study turns out to be simpler in structure while providing desirable results. Design and application of state feedback control is easier compared to other methods introduced in literature such as model-based control techniques and nonlinear approaches. It also alleviates the need for complex mathematical analysis. Considerations in the design of the LQR structure have provided favorable simplifications while maintaining the accuracy and control quality by eliminating unnecessary elements.

The optimization method applied in this study is also a relatively easy approach while providing an alternative way of describing the cost functions in such a way that the designer is not limited to choose quadratic forms of the function which might be impossible or time consuming approaches. It is further concluded that the QPSO would be a useful tool for the tuning of LQR weighting matrices in dealing with such dynamic systems.

In order to provide suitable vehicle response across a range of vehicle speeds, the feedback tuning procedure must be performed for several vehicle speeds due to the change in vehicle response at different speeds. As an important conclusion of this research, the LQR semitrailer steering is adjusted for several vehicle speeds from 10 km/h up to 120 km/h as the probable speed range, with interpolation and extrapolation for other vehicle speeds. It is also concluded that the PFOT minimization must be considered for speeds below 40 km/h and the RA regulation for speeds above 70 km/h based on the prescribed vehicle parameters.

As a result of ASC application, the overall low-speed response of the vehicle has improved significantly with acceptable levels of tracking error in the beginning of the turning maneuvers. It is worth to mention that the low-speed steering control might be more efficient by means of some tracking strategies such as those introduced in [6] for any type of transient vehicle maneuvers resulting in precise tracking ability, but the simplicity of the LQR semitrailer steering might be preferred over such complicated control systems from the economical and feasibility perspectives.

The high-speed performance of the LQR semitrailer steering control is highly desirable according to the simulation results specifically in yaw-plane motion and tracking perspective. The significant improvement that the LQR semitrailer steering provides for high-speed maneuvers totally outperforms the other complicated control strategies such as tracking algorithms and nonlinear analytical approaches with relatively small and smooth control input resulting in low energy requirement. The system also performs well for other high-speed maneuvers such as the severe sinusoidal input maneuver. Desired performance of the system also validates the rational simplifications made into the control system design and proves them to be carefully made. The only imperfection found in the LQR semitrailer steering response is that it could not reduce the semitrailer's roll angle significantly.

Desired reduction in the roll motion could be reached by further reduction of the RA at the expense of larger HASTO value. In order to compensate for this problem, another control input is added to the system by application of an active anti-roll bar. A roll control torque is introduced as another part of the LQR system for further reduction of the roll motion without degrading the perfect tracking ability achieved by active steering of the semitrailer wheels. A comparison between these two modified LQR approaches leads us to the conclusion that the ASC2 approach is able to reduce the roll motion as well as the RA, but with a slightly undesirable HSTO value. The ASC+ARC approach, on the other hand, has been able to reduce the roll motion dramatically while letting the semitrailer to follow the leading unit perfectly, but with more mechanical system complications and higher energy consumption. The final selection between these two control approaches, if the roll motion reduction is of significance, could be done regarding the vehicle design needs and the cost considerations as well as the corresponding compliance standards and regulations.

6.2. Suggestions for Future Researches

As a greatly important consideration, one should regard carefully that the single equivalent wheel assumption may not be realistic for nearly zero speeds and during tight turning maneuvers due to the geometrical limitation of the multi-axle vehicle configuration. This may increase slip angles in semitrailer's axles as stated in chapter 2, leading to tire and road scrub and large tire forces. In other words, the semitrailer's axle group may exhibit resistance against the turn in case of multi-axle semitrailer. Hence, the low speed considerations of this study are mainly valid for single axle semitrailers or multi-axle semitrailers not under extremely tight cornering maneuvers. Consequently, one suggestion for future researches could be the consideration of multi-axle semitrailers without any assumption of equivalent wheel configuration to obtain more accurate low-speed results.

Other potential control approaches for ASC+ARC combination may be studied and compared to the current work in order to limit the ASC system only to regulate the RA ratio and keep is as close as possible to 1.0 for best tracking ability.

Detailed hardware and real-life application of the proposed method on a proper test vehicle may be the topic of a practical future study.

Application of the linear model based controller design on some highly nonlinear and severe vehicle conditions such as sudden changes in friction, rollover threshold, etc. may be the topic of a future study to evaluate the performance of the LQR on nearly unstable vehicle motions and investigate whether the LQR improves such conditions or disturbs them further.

For more accurate rollover risk analysis, investigation of the lateral load transfer ratio and the rollover threshold are required which can be included in another more detailed study.

As stated, in this study, the vehicle forward speed is assumed to be constant during the maneuvers; a useful study may be conducted on the effect of variable vehicle speed for better and more realistic controller design and also to study the effects of such control applications on the forward speed of the vehicle. This can be done by consideration of an additional degree of freedom for longitudinal dynamics of the vehicle.

Finally, additional dynamic effects such as the side wind effect and other active and passive aerodynamic considerations may be investigated as well as other potential lateral and yaw regulation methods such as the combination of different control methods in details to make a comparison between all feasible and possible control combinations.

REFERENCES

[1] M. M. Islam, “Design synthesis of articulated heavy vehicles with active trailer steering systems,” *M.Sc. Thesis, Univ. Ontario Inst. Technol. Oshawa, Ontario, Canada*, 2010.

[2] P. Fancher and C. Winkler, “Directional performance issues in evaluation and design of articulated heavy vehicles,” *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 45, no. 7–8, pp. 607–647, 2007.

[3] H. Peng and B. Chen, “Rollover warning of articulated vehicles based on a time-to-rollover metric,” *Proc. 1999 ASME Int. Congr. Expo. Knoxville, TN*, 1999.

[4] “Traffic Safety Facts 1998: A Compilation of Motor Vehicle Crash Data from the Fatality Analysis Reporting System and the General Estimates System,” *Natl. Cent. Stat. Anal. U.S. Dep. Transp. Washington, DC*, 1999.

[5] M. M. Islam, “Parallel Design Optimization of Multi-Trailer Articulated Heavy Vehicles with Active Safety Systems,” *Ph.D. Thesis, Univ. Ontario Inst. Technol. Oshawa, Ontario, Canada*, 2013.

[6] B. A. Jujnovich and D. Cebon, “Path-Following Steering Control for Articulated Vehicles,” *J. Dyn. Syst. Meas. Control*, vol. 135, 2013.

[7] B. Jujnovich and D. Cebon, “Comparative Performance of Semi-Trailer Steering Systems,” *7th Int. Symp. Heavy Veh. Weight. Dimens. Delft, Netherlands, Eur. June 16 – 20*, 2002.

[8] M. F. J. Luijten, “Lateral Dynamic Behaviour of Articulated Commercial Vehicles,” *Master's Thesis, Eindhoven Univ. Technol.*, 2010.

[9] S. Kharrazi, "Steering based lateral performance control of long heavy vehicle combinations," *Ph.D. Thesis, Chalmers University of Technology, Gothenburg, Sweden*, 2012.

[10] C. Cheng, R. Roebuck, A. Odhams, and D. Cebon, “High-speed optimal steering of a tractor–semitrailer,” *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 49, no. 4, pp. 561–593, 2011.

[11] P. Fancher, C. Winkler, R. Ervin, and H. Zhang, “Using Braking To Control the Lateral Motions of Full Trailers,” *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 29, no. S1, pp. 462–478, 1998.

[12] S. Kharrazi, M. Lidberg, R. Roebuck, J. Fredriksson, and A. Odhams,

"Implementation of active steering on longer combination vehicles for enhanced lateral performance," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 50, no. 12, pp. 1949–1970, 2012.

[13] S. Kharrazi, M. Lidberg, and J. Fredriksson, "A generic controller for improving lateral performance of heavy vehicle combinations," *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 227, no. 5, pp. 619–642, 2012.

[14] J. P. Pauwelussen, "SAE Technical Paper: Excessive Yaw Behaviour of Commercial Vehicles, A Fundamental Approach," *Soc. Automot. Eng.*, no. 2001–06–0098, 2001.

[15] S. H. T. Oreh, R. Kazemi, S. Azadi, and A. Zahedi, "A New Method for Directional Control of a Tractor Semi-trailer," *Aust. J. Basic Appl. Sci.*, vol. 6, no. 12, pp. 396–409, 2012.

[16] D. Oberoi, "Enhancing roll stability and directional performance of articulated heavy vehicles based on anti-roll control and design optimization," *M.Sc. Thesis, Univ. Ontario Inst. Technol. Oshawa, Ontario, Canada*, 2011.

[17] S. Luo, "Relative Performance Analysis Of Articulated Vehicles With Multiple Conventional Liftable And Self-Steering Axels." *M.Sc. Thesis, Concordia University, CA,* 2004.

[18] J. R. Billing, C. P. Lam, and J. Couture, "Development of regulatory principles for multi-axle semitrailers," *2nd Int. Symp. Heavy Veh. Weight. Dimens. Kelowna, Br. Columbia June 18-22,* 1989.

[19] A. M. C. Odhams, R. L. Roebuck, D. Cebon, and C. B. Winkler, "Dynamic safety of active trailer steering systems," *Proc. Inst. Mech. Eng. Part K J. Multi-body Dyn.*, vol. 222, pp. 367–380, 2008.

[20] C. Cheng and D. Cebon, "Improving roll stability of articulated heavy vehicles using active semi-trailer steering," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 46, no. S1, pp. 373–388, 2008.

[21] R. K. Vallurupalli, "Directional Dynamic Analysis Of An Articulated Vehicle with articulation dampers and forced steering." *M.Sc. Thesis, Concordia University, CA,* 1993.

[22] E. Dahlberg and J. P. Wideberg, "Influence of the Fifth-Wheel Location on heavy articulated vehicle handling," *8th Int. Symp. Heavy Veh. Weight. Dimens. Johannesburg-South Africa*, 2004.

[23] K. H. Moon, S. H. Lee, S. Chang, J. K. Mok, and T. W. Park, "Method for

Control of Steering Angles for Articulated Vehicles Using Virtual Rigid Axles,” *Int. J. Automot. Technol.*, vol. 10, no. 4, pp. 441–449, 2009.

[24] L. Bortoni-Anzures, G. Herrera-Ruíz, D. Gómez-Meléndez, and M. Martínez-Madrid, “Fuzzy Controller for Automatic Steering in Heavy Vehicle Semi-Trailers,” *Ing. Investig. y Tecnol.*, vol. XIV, no. I, pp. 1–9, 2013.

[25] K. Rangavajhula and H.-S. J. Tsao, “Command steering of trailers and command-steering-based optimal control of an articulated system for tractor-track following,” *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 222, pp. 935–954, 2008.

[26] A. M. C. Odhams, R. L. Roebuck, B. A. Jujnovich, and D. Cebon, “Active steering of a tractor-semi-trailer,” *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 225, pp. 847–869, 2011.

[27] A. M. C. Odhams, R. L. Roebuck, and D. Cebon, “Implementation of active steering on a multiple trailer long combination vehicle,” *Proc AVEC 10, Loughborough, August 2010*, 2010.

[28] B. Jujnovich, R. Roebuck, A. Odhams, and D. Cebon, “Implementation of active rear steering of a tractor-semi-trailer,” *International Conference on Heavy Vehicles HVParis*, 2008

[29] R. L. Roebuck, a. M. Odhams, and D. Cebon, “An automatically reconfigurable software-based safety system for rear-steering multi-unit vehicles,” *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 229, no. 2, pp. 143–162, 2014.

[30] C. Cheng and D. Cebon, “Parameter and state estimation for articulated heavy vehicles,” *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 49, no. 1–2, pp. 399–418, 2011.

[31] D. H. Wu and J. Hai, “Analysis of dynamic lateral response for a multi-axle-steering tractor and trailer,” *Heavy Veh. Syst. Int. J. Veh. Des.*, vol. 10, no. 4, pp. 281–294, 2003.

[32] S. H. Tabatabaei Oreh, R. Kazemi, and S. Azadi, “A new desired articulation angle for directional control of articulated vehicles,” *Proc. Inst. Mech. Eng. Part K J. Multi-body Dyn.*, vol. 226, no. 4, pp. 298–314, 2012.

[33] S. H. Tabatabaei Oreh, R. Kazemi, and S. Azadi, “A sliding-mode controller for directional control of articulated heavy vehicles,” *Proc. Inst.*

Mech. Eng. Part K J. Multi-body Dyn., vol. 228, no. 3, pp. 245–262, 2013.

[34] M. M. Islam, X. Ding, and Y. He, “A closed-loop dynamic simulation-based design method for articulated heavy vehicles with active trailer steering systems,” *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 50, no. 5, pp. 675–697, 2012.

[35] X. Ding, S. Mikaric, and Y. He, “Design of an active trailer-steering system for multi-trailer articulated heavy vehicles using real-time simulations,” *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 227, no. 5, pp. 643–655, 2013.

[36] X. Ding, Y. He, J. Ren, and T. Sun, “A Comparative Study of Control Algorithms for Active Trailer Steering Systems of Articulated Heavy Vehicles,” *Am. Control Conf. Fairmont Queen Elizab. Montr. Canada June 27-June 29*, pp. 3617–3622, 2012.

[37] H. Prem, L. Mai, K. Atley, and H. Prem, “Application of TRACKAXLE ® to Multi-Combination Vehicles,” *9th Int. Symp. Heavy Veh. Weight. Dimens. June 18-22, Pennsylvania State Univ. State Coll. Pennsylvania*, 2006.

[38] H. Prem and E. Ramsey, “Performance evaluation of the Trackaxle Self-Steering System,” *7th Int. Svmposium Heavv Veh. Weight. Dimens. Delft, Netherlands, June 16 - 20*, pp. 405–422, 2002.

[39] F. Mobini, A. Ghaffari, and M. Alirezaei, “Non-linear optimal control of articulated-vehicle planar motion based on braking utilizing the state-dependent Riccati equation method,” *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 229, no. 13, pp. 1774–1787, 2015.

[40] L. Mai, P. Xie, and C. Zong, “Research on algorithm of stability control for tractor semi-trailer,” *IEEE Int. Conf. Mechatronics Autom. August 9 - 12, Chang. China*, pp. 4224–4228, 2009.

[41] Z. Tianjun and Z. Changfu, “Modelling and active safe control of heavy tractor semi-trailer,” *Second Int. Conf. Intell. Comput. Technol. Autom.*, pp. 112–115, 2009.

[42] J. Lu, D. Messih, and A. Salib, “Roll Rate Based Stability Control - The Roll Stability Control ™ System,” *Ford Mot. Company, United States*, paper number: 07–136.

[43] H.-J. Kim, “Robust roll motion control of a vehicle using integrated control strategy,” *Control Eng. Pract.*, vol. 19, pp. 820–827, 2011.

[44] T. Zhu and H. Zheng, "Active roll control of heavy tractor- semitrailer based on adaptive gain scheduling control," *Proc. - ISECS Int. Colloq. Comput. Commun. Control. Manag.*, pp. 131–134, 2008.

[45] H. Yu, L. Güvenç, and Ü. Özgüner, "Heavy-duty vehicle rollover detection and active roll control," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 46, no. 6, pp. 451–470, 2008.

[46] A. J. P. Miège and D. Cebon, "Design and Implementation of an Active Roll Control System for Heavy Vehicles," *6th Int. Symp. Adv. Veh. Control. AVEC, Hiroshima, Japan*, 2002.

[47] Z. Tianjun and J. Liyong, "Dynamic modeling and roll control of heavy tractor-semitrailer," *Proc. 2009 Pacific-Asia Conf. Circuits, Commun. Syst.*, pp. 508–511, 2009.

[48] A. J. P. Miège and D. Cebon, "Active roll control of an experimental articulated vehicle," *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 219, pp. 791–806, 2005.

[49] D. J. M. Sampson and D. Cebon, "An Investigation of Roll Control System Design for Articulated Heavy Vehicles," *4th Int. Symp. Adv. Veh. Control (AVEC), Nagoya, JAPAN*, 1998.

[50] E. J. Stone and D. Cebon, "Control of semi-active anti-roll systems on heavy vehicles," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 48, no. 10, pp. 1215–1243, 2010.

[51] A. J. P. Miege and D. Cebon, "Optimal roll control of an articulated vehicle: theory and model validation," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 43, no. 12, pp. 867–884, 2005.

[52] J. Edgar, "Development of Performance Standards for Australian Heavy Vehicles," *Proc. 8th Int. Symp. Heavy Veh. Weight. Dimens. 14th - 18th March, Johannesburg, South Africa*, 2004.

[53] M. U. Dilberoğlu, "Modeling and Active Steering Control of Articulated Vehicles with Multi-Axle Semi-Trailers," *M.Sc. Thesis, Middle East Tech. Univ. Ankara, Turkey*, 2015.

[54] D. J. M. Sampson, "Active Roll Control of Articulated Heavy Vehicles," *Ph.D. Thesis, Univ. Cambridge, Cambridge, UK*, 2000.

[55] C. Chen and M. Tomizuka, "Dynamic modeling of articulated vehicles for automated highway systems," *Proc. Am. Control Conf. Seattle, Washingt.*, pp. 653–657, 1995.

[56] C. Chen and M. Tomizuka, “Lateral Control of Commercial Heavy Vehicles,” *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 33, no. 6, pp. 391–420, 2000.

[57] C. Chen and M. Tomizuka, “Dynamic Modeling of Tractor-Semitrailer Vehicles in Automated Highway Systems,” *Work. Pap. Calif. PATH Program, Ist. Transp. Stud. Univ. California, Berkeley*, 1995.

[58] D. Hyun and R. Langari, “Modeling to predict rollover threat of tractor-semitrailers,” *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 39, no. 6, pp. 401–414, 2003.

[59] “National Transport Commission: Performance Based Standards Scheme – The Standards and Vehicle Assessment Rules,” 2008.

[60] M. M. Islam, Y. He, S. Zhu, and Q. Wang, “A comparative study of multi-trailer articulated heavy-vehicle models,” *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, 2014.

[61] “SAE J2179: A Test for Evaluating the Rearward Amplification of Multi-Articulated Vehicles,” *American National Standard, Soc. Automot. Eng.*, 2000.

[62] Katsuhiko Ogata, "Modern Control Engineering", *Fifth Edition*, *Prentice Hall*, 2010.

[63] A. Hac and M. D. Simpson, “SAE Technical Paper: Estimation of Vehicle Side Slip Angle and Yaw Rate,” *Soc. Automot. Eng.*, no. 2000–01–06, 2000.

[64] M. A. Johnson and M. J. Grimble, “Recent trends in linear optimal quadratic multivariable control system design,” *Control Theory Appl. IEE Proc. D*, vol. 134, no. 1, pp. 53–71, 1978.

[65] M. Saif, “Optimal linear regulator pole-placement by weight selection,” *Int. J. Control*, vol. 50, no. 1, pp. 399–414, 1989.

[66] R. Eberhart and J. Kennedy, “A new optimizer using particle swarm theory,” *MHS’95. Proc. Sixth Int. Symp. Micro Mach. Hum. Sci.*, pp. 39–43, 1995.

[67] J. Sun, B. Feng, and W. Xu, “Particle swarm optimization with particles having quantum behavior,” *Proc. 2004 Congr. Evol. Comput. (IEEE Cat. No.04TH8753)*, pp. 325–331, 2004.

[68] K. Hassani and W. Lee, “Optimal Tuning of Linear Quadratic Regulators

Using Quantum Particle Swarm Optimization," *Proc. Int. Conf. Control. Dyn. Syst. Robot. Ottawa, Ontario, Canada*, no. 59, 2014.

[69] D. J. M. Sampson and D. Cebon, "Active Roll Control of Single Unit Heavy Road Vehicles," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 40, no. 4, pp. 229–270, 2003.

[70] H. H. Huang, R. K. Yedavalli, and D. a. Guenther, "Active roll control for rollover prevention of heavy articulated vehicles with multiple-rollover-index minimisation," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 50, no. 3, pp. 471–493, 2012.

[71] F. Braghin, F. Cheli, R. Corradi, G. Tomasini, and E. Sabbioni, "Active anti-rollover system for heavy-duty road vehicles," *Veh. Syst. Dyn. Int. J. Veh. Mech. Mobil.*, vol. 46, no. S1, pp. 653–668, 2008.

APPENDIX A

EQUATIONS OF MOTION

General Equations of Motion

Based on the vehicle model presented in Figure 3.5 and using the ISO frame convention, the equations of motion for AHV are obtained as:

Tractor's Lateral:

$$m_1\left(u_1\dot{\psi}_1 + \dot{V}_{y1}\right) - m_{1s}(h_{1s} - h_{1r})\ddot{\phi}_1 = F_{y1f} + F_{y1r} + F_{cy} \tag{A.1}$$

Tractor's Yaw:

$$-I_{1xz}\ddot{\phi}_1 + I_{1zz}\ddot{\psi}_1 = aF_{y1f} - bF_{y1r} - l_{1c}F_{cy} \tag{A.2}$$

Tractor's Roll:

$$\begin{aligned} I_{1x'x'}\ddot{\phi}_1 - I_{1x'z'}\ddot{\psi}_1 &= m_{1s}g(h_{1s} - h_{1r})\phi_1 + m_{1s}(h_{1s} - h_{1r})\left(\dot{V}_{y1} + u_1\dot{\psi}_1\right) \\ &\quad - \left(K_{1f}^* + K_{1r}^*\right)\phi_1 - \left(C_{1f} + C_{1r}\right)\dot{\phi}_1 + K_{12}(\phi_2 - \phi_1) \\ &\quad - F_{cy}h_{1cr} \end{aligned} \tag{A.3}$$

Semitrailer's Lateral:

$$m_2\left(u_2\dot{\psi}_2 + \dot{V}_{y2}\right) - m_{2s}(h_{2s} - h_{2r})\ddot{\phi}_2 = F_{y2} - F_{cy} \tag{A.4}$$

Semitrailer's Yaw:

$$-I_{2xz}\ddot{\phi}_2 + I_{2zz}\ddot{\psi}_2 = -dF_{y2} - l_{2c}F_{cy} \tag{A.5}$$

Semitrailer's Roll:

$$
\begin{aligned}
I_{2x'x'}\ddot{\phi}_2 - I_{2x'z'}\ddot{\psi}_2 \\
&= m_{2s}g(h_{2s} - h_{2r})\phi_2 + m_{2s}(h_{2s} - h_{2r})\left(\dot{V}_{y2} + u_2\dot{\psi}_2\right) \\
&- K_2^*\phi_2 - C_2\dot{\phi}_2 - K_{12}(\phi_2 - \phi_1) + F_{cy}h_{2cr} + T_{AR2}
\end{aligned}
\tag{A.6}
$$

Model 1: Nonlinear Bicycle Model

Eliminating roll motion and substituting F_{cy} from the very first equation, gives simplified equations of motion as follows (Tractor Yaw, Semitrailer Yaw, Lateral):

$$
m_1 l_{1c} V_{y1} + I_{1zz}\dot{\psi}_1 + m_1 l_{1c} u_1 \psi_1 = (a + l_{1c}).F_{y1f} + (l_{1c} - b).F_{y1r} \tag{A.7}
$$

$$
m_1 l_{2c} V_{y1} + m_1 l_{2c} u_1 \psi_1 + I_{2zz}\dot{\psi}_2 = l_{2c}.\left(F_{y1f} + F_{y1r}\right) - d.F_{y2} \tag{A.8}
$$

$$
m_1 V_{y1} + m_1 u_1 \psi_1 + m_2 V_{y2} + m_2 u_2 \psi_2 = F_{y1f} + F_{y1r} + F_{y2} \tag{A.9}
$$

And the nonlinear kinematic constraint equations at fifth wheel are derived by equating longitudinal and lateral velocities obtained from the motion of both vehicle units:

$$
u_2 = u_1 \cos(\psi) + \left(\dot{\psi}_1 l_{1c} - V_{y1}\right)\sin(\psi) \tag{A.10}
$$

$$
\begin{aligned}
-V_{y1}\cos(\psi) + l_{1c}\dot{\psi}_1\cos(\psi) + V_{y2} + l_{2c}\dot{\psi}_2 \\
+ \psi.\left(V_{y1}\sin(\psi) - u_1\cos(\psi) - \dot{\psi}_1 l_{1c}\sin(\psi)\right) = 0
\end{aligned}
\tag{A.11}
$$

With the presented sign convention and considering negative values for tires cornering stiffness, slip angle equations become:

$$
\alpha_{1f} = -\frac{V_{y1} + a\dot{\psi}_1}{u_1} + \delta_1 \tag{A.12}
$$

$$
\alpha_{1r} = -\frac{V_{y1} - b\dot{\psi}_1}{u_1} \tag{A.13}
$$

$$\alpha_2 = -\frac{V_{y2} - d\dot{\psi}_2}{u_2} + \delta_2 \tag{A.14}$$

The following equations are then used to describe the linear tire behavior and are substituted to the equations of motion when constructing the state space matrices:

$$F_{y1f} = -C_{t1f}.\,\alpha_{1f}.\cos(\delta_1) \tag{A.15}$$

$$F_{y1r} = -C_{t1r}.\,\alpha_{1r} \tag{A.16}$$

$$F_{y2} = -C_{t2}.\,\alpha_2.\cos(\delta_2) \tag{A.17}$$

Then equations are put into a nonlinear state-space form:

$$\{x\} = \begin{bmatrix} V_{y1} & \dot{\psi}_1 & \psi_1 & V_{y2} & \dot{\psi}_2 & \psi_2 & \psi \end{bmatrix}^T \tag{A.18}$$

$$[M]\{\dot{x}\} = [F(x,u)] \tag{A.19}$$

$$\{\dot{x}\} = [M]^{-1}[F(x,u)] \tag{A.20}$$

$$[M] = \begin{bmatrix} l_{1c}m_1 & I_{1zz} & l_{1c}u_1m_1 & 0 & 0 & 0 & 0 \\ m_1 & 0 & u_1m_1 & m_2 & 0 & u_2m_2 & 0 \\ l_{2c}m_1 & 0 & l_{2c}u_1m_1 & 0 & I_{2zz} & 0 & 0 \\ -\cos(\psi) & l_{1c}\cos(\psi) & 0 & 1 & l_{2c} & 0 & ** \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{A.21}$$

$$** \equiv V_{y1}\sin(\psi) - u_1\cos(\psi) - l_{1c}\dot{\psi}_1\sin(\psi)$$

$$[F] = \begin{bmatrix} (a + l_{1c})F_{y1f} + (l_{1c} - b)F_{y1r} \\ F_{y1f} + F_{y1r} + F_{y2} \\ l_{2c}F_{y1f} + l_{2c}F_{y1r} - dF_{y2} \\ 0 \\ \dot{\psi}_1 \\ \dot{\psi}_2 \\ \dot{\psi}_1 - \dot{\psi}_2 \end{bmatrix} \tag{A.22}$$

Model 2: Linear Bicycle Model Including Roll Motion

Modifying the general equations of motion based on [54]:

Tractor's Lateral:

$$m_1 u_1\left(\dot{\beta}_1 + \dot{\psi}_1\right) - m_{1s}(h_{1s} - h_{1r})\ddot{\phi}_1 = Y_{\beta_1}\beta_1 + Y_{\dot{\psi}_1}\dot{\psi}_1 + Y_{\delta_1}\delta_1 + F_{cy} \quad \text{(A.23)}$$

Tractor's Yaw:

$$-I_{1xz}\ddot{\phi}_1 + I_{1zz}\ddot{\psi}_1 = N_{\beta_1}\beta_1 + N_{\dot{\psi}_1}\dot{\psi}_1 + N_{\delta_{1f}}\delta_{1f} - F_{cy} l_{1c} \quad \text{(A.24)}$$

Tractor's Roll:

$$\begin{aligned} I_{1x'x'}\ddot{\phi}_1 - I_{1x'z'}\ddot{\psi}_1 &= m_{1s} g(h_{1s} - h_{1r})\phi_1 + m_{1s} u_1(h_{1s} - h_{1r})\left(\dot{\beta}_1 + \dot{\psi}_1\right) \\ &\quad - \left(K_{1f}^* + K_{1r}^*\right)\phi_1 - \left(C_{1f} + C_{1r}\right)\dot{\phi}_1 + K_{12}(\phi_2 - \phi_1) \\ &\quad - F_{cy} h_{1cr} \end{aligned} \quad \text{(A.25)}$$

Semitrailer's Lateral:

$$\begin{aligned} m_2 u_2\left(\dot{\beta}_2 + \dot{\psi}_2\right) - m_{2s}(h_{2s} - h_{2r})\ddot{\phi}_2 \\ = Y_{\beta_2}\beta_2 + Y_{\dot{\psi}_2}\dot{\psi}_2 + Y_{\delta_{2r}}\delta_{2r} - F_{cy} \end{aligned} \quad \text{(A.26)}$$

Semitrailer's Yaw:

$$-I_{2xz}\ddot{\phi}_2 + I_{2zz}\ddot{\psi}_2 = N_{\beta_2}\beta_2 + N_{\dot{\psi}_2}\dot{\psi}_2 + N_{\delta_{2r}}\delta_{2r} - F_{cy} l_{2c} \quad \text{(A.27)}$$

Semitrailer's Roll:

$$\begin{aligned} I_{2x'x'}\ddot{\phi}_2 - I_{2x'z'}\ddot{\psi}_2 &= m_{2s} g(h_{2s} - h_{2r})\phi_2 + m_{2s} u_2(h_{2s} - h_{2r})\left(\dot{\beta}_2 + \dot{\psi}_2\right) \\ &\quad - K_2^*\phi_2 - C_2\dot{\phi}_2 - K_{12}(\phi_2 - \phi_1) + F_{cy} h_{2cr} + T_{AR2} \end{aligned} \quad \text{(A.28)}$$

And the kinematic constraint equation assuming small angles becomes:

$$\dot{\beta}_2 = \dot{\beta}_1 + \frac{h_{1r} - h_{1c}}{u_1}\ddot{\phi}_1 - \frac{h_{2r} - h_{2c}}{u_2}\ddot{\phi}_2 - \frac{l_{1c}}{u_1}\ddot{\psi}_1 - \frac{l_{2c}}{u_2}\ddot{\psi}_2 + \dot{\psi}_1 - \dot{\psi}_2 \quad \text{(A.29)}$$

With the presented sign convention and considering negative values for tires cornering stiffness, slip angle equations become:

$$\alpha_{1f} = -\beta_1 - \frac{a\dot{\psi}_1}{u_1} + \delta_1 \tag{A.30}$$

$$\alpha_{1r} = -\beta_1 + \frac{b\dot{\psi}_1}{u_1} \tag{A.31}$$

$$\alpha_2 = -\beta_2 + \frac{d\dot{\psi}_2}{u_2} [+\delta_2] \tag{A.32}$$

The following equations are then used to describe the linear tire behavior and are substituted to the equations of motion.

$$Y_{\beta_1} = C_{t1f} + C_{t1r} \tag{A.33}$$

$$Y_{\beta_2} = C_{t2} \tag{A.34}$$

$$Y_{\dot{\psi}_1} = \frac{aC_{t1f} - bC_{t1r}}{u_1} \tag{A.35}$$

$$Y_{\dot{\psi}_2} = -C_{t2} \frac{d}{u_2} \tag{A.36}$$

$$Y_{\delta_1} = -C_{t1f} \tag{A.37}$$

$$N_{\beta_1} = aC_{t1f} - bC_{t1r} \tag{A.38}$$

$$N_{\beta_2} = -dC_{t2} \tag{A.39}$$

$$N_{\dot{\psi}_1} = \frac{a^2 C_{t1f} + b^2 C_{t1r}}{u_1} \tag{A.40}$$

$$N_{\dot{\psi}_2} = C_{t2} \frac{d^2}{u_2} \tag{A.41}$$

$$N_{\delta_1} = -aC_{t1f} \tag{A.42}$$

Then equations are put into linear state-space form:

$$\{x\} = [\phi_1 \quad \dot{\phi}_1 \quad \beta_1 \quad \dot{\psi}_1 \quad \phi_2 \quad \dot{\phi}_2 \quad \beta_2 \quad \dot{\psi}_2]^T \tag{A.43}$$

$$[M] = \begin{bmatrix} 0 & M_{12} & M_{13} & M_{14} & 0 & 0 & 0 & 0 \\ 0 & M_{22} & M_{23} & M_{24} & 0 & 0 & 0 & 0 \\ 0 & M_{32} & M_{33} & 0 & 0 & M_{36} & M_{37} & 0 \\ 0 & 0 & 0 & 0 & 0 & M_{46} & M_{47} & M_{48} \\ 0 & 0 & 0 & 0 & 0 & M_{56} & M_{57} & M_{58} \\ 0 & M_{62} & 1 & M_{64} & 0 & M_{66} & -1 & M_{68} \\ 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \end{bmatrix} \tag{A.44}$$

In which:

$M_{12} = -m_{1s}(h_{1s} - h_{1r})l_{1c} - I_{1,xz}$

$M_{13} = m_1 u_1 l_{1c}$

$M_{14} = I_{1,zz}$

$M_{22} = I_{1,xxr} - m_{1s}(h_{1s} - h_{1r})h_{1cr}$

$M_{23} = m_1 u_1 h_{1cr} - m_{1s} u_1 (h_{1s} - h_{1r})$

$M_{24} = -I_{1,xzr}$

$M_{32} = -m_{1s}(h_{1s} - h_{1r})$

$M_{33} = m_1 u_1$

$M_{36} = -m_{2s}(h_{2s} - h_{2r})$

$M_{37} = m_2 u_2$

$M_{46} = m_{2s}(h_{2s} - h_{2r})l_{2c} - I_{2,xz}$

$M_{47} = -m_2 u_2 l_{2c}$

$M_{48} = I_{2zz}$

$M_{56} = I_{2,xxr} - m_{2s}h_{2cr}(h_{2s} - h_{2r})$

$M_{57} = -m_{2s}u_2(h_{2s} - h_{2r}) + m_2u_2h_{2cr}$

$M_{58} = -I_{2,xzr}$

$M_{62} = \dfrac{h_{1r} - h_{1c}}{u_1}$

$M_{64} = -l_{1c}/u_1$

$M_{66} = (h_{2c} - h_{2r})/u_2$

$M_{68} = -l_{2c}/u_2$

$$[N] = \begin{bmatrix} 0 & 0 & N_{13} & N_{14} & 0 & 0 & 0 & 0 \\ N_{21} & N_{22} & N_{23} & N_{24} & N_{25} & 0 & 0 & 0 \\ 0 & 0 & N_{33} & N_{34} & 0 & 0 & N_{37} & N_{38} \\ 0 & 0 & 0 & 0 & 0 & 0 & N_{47} & N_{48} \\ N_{51} & 0 & 0 & 0 & N_{55} & N_{56} & N_{57} & N_{58} \\ 0 & 0 & 0 & -1 & 0 & 0 & 0 & 1 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \end{bmatrix} \quad \text{(A.45)}$$

In which:

$N_{13} = Y_{\beta 1}l_{1a} + N_{\beta 1}$

$N_{14} = \left(Y_{\dot{\psi}1} - m_1u_1\right)l_{1c} + N_{\dot{\psi}1}$

$N_{21} = m_{1s}g(h_{1s} - h_{1r}) - K_{1f}^{*} - K_{1r}^{*} - K_{12}$

$N_{22} = -\left(C_{1f} + C_{1r}\right)$

$N_{23} = Y_{\beta 1}h_{1cr}$

$N_{24} = \left(Y_{\dot{\psi}1} - m_1u_1\right)h_{1cr} + m_{1s}u_1(h_{1s} - h_{1r})$

$N_{25} = K_{12}$

$N_{33} = Y_{\beta 1}$

$N_{34} = Y_{\dot{\psi}1} - m_1 u_1$

$N_{37} = Y_{\beta 2}$

$N_{38} = Y_{\dot{\psi}2} - m_2 u_2$

$N_{47} = N_{\beta 2} - Y_{\beta 2} l_{2c}$

$N_{48} = N_{\dot{\psi}2} - \left(Y_{\dot{\psi}2} - m_2 u_2\right) l_{2c}$

$N_{51} = K_{12}$

$N_{55} = m_{2s} g (h_{2s} - h_{2r}) - K_{2r}^{*} - K_{12}$

$N_{56} = -C_{2r}$

$N_{57} = Y_{\beta 2} h_{2cr}$

$N_{58} = m_{2s} u_2 (h_{2s} - h_{2r}) - m_2 u_2 h_{2cr} + Y_{\dot{\psi}2} h_{2cr}$

$$[O] = \begin{bmatrix} Y_{\delta 1f} l_{1c} + N_{\delta 1f} & Y_{\delta 1f} h_{1cr} & Y_{\delta 1f} & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & Y_{\delta 2r} & N_{\delta 2r} - Y_{\delta 2r} l_{2c} & Y_{\delta 2r} h_{2cr} & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \end{bmatrix} \tag{A.46}$$

$$\{\dot{x}\} = [M]^{-1}[N]\{x\} + [M]^{-1}[O]\{u\} = [A]\{x\} + [B]\{u\} \tag{A.47}$$

APPENDIX B

LQR PARAMETERS

Table B. 1- Weighting Factors for Low-Speed

	q_1	q_2	q_3	r_1
10 km/h	6.5631	-0.5714	0.1192	0.1236
20 km/h	1.3052	-0.5883	0.5478	0.1821
30 km/h	1.5890	-0.5518	0.2978	0.6402

Table B. 2 - Weighting Factors for High-Speed

	q_1	q_2	q_3	q_4	r_1
80 km/h	0.0220	0.0028	0.0420	0.0386	0.8705
90 km/h	0.1059	0.0189	0.0314	0.0719	0.7025
100 km/h	0.0513	0.0887	0.1393	0.0658	0.5711
110 km/h	0.0920	0.1095	0.2235	0.0586	0.4302
120 km/h	0.0239	0.0831	0.2719	0.0275	0.1869

Table B. 3 - Feedback Matrices of Different Vehicle Speeds

	$[K]$							
10 km/h	0	0	0.3922	1.1559	0	0	0.0000	-0.2106
20 km/h	0	0	0.3089	1.3967	0	0	0.0000	-0.9536
30 km/h	0	0	0.2299	0.5982	0	0	0.0000	-0.2635
80 km/h	-0.0146	-0.0026	0.0446	0.0122	0.0182	0.0050	0.0292	-0.0677
90 km/h	-0.0308	-0.0058	0.1119	0.0279	0.0469	0.0176	0.0221	-0.1550
100 km/h	-0.0355	-0.0073	0.1952	0.0177	0.1479	0.0697	0.1153	-0.2023
110 km/h	-0.0426	-0.0088	0.2758	0.0110	0.2241	0.1034	0.2327	-0.2539
120 km/h	-0.0528	-0.0101	0.3682	-0.0021	0.3294	0.1534	0.5668	-0.3106

APPENDIX C

VEHICLE PARAMETERS

Table C. 1 - Vehicle Parameters (Based on [5])

Parameter	Value	Parameter	Value
m_{1s}	4819 kg	I_{1xz}	2175.5 kg.m^2
m_{2s}	30821 kg	I_{2xz}	18497.43 kg.m^2
m_{1uf}	650 kg	$I_{1x'x'}$	4348.41 kg.m^2
m_{1ur}	1300 kg	$I_{2x'x'}$	42025.2 kg.m^2
m_{2u}	1330 kg	$I_{1x'z'}$	2175.5 kg.m^2
h_{1s}	1.058 m	$I_{2x'z'}$	18497.43 kg.m^2
h_{2s}	1 m	K_{12}	114590 Nm/rad
h_{1r}	0.558 m	K_{1f}^*	815220.2 Nm/rad
h_{2r}	0.723 m	K_{1r}^*	655023.6 Nm/rad
h_c	1.1 m	K_2^*	409960 Nm/rad
l_{1c}	1.959 m	C_{1f}	160000 Nm.s/rad
l_{2c}	5.853 m	C_{1r}	160000 Nm.s/rad
a	1.115 m	C_2	270000 Nm.s/rad
b	1.959 m	C_{t1f}	277200 N/rad
d	1.147 m	C_{t1r}	740280 N/rad
I_{1zz}	20606.07 kg.m^2	C_{t2}	2646000 N/rad
I_{2zz}	226271.79 kg.m^2		

APPENDIX D

TRUCKSIM® MODEL DETAILS

Table D. 1 - TruckSim Model Parameters

Lead Unit		**Trailer**	
Type: Lead Unit with 2 Axles		Type: Trailer with 3 Axles	
Sprung Mass: Rigid Sprung Mass		Sprung Mass: Rigid Sprung Mass	
Mass:	4819 kg	Mass:	30821 kg
Roll Inertia:	4384.41 $kg.m^2$	Roll Inertia:	42025.2 $kg.m^2$
Pitch Inertia:	2000 $kg.m^2$	Pitch Inertia:	30000 $kg.m^2$
Yaw Inertia:	20606.07 $kg.m^2$	Yaw Inertia:	226271.79 $kg.m^2$
Product (Ixz):	2175.5 $kg.m^2$	Product (Ixz):	18497.43 $kg.m^2$
Radius of Gyration (Rx):	0.950 m	Radius of Gyration (Rx):	1.168 m
Radius of Gyration (Ry):	0.644 m	Radius of Gyration (Ry):	0.987 m
Radius of Gyration (Rz):	2.068 m	Radius of Gyration (Rz):	2.710 m
CG Coordinate:	1115, 1058 mm	CG Coordinate:	5853, 1000 mm
		Hitch Height:	1100 mm
Tires: Linear		Tires: Linear	
Steering Wheel Torque: 1/25		Axle 1 Distance Back: 5900 mm	
Powertrain: 300 kW, 10-spd, 2WD		Axle 2 Distance Back: 7000 mm	
Hitch: 5th Wheel (220 deg. Lash)		Axle 3 Distance Back: 8100 mm	

Table D. 1 (Continued)

Front Axle Type: Solid Axle

Unsprung Mass:	650 kg
Axle Roll & YawIinertia:	1000 kg.m^2
Wheel Center Height:	520 mm
Track:	2200 mm
Roll Center Height:	558 mm
Auxiliary Roll Moment Coefficient:	17000 Nm/deg
Auxiliary Roll Damping:	2792.5 Nm.s/deg

Axles Type: Solid Axles

Unsprung Mass:	443.33 kg
Axle Roll & YawIinertia:	1000 kg.m^2
Wheel Center Height:	520 mm
Track:	2000 mm
Roll Center Height:	723 mm
Auxiliary Roll Moment Coefficient:	3000 Nm/deg
Auxiliary Roll Damping:	1570.8 Nm.s/deg

Rear Axle Type: Drive Axle

Unsprung Mass:	1300 kg
Axle Roll & YawIinertia:	1000 kg.m^2
Wheel Center Height:	520 mm
Track:	2000 mm
Roll Center Height:	558 mm
Auxiliary Roll Moment Coefficient:	17000 Nm/deg
Auxiliary Roll Damping:	2792.5 Nm.s/deg