THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

IN

MACHINE AND VEHICLE SYSTEMS

Steering Based Lateral Performance Control of Long Heavy Vehicle Combinations

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Gothenburg, Sweden 2012

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ISBN 978-91-7385-724-6

Doktorsavhandlingar vid Chalmers tekniska högskola.
Ny serie nr 3405
ISSN 0346-718X

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Chalmers Reproservice
Gothenburg, Sweden 2012

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Abstract

In this thesis the lateral performance of heavy vehicle combinations, specifically longer combination vehicles, is discussed. The use of longer combination vehicles is promoted by their positive impact on the traffic congestion problem, as well as their economic and environmental benefits due to reduced fuel consumption and emissions. However, from a safety perspective, there are concerns about their impact on the traffic. In a heavy vehicle combination maneuvering at high speeds, lateral motions get amplified at the towed units, which causes trailer swing and large path deviation and side slip. These amplified motions are dangerous for any nearby cars as well as the vehicle combination itself and can lead to instability. The main goal of the research presented in this thesis is to develop control strategies for improving the lateral performance of heavy vehicle combinations at high speeds by suppression of amplified motions at the towed units. As a starting point, the heavy vehicle accidents are investigated and the relevant critical maneuvers are identified. Subsequently, the lateral performance of heavy vehicle combinations in the identified critical maneuvers is investigated by simulations to obtain a better understanding of the causes behind rearward amplification of motions in heavy vehicle combinations and to specify the control objectives. Accordingly, a generic controller for improving the lateral performance of heavy vehicle combinations by active steering of the towed units is developed. The developed controller is verified for various heavy vehicle combinations by simulation, with respect to the identified critical maneuvers. The verification results confirm the effectiveness of the controller and show significant reductions in yaw rates, side slip and path deviation of the towed units of the heavy vehicle combinations, up to 70%. Additionally, the robustness of the controller is evaluated by extensive analysis of its performance in various driving conditions and presence of parameter uncertainties for a sample heavy vehicle combination. Furthermore, the controller is implemented on a truck-dolly-semitrailer test vehicle and verified in a series of single and double lane changes. The experimental results approve the simulation outcomes. The developed controller can be easily implemented on steerable trailers; since it utilizes common sensors for steering input, speed and yaw rates and does not require large computing capacity. The significant improvements obtained by the developed controller can promote the use of longer combination vehicles in traffic, which will result in a reduction of traffic congestion problem, as well as substantial environmental and economic benefits.

Keywords: Heavy Vehicle, Longer Combination Vehicle, Lateral Performance, Active Steering, Rearward Amplification, Offtracking, Trailer Swing, Accident Analysis, Sine with Dwell

Acknowledgements

Studying for a PhD is commonly referred to as a journey, for me, it actually involved several journeys; lots of travelling between Gothenburg and Linköping, going to Cambridge for experimental tests, travelling to north of Sweden for winter testing on frozen lakes, and of course many fantastic trips to different parts of the world to attend conferences. Several people helped me during these journeys, some stayed with me all along and some joined for a short time; I would like to acknowledge them all here.

First of all, I would like to thank my supervisor, Associate Professor Mathias Lidberg, for his guidance throughout the whole duration of my studies and his never ending enthusiasm for this research. I am also thankful to Associate Professor Robert Thomson and Associate Professor Jonas Fredriksson for their supervision and advice. Special thanks go to Professor Bengt Jacobson and Professor Anders Boström for their leadership at different phases of this research. This research was supported by the Swedish Intelligent Vehicle Safety Systems program (IVSS), for this I am very grateful.

During my studies, I had the opportunity to collaborate with Volvo Trucks Group Trucks Technology, and therefore, I had the chance to test my ideas on real vehicles; I am really grateful for that. I would like to specially thank Dr. Leo Laine and Dr. Peter Lingman for their guidance and the enlightening discussions. I am thankful to Kristoffer Tagesson, Nicolas Dela and John Aurell for their valuable inputs and collaboration. I also thank all the members of the steering group for the fruitful discussions.

The experimental tests were possible, thanks to the Cambridge Vehicle Dynamics Consortium and its research director, Professor David Cebon. Special thanks go to Dr. Richard Roebuck and Dr. Andrew Odhams for their efforts to make the MIRA testing happen.

Another great opportunity I had during my studies was to meet and collaborate with Professor Tim Gordon from University of Michigan; Thanks for the inspiring discussions and great collaboration.

I also thank all my colleagues at Vehicle Engineering and Autonomous Systems Division and SAFER, it has been a privilege working with you. I would like to specially thank Lic Derong Yang and Dr. Matthijs Klomp for making the Vehicle Dynamics Group, a fun and inspirational group. I am also grateful to Malin Kjellberg, head of VEAS, and Anna Nilsson-Ehle, director of SAFER, and the administrative staff, Sonja Laakso Gustafsson, Marianne Hedfors and Lisa Knutsson for creating a stimulating working environment.

I am thankful to the staff of fluid and mechatronic systems division at Linköping University, especially Rita Enquist, for welcoming me to join their coffee breaks after I moved to Linköping. I would also like to thank Dr. Jonas Jansson and his colleagues for welcoming me to VTI, Linköping, in my last year of studies, where I had a chance to write this thesis in a calm and pleasant atmosphere.

I would like to thank all my friends in Gothenburg and Linköping and elsewhere in the world, for making my spare time amusing and memorable. I am especially grateful to Sahar and Ali for their hospitality and for making my trips to Gothenburg much more enjoyable.

Finally I would like to express my love and gratitude to my family, my parents, for their continuous support and encouragements, my sisters and brother, for always being there for me, Tohid, for his love and Adrian, for bringing everlasting joy to my life and greeting me with a big hug and cheerful smiles after each day of work.

Sogol Kharrazi
Linköping, July 2012

List of Publications

This thesis is based on the following publications:

Paper A
S. Kharrazi and R. Thomson, "Study of Heavy Truck Accidents with Focus on Manoeuvres Causing Loss of Control," International Journal of Vehicle Safety, vol. 3 (1), pp. 32-44, 2008.

Paper B
S. Kharrazi, M. Lidberg, and J. Fredriksson, "A Generic Controller for Improving Lateral Performance of Heavy Vehicle Combinations," Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering (in press), 2012.

Paper C
S. Kharrazi, M. Lidberg, R. Roebuck, J. Fredriksson, and A. Odhams, "Implementation of Active Steering on Longer Combination Vehicles for Enhanced Lateral Performance, " Journal of Vehicle System Dynamics (in press), 2012.

Paper D
S. Kharrazi, T. Gordon, and M. Lidberg, "Improving Lateral Performance of Longer Combination Vehicles - An Approach Based on Eigenstructure Assignment," in IEEE Intelligent Vehicles Symposium, Alcala de Henares, Spain, 2012, pp. 328-333.

Related work by the author, not included in the thesis

S. Kharrazi, M. Lidberg, P. Lingman, J. I. Svensson, and N. Dela, "The Effectiveness of Rear Axle Steering on the Yaw Stability and Responsiveness of a Heavy Truck," Vehicle System Dynamics, vol. 46 (S1), pp. 365-372, 2008.

L. Laine, S. Kharrazi, and N. Dela, "Proposal for Using Sine With Dwell on Low Friction for the Evaluation of Yaw Stability for Heavy Vehicle Combinations," in IEEE International Conference on Vehicular Electronics and Safety, Columbus, OH, USA, 2008, pp. 163-167.

S. Kharrazi and M. Lidberg, "Allocation of Braking and Steering Forces for Lateral Stability Control of a Heavy Vehicle Combination," in 21st International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD'09), Stockholm, Sweden, 2009.

S. Kharrazi, J. Fredriksson, and M. Lidberg, "Lateral Stability Control of a Long Heavy Vehicle Combination by Active Steering of the Towed Units," in 13th International IEEE Conference on Intelligent Transportation Systems, Madeira Island, Portugal, 2010, pp. 168-173.

M. K. Bahaghighat, S. Kharrazi, M. Lidberg, P. Falcone, and B. Schofield, "Predictive Yaw and Lateral Control in Long Heavy Vehicles Combinations," in 49th IEEE Conference on Decision and Control, Atlanta, GA, USA, 2010, pp. 6403-6408.

S. Kharrazi, M. Lidberg, J. Fredriksson, "Robustness Analysis of a Steering-Based Control Strategy for Improved Lateral Performance of a Truck-Dolly-Semitrailer," Submitted to International Journal of Heavy Vehicle Systems, 2012.

P. Falcone, B. Augusto, A. Odhams, S. Kharrazi, M. Lidberg and R. Roebuck, "Design and Experimental Validation of Predictive Active Steering Strategies for Yaw and Lateral Control in Long Heavy Vehicles Combinations," In preparation for submission to IEEE Transactions on Vehicular Technology, 2012.

K. Tagesson, L. Laine and S. Kharrazi, “Method and Arrangement for Vehicle Stabilization,” Filed patent application, *The Patent Cooperation Treaty*, PCT/SE2011/000193, 2011.

Contents

Chapter 1

Introduction

Road freight accounts for approximately 45% of the total transport (tones-km) within Europe and heavy vehicle combinations are an important part of it all over Europe [1]. However, the existing regulations on the permitted length and weight of heavy vehicle combinations vary between different regions. In major parts of Europe, only the conventional combination vehicles are permitted with the maximum length of 18.75 m and maximum weight of 40 ton (44 ton if carrying an ISO container); whereas in Sweden and Finland longer combination vehicles (LCV) up to 25.25 m long and 60 ton are allowed. Still, some countries like Canada, USA, Brazil and Mexico allow even longer and heavier combinations and in Australia long and heavy road trains, which are more than 50 m long and weigh more than 100 ton, are used in remote areas [2]. The legalized operation of longer combination vehicles in the aforementioned countries is due to their economic and environmental benefits and positive impact on the traffic congestion problem. A Swedish study on the benefits of LCVs has shown their potential in reducing the fuel consumption and emissions in goods transport by 15%, the operational costs by 23% and the number of trips by 32% [3]. In a report by Woodrooffe and Ash, it is estimated that use of LCVs in Alberta, Canada has resulted in a 29% saving in transportation costs, 44% reduction in heavy vehicle-km travelled, 32% reduction in fuel consumption and emissions and 40% decrease in road wear [4].

The reluctance towards LCVs in Europe is mostly because of concerns about their safety issues; the pros and cons of LCVs and the conducted research in this area are summarized in a review by Grislis, which shows the conflicting views on the influences of LCVs on the traffic safety among researchers [5]. Due to lack of adequate accident data for LCVs it is not easy to draw any empirical conclusion on their safety aspects. Nonetheless, considering the advantages of LCVs and the fact that the amount of transported goods is expected to increase by 55% from year 2000-2020, many organizations are supporting and encouraging the European Modular System [1]. European Modular System, which is in operation in Sweden and Finland, is a concept of allowing LCVs that consist of existing loading units (modules). Currently in Sweden, field tests of even longer and heavier combinations than the existing LCVs is being considered, for instance in the timber haulage industry [6]. Therefore, to promote the operation of LCVs and ease the concerns about their impact on traffic safety, there is a crucial need for technical solutions which enhance the safety performance of LCVs and prevent loss of control by the driver and consequent vehicle instability.

Two major safety issues of LCVs which require improvement is their rollover tendency and poor lateral performance at high speeds, such as trailer swing and large path deviation. Considering the fact that rollover issues of heavy vehicle combinations including LCVs have been addressed more widely than their lateral performance, the research presented in this thesis is focused on the latter. Nonetheless, the rollover issues of LCVs are not excluded from this research completely, as rollover and lateral performance are not two separate topics. Improvements in the lateral performance of an LCV can also lead to a reduced risk of turn-over, which is a rollover solely due to severe steering maneuver and consequent excessive lateral acceleration.

1.1 Research Objectives

The main goal of the research presented in this thesis is to develop control strategies for improving the lateral performance of heavy vehicle combinations at high speeds, with focus on LCVs. A problem-based approach is used to reach this goal; in other words, to obtain a better understanding of the problem, first the heavy vehicle accidents are investigated to determine the relevant critical maneuvers. Subsequently, the lateral performance of LCVs in the determined critical maneuvers is investigated by simulations. Finally steering based control strategies are developed to overcome this problem and are evaluated with respect to the determined critical maneuvers. The evaluation is performed by simulations as well as experiments on a test track.

1.2 Limitations of Scope

In this thesis only the control systems that utilize measured/estimated data about the internal states of the vehicle are considered; no perception system or communication with other vehicles or infrastructure is included.

In this thesis only steering and braking actuators for control of the vehicle dynamics are considered; other actuators such as active suspension components or devices mounted at the articulation joints are excluded.

In this thesis the driver modeling is not considered and the driver role is limited to a steering input in the performed simulations.

1.3 Thesis Outline

In Chapter 2 the background information about heavy vehicle accidents, lateral performance measures for heavy vehicle combinations and relevant existing literatures are presented. Chapter 3 gives an overview of the conducted study on heavy vehicles accidents and obtained results. It is followed by a summary of the investigation on the lateral performance of passive heavy vehicle combinations in Chapter 4, by means of simulation. In Chapter 5, the developed control strategies for improving the lateral performance of heavy vehicle combinations are presented and their effectiveness is analyzed. Finally, Chapter 6 summarizes the scientific contributions and provides concluding remarks.

Chapter 2

Background

In this chapter a review of heavy vehicle accidents, standard lateral performance measures for heavy vehicle combinations and relevant existing literatures are presented.

2.1 Heavy Vehicle Accidents

The severe consequences of heavy vehicle accidents capture public attention and concern; these consequences are not only limited to personal injuries and fatalities, but also include substantial financial costs and environmental hazards such as spill of hazardous material. Figure 2.1, shows the contribution of heavy vehicles to the traffic fatalities in US along with their share in the number of registered vehicles and total vehicles miles traveled during the period of 2003-2010 [7-15]; it can be seen that heavy vehicles are overrepresented in the traffic fatalities, but there is a trend towards improvement. There exist considerable amounts of general statistics on heavy vehicles accidents, for instance in [16]; however, in-depth accident investigations of heavy vehicles are rather limited. In the next paragraph a summary of the relevant accident studies are provided.

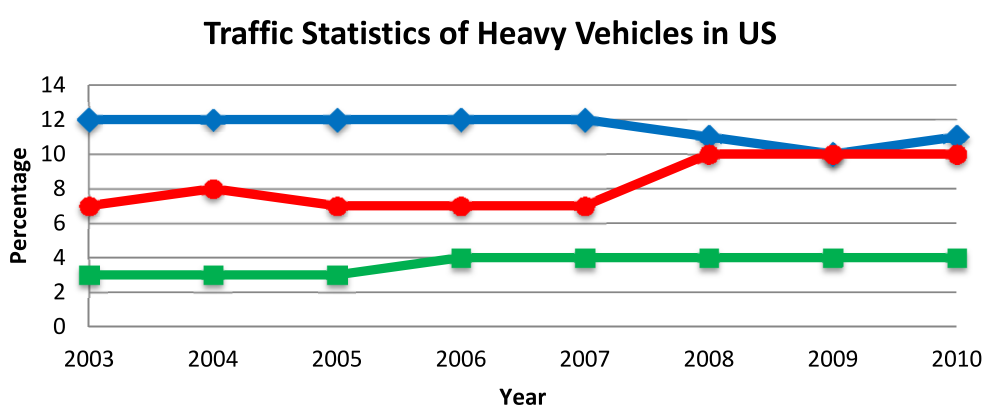


Figure 2.1 Traffic statistics of heavy vehicles in US, data from [7-15], blue: contribution to traffic fatalities, red: contribution to vehicles miles traveled, green: contribution to registered vehicles

In a study by Milliken and de Pont, heavy vehicles performance in New Zealand was investigated. According to their report, 20% of heavy vehicle accidents were due to loss of control (rollover and lateral instability) of the heavy vehicle; 66% of these occurred while the heavy vehicle was cornering.

It was also mentioned that 55% of truck rollover accidents in New Zealand were attributed to speed through curves, 21% were related to running off the edge of the roadway, and 6% were result of an evasive maneuver [17]. In another study, de Pont observed that in Tasmania 16% of all heavy vehicle accidents were rollover accidents, 50% of which were related to speed through curves, 27% to running off the edge of the roadway, 9% to vehicle defects, 7% to load shift and 2% were due to evasive maneuvers [18]. A similar investigation performed in the Netherlands showed that 61% of heavy vehicle rollover accidents were due to speed through curves, 26% were related to running onto the soft shoulder and 10% were caused by evasive maneuvers [19]. In 2006, the Knorr Bremse Group presented an analysis of heavy vehicle accidents in Germany with regard to the fitment of ESC; in this analysis, two types of accidents were determined to be ESC relevant, namely collision with obstacle in the same lane and curve departure. According to their study, 11% of heavy vehicle accidents in 2004 were ESC relevant [20]. In another German study, the heavy vehicle accidents reported by the police that involved fatalities or serious injuries in Bavaria in 1997 were analyzed with respect to ESC effectiveness. The analysis showed that if the heavy vehicles were fitted with an ESC system, 73 accidents out of a total of 850 accidents (8.6%) could have been prevented. Moreover, the 73 ESC-relevant accidents were studied to establish primary accident causes. The most prominent causes were violent steering reactions following inattentiveness, skidding after collisions, inappropriate speed and skidding in a curve [21].

The abovementioned studies show that a considerable portion of heavy vehicle accidents are related to poor lateral performance. However, all these studies include a rather small population of vehicles; furthermore, the critical maneuvers causing lateral instability are not investigated thoroughly in these studies. Therefore, an analysis of heavy vehicle accidents with respect to the lateral performance at high speeds was conducted, using a large in-depth accident database, as the first step of the research presented in this thesis.

2.2 Standard Lateral Performance Measures

To be able to reduce the number of accidents of heavy vehicle combinations, it is crucial to characterize their dynamic performance. There have been efforts to develop standard test procedures and performance measures that relate the dynamics properties of a heavy vehicle combination with its likelihood of being involved in accidents. The most commonly used performance measures that characterize different aspects of the lateral performance of heavy vehicle combinations are: rearward amplification (RWA), offtracking and yaw damping [22, 23].

Rearward amplification is defined as the ratio of the peak value of a motion variable of interest for the rearmost unit to that of the lead unit, see Figure 2.2. It is usually given in terms of yaw rate or lateral acceleration. This performance measure indicates the increased risk for a swing out or rollover of the last unit compared to what the driver is experiencing in the lead unit. RWA may be determined based on the vehicle's response gain in the frequency domain or in a specific transient maneuver. Offtracking is also a comparison between the lead and last unit, but in terms of the additional road space required for the last unit maneuvering; offtracking is defined as the lateral deviation between path of the front axle of the vehicle and path of the rearmost axle, see Figure 2.3. If a single value is given for the offtracking, it is the maximum lateral deviation. High speed offtracking, which is an outboard offtracking, can be either determined in a steady state turn or in a transient maneuver such as lane change; the latter is termed as high speed transient offtracking. The third common performance measure, yaw damping, is the damping ratio of the least damped articulation joint of the vehicle combination during free oscillations, see Figure 2.4. Yaw damping ratio of an articulation joint is determined from the amplitudes of the articulation angle of subsequent oscillations [22, 23].

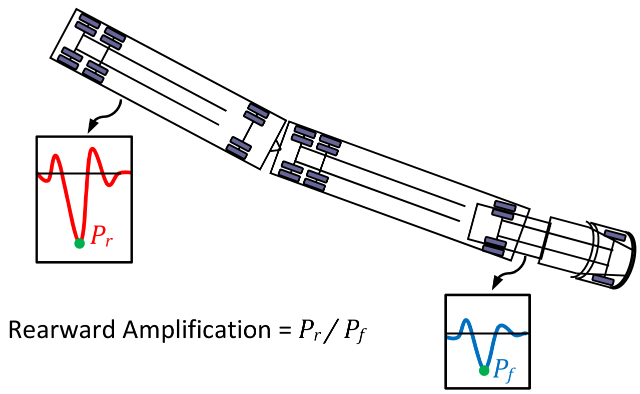


Figure 2.2 Illustration of rearward amplification, *P* denotes peak value of the motion variable of interest

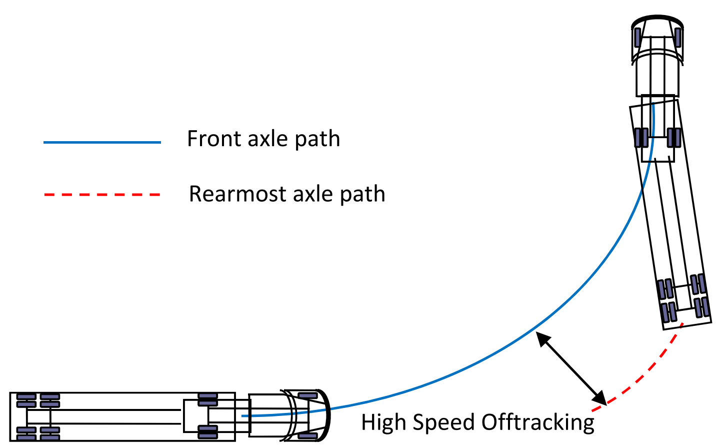


Figure 2.3 Illustration of high speed offtracking

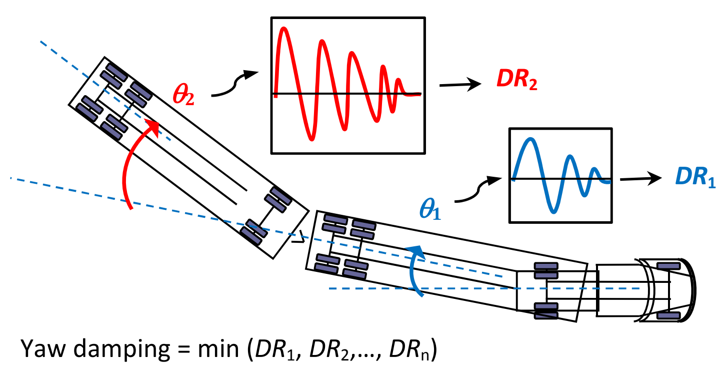


Figure 2.4 Illustration of yaw damping (here n=2), *DR* denotes damping ratio of the articulation joint

Ideally, there should be standard performance targets for the described measures based on in-depth analysis of accident records and heavy vehicle combinations dynamics; such targets do not exist. However, some advisory target levels can be found in [24-26]. In a survey by Mueller, et al., performance measures of the New Zealand heavy vehicle combinations fleet with those involved in accidents classified as rollover or loss of control have been compared. This survey demonstrates that vehicles with poor performance measures have a higher likelihood of being involved in such accidents [25].

So far the lateral performance and relevant accidents of heavy vehicle combinations in general has been discussed. Next sections are focused on existing studies on the lateral performance of LCVs.

2.3 Studies on Lateral Performance of LCVs

In order to investigate the safety implications of longer combination vehicles, their lateral performance has been studied by various researchers. In an article by Wideberg and Dahlberg, stability of heavy vehicle combinations dominant in different part of the world are compared with respect to their lateral acceleration RWA, offtracking, damping ratio and load transfer ratio (measuring rollover risk), using a multi-body dynamics software. They have concluded that longer and heavier combinations are more likely to have poor performance [27]. On the contrary, in a study by Aurell and Wadman it is argued that longer combination vehicles have in general better dynamic stability than conventional combinations; this argument is based on a comparative study of yaw rate RWA and offtracking of different combinations [2]. In a similar study by Bozsvari, et al., the lateral acceleration of different vehicle combinations during a lane change maneuver are investigated and it is concluded that some LCVs are as stable as conventional combinations [28]. Danielsson has also studied the lateral stability of two prospective LCVs in Sweden and concluded that their lateral stability is comparable with the existing heavy vehicle combinations in traffic, [29]. The dissimilarities between the drawn conclusions in the aforementioned studies are due to the differences in the vehicle configuration features considered in each study. In a comprehensive report by Ervin, safety performance (including lateral performance) of 22 different heavy vehicle combinations common in Canada are compared using seven performance measures, covering both low and high speeds. Ervin has divided the vehicles into 4 groups based on their performance which is not directly related to the length and weight of the combination, but depends on the various configuration features [24]. Luijten, et al., have also investigated the lateral dynamic behavior of different heavy vehicle combinations in frequency domain and concluded that configuration features such as wheelbase and coupling positions play an important role in stability of the vehicle combination [30]. Winkler and Bogard have studied the lateral acceleration experience of multi-trailer combinations in service and analyzed data gathered during 350,000 miles of travel over a period of eight months. The results obtained show the significant influence of the dolly type (which mainly reflects different number of articulation points in two otherwise similar combinations) on RWA of a heavy vehicle combination [31]. In a report by Fancher and Mathew, it is argued that the safety impacts of a specified set of length and weight regulations on heavy vehicle combinations will depend to a large extent upon the safety of the combinations that would be very productive and hence favored under the given allowances or constraints. They have also reviewed the relationship between the safety performance (including lateral performance) of heavy vehicle combinations and the design features that would be adjusted to promote productivity, such as number of articulation joints, wheelbase length and number of axles in a suspension. The investigation results indicate the need for compromises in designing vehicles to meet opposing demands at low speed and high speed [32]. Fancher has further discussed the influences of multiple axles and articulation points on the lateral performance characteristics of heavy vehicle combinations in [33]. For a broader review of the lateral performance issues in evaluation and design of articulated heavy vehicles refer to the article by Fancher and Winkler [34].

This section provided a summary of existing studies on the lateral performance of passive LCVs. In next section existing research on control systems for improving the lateral performance of LCVs are reviewed. It should be noted that only a subgroup of control systems which follow the driver steering input to offer enhanced stability and maneuverability are considered. Autonomous systems, which handle a situation when the driver's intention to steer is missing such as lane departure systems, are not included.

2.4 Studies on Lateral Stability Control of LCVs

The existing braking based electronic stability control (ESC) for the lateral dynamics of heavy vehicle combinations is generally limited to first unit control; some ESC functions additionally send a braking demand to the trailers for speed reduction and avoiding jackknife [35], which is not sufficient for suppressing the amplified motions of the towed units (see Paper C). In addition to ESC, rear axle steering (RAS) exists on some trucks and tractors and can affect the behavior of the towed units to some extent, by manipulation of tire slips on the rear axle [36]; but again it is not sufficient especially in LCVs. Moreover, passive steering exists on certain type of trailers that steer some wheels on the trailer according to a geometrical relationship or force/moment balance to improve low speed maneuverability as well as to decrease tire wear. However, such systems generally increase rearward amplification and offtracking in high speeds and are locked at high speeds [37]. In this scope, different studies have been carried out to develop control strategies for further improvement of the lateral performance of heavy vehicle combinations; these control strategies can be divided into two major groups: braking-based and steering-based.

An example of a braking-based control scheme is the LQR controller designed by Palkovics and El-Gindy for a tractor-semitrailer combination. The controller applies differential braking at the tractor rear axle to reduce the yaw rate and side slip angle of the tractor, as well as the articulation angle and rate [38]. In a follow-up study, they examined control schemes with the same objectives and cost function structure of the LQR controller but different actuators, such as active steering of the trailer steering [39]. Braking-based solutions for LCVs have been considered in a study at University of Michigan Transportation Research Institute (UMTRI), in which application of differential braking for the lateral stability control of a tractor – semitrailer – dolly – semitrailer combination is investigated by simulation. In the proposed controller, a yaw torque is applied to the dolly by differential braking to steer the dolly so that the path of the full trailer will more closely approximate the path of the tractor's front axle [40]. In another study by UMTRI, various active braking strategies for suppression of rearward amplification of a triple trailer combination were implemented and studied on an experimental vehicle. The study was focused on a simple trailer-by-trailer basis system, that is, the proposed system, implemented on a specific trailer, did not depend on information from other units. The proposed system was tested in a 2.5 m single lane change maneuver at an initial speed of 88 km/h and could reduce the lateral acceleration RWA from 2.7 to 1.7 with a resulted speed reduction of approximately 15 km/h [41].

An example of a steering-based control strategy for conventional combination vehicles can be found in the study conducted at Cambridge Vehicle Dynamics Consortium (CVDC) where an LQR controller is developed and implemented on a tractor-semitrailer test vehicle. The CVDC controller steer the semitrailer axles to improve path following and roll stability [42]. Use of active steering for LCVs has been investigated by Rangavajhula and Tsao in [43], where an LQR controller is proposed to minimize the lateral acceleration rearward amplification as a surrogate for offtracking reduction. The proposed LQR controller uses the feedback from all vehicle states and steers the front axles of the first and second trailers and eliminates the offtracking in the simulated lane change maneuver at moderate speed of 54 km/h.

The presented summary of the made efforts for the enhancement of lateral performance of heavy vehicle combinations, shows that number of studies on longer combination vehicles and achieved outcomes are rather limited; hence, there is a necessity for further investigations and improvements. In this thesis steering-based control strategies for this purpose are presented that can enhance the lateral performance of heavy vehicle combinations, including LCVs. The choice of steering over braking is due to the fact that braking-based systems cannot operate without slowing down the vehicle. Furthermore, a continuous and fine-tuned control intervention is not possible with a braking based system which should have a dead-band to prevent the continuous action of brakes and consequent excessive wear of the brake lining and tires and an undesired speed reduction. Thus, the braking-based systems are commonly designed to intervene in critical situations; however, the lateral performance of an LCV operating at high speeds need to be improved by suppression of rearward amplification of motions even in normal lateral maneuvering. This calls for a control system that can be in continuous action at high speeds maneuvering with non-aggressive interaction, which is feasible by utilizing steering actuators. Moreover, steering-based systems, unlike the braking-based systems, can influence the lateral forces and side slip angles directly. This does not mean that steering-based systems should replace braking-based systems; they are complementary to each other, in situations close to the handling limit, a speed reduction is favorable and the steering-based system can be augmented by brake interventions. The hardware required for active steering of the towed units is justifiable considering the consequent improvement in the lateral performance at high speeds and also the fact that active steering is of great importance to low speed maneuverability as well. To be able to maneuver an LCV on narrow roads with tight roundabouts, steering of the towed units is necessary [44].

Chapter 3

Accident Analysis

The heavy vehicle accidents were investigated to obtain a better understanding of the problems associated with poor lateral performance of heavy vehicles at high speeds and to determine the relevant critical maneuvers. Here an overview of the conducted study and the obtained results are presented. A more comprehensive description is provided in Paper A.

There are two main approaches to causation study of traffic accidents. The first approach is called the expert or clinical method and the other one is the statistical method. In the clinical method a group of multidisciplinary experts investigate each accident to determine its causing factors. A clinical approach depends on the judgment of the experts and is inevitably subjective. On the other hand, in the statistical method researchers attempt to collect objective data describing the crash based on a predefined accident coding methodology. In both cases, association between accidents and different causing factors can be identified afterwards by statistical analysis of the resulted data [45]. For the purpose of this study, the statistical approach was used considering the available resources and also to decrease the subjectivity of the results.

To select an appropriate accident database for the analysis, existing accident databases, which can be divided into two main groups of exhaustive databases and in-depth databases, were reviewed. Exhaustive databases contain a large amount of information, for example information on all accidents reported through a specific agency such as the police, hospitals or emergency service providers. Although these databases contain information on a large number of accidents, they usually do not provide detailed information. In contrast, in-depth databases usually have a smaller number of accidents, but have more detailed information on each case. Typically the information is collected by specially trained teams based on a predefined coding method [46]. For the purpose of this research, an in-depth database with detailed information on pre-collision events and contributing causal factors, rather than injury consequences, had to be selected. After reviewing available accident databases, the Large Truck Crash Causation Study (LTCCS) database was selected.

3.1 LTCCS Database

LTCCS is an in-depth database collected by the National Highway Traffic Safety Administration (NHTSA) and the Federal Motor Carrier Safety Administration (FMCSA) of the US Department of Transportation. LTCCS includes accidents involving at least one heavy vehicle which caused an identifiable injury. The collected data provide detailed information about the crash environment, drivers, vehicles and non-motorists involved in the crash, via approximately 1000 variables. The investigated accidents occurred from 1 April, 2001, to 31 December, 2003 in 24 locations within the United States, classified by geographic region and population size. There were 1,070 accidents involving 2,284 vehicles. Scaling factors were calculated for each crash through the use of statistical sampling method and USA national estimates were determined by applying these scaling factors to each accident. The collected data represent 120,000 accidents involving a total of 241,000 vehicles, of which 141,000 are heavy vehicles [47].

Although LTCCS is essentially a collision-avoidance study, its accident categorization was not directly applicable in the research presented herein. The accidents in the LTCCS database were categorized based on the widely used accident type classification of head-on, rear-end, side-swipe, etc. However, for the purpose of this research, the critical maneuver causing the accident was of crucial importance rather than the accident type. Therefore, a new categorization method was developed using the available information and the accidents in the LTCCS database were rearranged.

3.2 Accident Categorization

First, the involved heavy vehicles were categorized based on their role in the accident. These categories are:

1. Striking vehicle – loss of control
2. Striking vehicle – other than loss of control
3. Struck vehicle.

The target population of heavy vehicles in this study consisted of those which caused an accident due to loss of control, in other words, vehicles which belong to the first category. Herein, loss of control refers to lateral instability and turn-over (a rollover which is solely due to severe steering maneuver and consequent excessive lateral acceleration).

As mentioned previously, the main goal of this study was to determine the most common maneuvers causing loss of control. In this scope, real accident scenarios were studied to determine the critical maneuvers which lead to loss of control. The resulted categorization for critical maneuvers is:

1. Negotiating a curve
2. Turn at intersection
3. Avoidance maneuver
4. Lane change
5. Road edge recovery
6. Heavy braking on straight road
7. Avoidance maneuver or lane change in a curve
8. Going fast on a low friction straight road.

3.3 Accident Study Results

After re-classification of the database, loss of control accidents were analyzed with respect to the accident type, loss of control type, critical maneuver, vehicle combination type and different road characteristics. It was found that loss of control was associated with 19% of heavy vehicles involved in accidents. Turn-over was a more common type of loss of control than lateral instability; the former was associated with 55% of loss of control accidents, while the latter to 31%, the remaining

14% involved both lateral instability and turn-over. In other words, turn-over was attributed to 13% of all the heavy vehicles involved in traffic accidents and lateral instability to 9% of them.

Negotiating a curve was the main critical maneuver leading to loss of control (59%), followed by avoidance maneuver (11%) and road edge recovery (11%). Considering only lateral instability (excluding turn-over), negotiating a curve was still the main critical maneuver but with a lower contribution of 35%, while the avoidance maneuver involvement was increased to 22%. Adding the lateral instabilities associated with lane change to those caused by avoidance maneuver (considering their similarity), would result in total contribution of 28%.

After determining the main critical maneuvers causing lateral instability of heavy vehicles and consequently accidents, existing test maneuvers imitating these real world situations were compared to define the most suitable one for evaluating the lateral stability of heavy vehicles. A sine with dwell maneuver, which imitates a lane change/avoidance maneuver, was selected; since it can cause high side slip angles and yaw rates. For more information regarding the outcome of the accident analysis refer to Paper A.

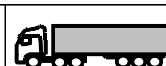
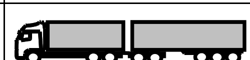
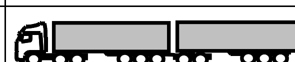
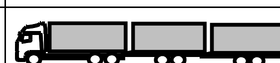
The application of the sine with dwell maneuver for evaluation of the lateral performance of heavy vehicle combinations on low friction surface was also investigated experimentally with three different combinations. By performing the maneuver on a low friction surface, rollover due to high lateral acceleration can be avoided and the vehicle can be pushed further towards lateral instability. Based on the test results, stability and maneuverability criteria originally proposed by NHTSA for passenger cars were adapted for heavy vehicle combinations [48]. The outcome of this experiment was presented at the 2008 IEEE international conference on vehicular electronics and safety, see the publication list.

Chapter 4

Lateral Performance of Heavy Vehicle Combinations

To obtain a better understanding of the lateral performance of heavy vehicle combinations and to specify the objectives of the control strategy that should be developed, a selection of various heavy vehicle combinations were studied and compared, utilizing the outcome of the accident analysis and standard performance measures. For these purposes, nine heavy vehicle combinations were chosen which included three common conventional combination vehicles in Europe, three existing longer combination vehicles common in some regions of Europe and three prospective longer combination vehicles, see Table 4.1. For better comparison and to avoid excessive diverse configuration features, the considered combinations consist of analogous loading units. The only exception is the truck-full trailer, which has a one-axle dolly and a semitrailer with a relatively short wheelbase.

Table 4.1 Investigated heavy vehicle combinations

Conventional Combination Vehicles	Tractor-Semitrailer (Tractor-ST)	16.5 m/40 ton	
	Truck-Center Axle Trailer (Tractor-CAT)	18.75 m/40 ton	
	Truck-Full Trailer (Truck-FT)	18.75 m/40 ton	
Existing Longer Combination Vehicles	Tractor-Link Semitrailer-Semitrailer (B-Double)	25.25 m/60 ton	
	Tractor-Semitrailer-Center Axle Trailer (Tractor-ST-CAT)	25.25 m/60 ton	
	Truck-Dolly-Semitrailer (Truck-Dolly-ST)	25.25 m/60 ton	
Prospective Longer Combination Vehicles	Tractor-Semitrailer-Dolly-Semitrailer (A-Double)	31.5 m/80 ton	
	Truck-Duo Center Axle Trailer (Truck-Duo CAT)	27.5 m/66 ton	
	Truck-Dolly-Link Semitrailer-Semitrailer (Truck-B-Double)	34 m/90 ton	

To study the lateral performance of the selected heavy vehicle combinations, their eigenstructure, frequency response and time response were analyzed. A brief summary of the outcomes of the study is presented in this chapter. It should be noted that in the performed simulations, the driver role is limited to a steering input and the driver interaction with the vehicle is not investigated. For description of vehicle modeling refer to Appendix B.

4.1 Eigenstructure

Eigenvalues of a system determine the convergence/divergence rate of the system to/from the steady state after a disturbance and can be used as an indicator of the lateral stability of heavy vehicle combinations. In Figure 4.1, eigenvalues of the combinations under study are plotted for a velocity range of 60 to 120 km/h. Eigenvalues of each combination are shown by a separate color but they are grouped into different clusters based on the different units in the combinations in order to make the analysis easier. It can be seen that the existence of a center-axle-trailer in the heavy vehicle combination results in higher sensitivity to disturbances.

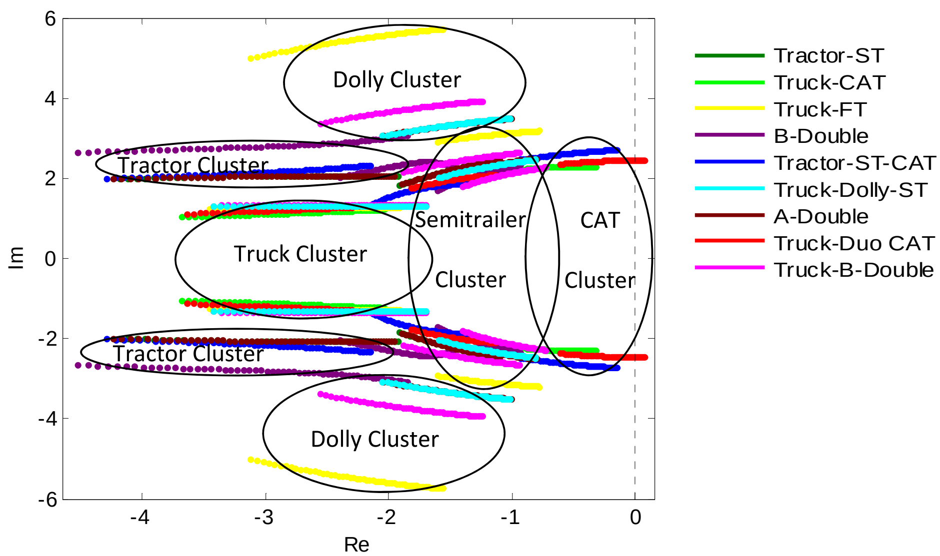


Figure 4.1 Eigenvalues of the heavy vehicle combinations for the speed interval of 60 to 120 km/h

Eigenvalues only determine the decay/growth rate of a system and it is the Eigenvectors that determine shape of the response. Thus, to obtain a better understanding of different response modes of a heavy vehicle combination, its modal composition should be studied. For instance, Figure 4.2 illustrates the modal composition of the truck-dolly-semitrailer combination, which is the main test combination in this thesis. The truck-dolly-semitrailer has three response modes:

- Mode 1. *Follow the lead*: the motion magnitudes of all the three units are comparable and the towed units are following the lead unit with a certain phase delay.
- Mode 2. *Wagging the tail*: the motion magnitudes of the lead and middle units are negligible compared with the last unit, which represents oscillation of the last unit.
- Mode 3. *Out of phase oscillation*: the motion magnitude of the lead unit is negligible compared with the towed units and the phase difference between motions of the towed units is almost 180˚.

The last two modes, which also have low damping ratios, show a natural tendency toward rearward amplifications of the lateral motions. For more information on the eigenstructure analysis of the truck-dolly-semitrailer combination refer to Paper D.

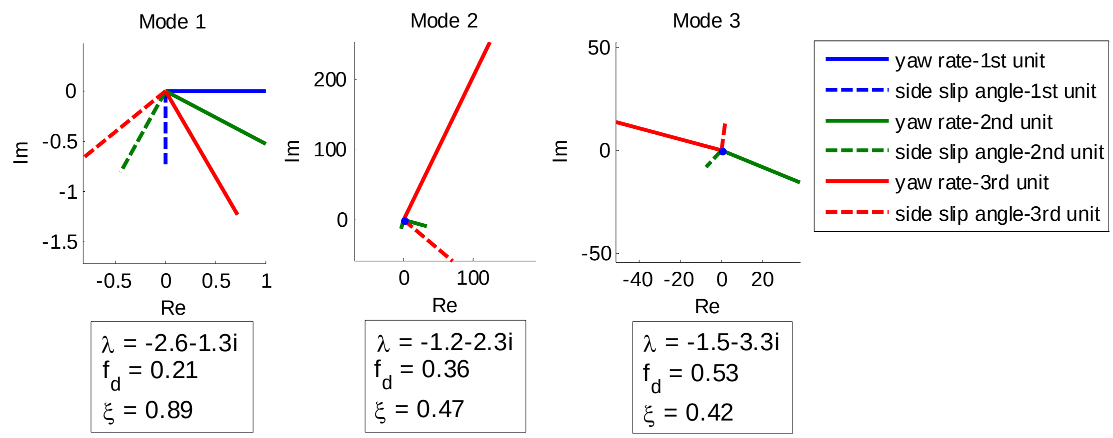


Figure 4.2 Modal composition of the truck-dolly-semitrailer combination (λ=eigenvalue, f_d=damped natural frequency [Hz], ζ=damping ratio)

4.2 Frequency Response

To investigate the effect of the steering input frequency on the vehicles performance and to find the critical frequencies, the frequency responses of the linear models of the vehicles were studied in the frequency range of interest. An example of the obtained results is shown in Figure 4.3, in which the rearward amplification of the yaw rate gain is plotted for all nine combinations. It can be seen that most of the vehicle combinations have a peak yaw rate RWA around 0.4 Hz and approaches unity at 0.5-0.6 Hz frequency.

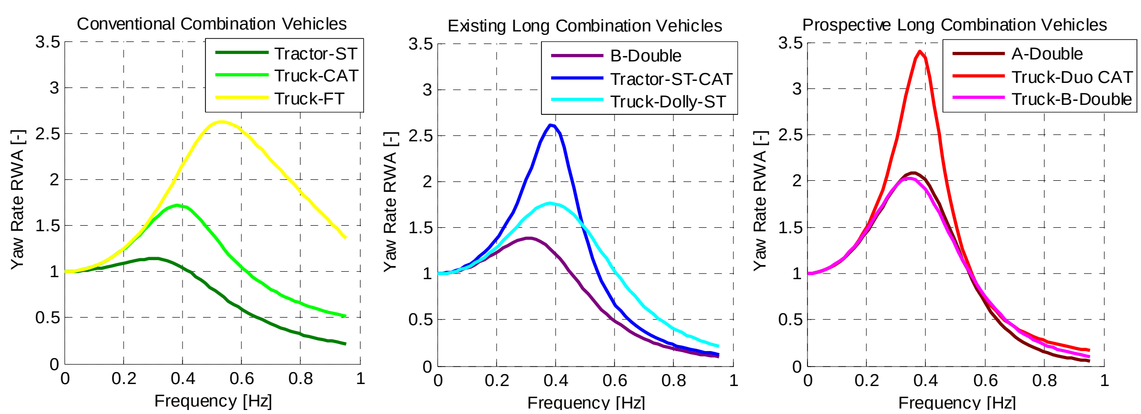


Figure 4.3 Rearward amplification of the yaw rate gain versus input frequency at a speed of 80 km/h

4.3 Time Response - Sine with Dwell Maneuver

In addition to the frequency response and eigenstructure analysis, the time responses of the selected heavy vehicle combinations in a sine with dwell maneuver were studied using nonlinear vehicle models. The maneuver, which was the outcome of the test maneuver study based on the accident analysis (Paper A), was simulated at a speed of 80 km/h, the speed limit for heavy vehicle combinations in Sweden. The steering input frequency was chosen based on the frequency response of the linear vehicle models, depicted in Figure 4.3. As it can be seen, most of the vehicle combinations have a peak yaw rate RWA around 0.4 Hz; thus, this frequency was chosen for the simulation of the sine with dwell maneuver. To have a fair comparison, the amplitude of the sine with dwell input for each vehicle combination was tuned so it would result in a lateral displacement of 3 m at the front axle of the vehicle. The lateral performances of the vehicles were compared using the described measures in chapter 2; the obtained results are summarized in Table 4.2.

Table 4.2 Summary of the lateral performance of the investigated heavy vehicle combinations

		Yaw Rate RWA	Offtracking	Yaw Damping Ratio
Conventional Combination Vehicles	Tractor-ST	1.28	0.8	0.54
	Truck-CAT	1.61	1.4	0.24
	Truck-FT	2.27	1.4	0.28
Existing Longer Combination Vehicles	B-Double	1.60	1.3	0.45
	Tractor-ST-CAT	2.61	2.4	0.14
	Truck-Dolly-ST	1.84	1.6	0.43
Prospective Longer Combination Vehicles	A-Double	2.47	2.8	0.36
	Truck-Duo CAT	3.63	3.4	0.03
	Truck-B-Double	2.26	2.4	0.37

It can be concluded that the vehicle configuration features plays a more important role than the vehicle total length in the lateral performance of the vehicle and there are conventional combinations with worse performance than LCVs. However, this does not mean that there is no need for improvement in the lateral performance of LCVs; even the B-double, which has the second best performance, has a noticeably inferior performance in comparison with the tractor-semitrailer. Besides, the tractor-semitrailer itself is the subject of several investigations attempting to improve its lateral performance, see for example [39, 42]. Therefore, there is a necessity to improve the lateral performance of heavy vehicle combinations at high speeds; in other words, it is necessary to decrease their yaw rate RWA and offtracking and increase their yaw damping ratio. The results provided in Table 4.2 indicate a correlation between these performance measures which is stronger between the yaw rate RWA and offtracking. This can be explained by the underlying dynamics as follows: The amplification of the lateral motions at the towed units of the heavy vehicle combinations is due to the significant time delay between the driver steering and generation of lateral forces at the towed units, which causes large yaw motions of the towed units and consequently lead to substantial side slip and offtracking. Therefore, the lateral performance of a heavy vehicle combination can be improved by timing adjustment of the lateral force generation at the tires of the towed units. This is further explained in the next chapter. For more information on the lateral performance analysis of the passive vehicles under study refer to Paper B.

Chapter 5

Steering-Based Lateral Performance Control

The poor lateral performance of heavy vehicle combinations at high speeds, namely the rearward amplification of lateral motions, is due to the significant time delay between the driver steering and generation of tire lateral forces at the towed units. This can be explained by the underlying dynamics, depicted in Figure 5.1; the driver's steering causes a lateral motion in the lead unit which consequently introduces a lateral force at the articulation joint of the succeeding unit. This is followed by large yaw motions of the unit which in turn results in substantial side slip causing large offtracking as well. It is after a substantial side slip at axles that the tires slip angles reach the required level for generation of large enough lateral forces to regulate the yaw motion. Therefore, if the time span between the driver steering and generation of tire lateral forces in the towed units is reduced, the RWA of yaw motions can be suppressed. This is achievable by active steering of the towed units of the heavy vehicle combination.

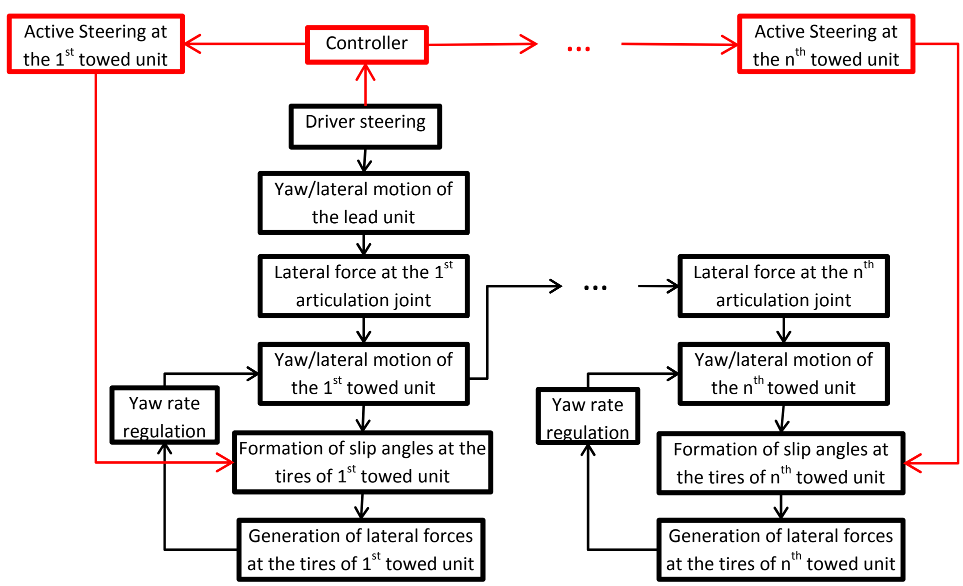


Figure 5.1 Dynamic flowchart of lateral/yaw motion of a *n*-unit heavy vehicle combination, the red parts show the active steering addition to the system (Inspired by the flow chart presented in [36])

By active steering, the required lateral force for yaw motion regulation can be generated earlier, before the yaw rates become large; therefore, the rearward amplification will be suppressed. Furthermore, since the lateral forces are generated due to imposed steer angles rather than side slip of tires, the side slip angle of the towed units and consequently offtracking will be decreased significantly without diminishing the maneuverability. Besides, since the yaw motion regulation of the towed units occur earlier and before the yaw rates become too large, the required tire lateral forces for this purpose may decrease to some extent and the roll stability may be improved as a byproduct.

The magnitude and timing of the imposed steering angles to the towed units' axles can be calculated by different approaches; in this thesis two control schemes are developed: *lead-unit-following control* and *eigenstructure-assignment control*. Both approaches utilize linear models of the heavy vehicle combination to predict and suppress the vehicle lateral motions, mainly the yaw rate rearward amplification and offtracking. The application of linear models is motivated by the outcome of the analysis on the passive vehicle dynamics, which revealed the agreement between the obtained results from the linear model and the nonlinear model of the heavy vehicle combinations.

Lead-Unit-Following Control – In this control scheme the required steer angles are determined so that the yaw rates of the towed units follow that of the lead unit; therefore, it is entitled the *lead-unit-following control*. This control scheme includes a dynamic feedforward based on the linear model, augmented by a proportional feedback to compensate for unmodeled dynamics, parameter uncertainties and disturbances. It only incorporates feedback gains from the yaw rates of the vehicle's units, and hence does not need state estimation. A generic formulation of this control scheme was developed and verified on the nonlinear models of all nine heavy vehicle combinations under study. Moreover, it was implemented on a truck-dolly-semitrailer test vehicle and was verified on a test track.

Eigenstructure-Assignment Control – The second control scheme is based on eigenstructure assignment theory, which is a state feedback control method that alters the modal composition of a system. To alter the modal composition of a system, both eigenvectors and eigenvalues are reformed; the alteration of eigenvalues is for modification of the decay/growth rate of the response and the alteration of eigenvectors is for shaping the response. For more information on the theory of eigenstructure assignment, refer to [49, 50]. This control scheme incorporates feedback gains from all vehicle states, that is, yaw rate and side slip angle of each unit; thus, it requires a state observer for implementation on a heavy vehicle combination. This approach was not investigated as thoroughly as the first approach; however, its feasibility and effectiveness were explored by simulations in form of a case study on a truck-dolly-semitrailer combination.

In addition to the two abovementioned main approaches, a control scheme based on Model Predictive Control (MPC) was also considered which is not discussed in this thesis. The relevant papers for the MPC approach can be found in the publication list in the beginning of this thesis. In the remaining of this chapter, the two main control schemes are presented in more details.

5.1 Design of the Lead-Unit-Following Control

The objective of this controller is to determine the required steer angle for the axles of the towed units so that their yaw rates follow that of the lead unit; in other words, the objective is to reduce the yaw rate RWA to one. It consists of a dynamic feedforward, designed based on the linear model of the vehicle, augmented by a proportional feedback to compensate for unmodeled dynamics, parameter uncertainties and disturbances [51]. Both parts are crucial for proper functioning of the controller; without the feedforward part the controller will not be responsive enough while the feedback part is necessary to insure robustness and mitigate sensitivity to model uncertainties. Considering the controller structure, the steering control inputs for the towed units, δ_i , can be stated as:

$$\delta_i = \delta_{i,ff} + \delta_{i,fb} \text{ for } i = 2, 3{,}4, \tag{5.1}$$

where $\delta_{i,ff}$ denotes the steering demand by the feedforward part and $\delta_{i,fb}$ denotes the steering demand by the feedback part. Here the required steer angle equations are derived for a combination of maximum four units, but the provided procedure can be easily extended to longer combinations.

5.1.1 Feedforward Control

Using the linear model described in Appendix B, the relation between the yaw rates of each unit and the steer inputs can be written as in

$$r_i = \sum_{j=1}^{4} G_{ri\_\delta j}\delta_j \text{ for } i = 1, 2, 3{,}4, \tag{5.2}$$

where $G_{ri\_\delta j}$ is the transfer function between the yaw rate of the *i*th unit and steer angle of the *j*th unit. It should be noted that $G_{ri\_\delta j} \approx 0$ for $i < j$ and can be neglected.

The basic idea of the feedforward control is to steer the axles of the towed units in a way that their yaw rates follow the yaw rate of the truck with a time lag. This can be expressed as

$$r_{i_{des}} = r_1 e^{-\tau_i s} = e^{-\tau_i s} G_{r1\_\delta 1}\delta_1 \text{ for } i = 2, 3{,}4, \tag{5.3}$$

where τ_i is the time lag between the yaw rates of the lead unit and the *i*th unit, chosen based on the passive vehicle dynamics. The time lag is approximated using second order Padé approximation as in

$$e^{-\tau_i s} = \frac{1-\frac{\tau_i}{2}s+\frac{\tau_i^2}{12}s^2}{1+\frac{\tau_i}{2}s+\frac{\tau_i^2}{12}s^2} = P_{di} \text{ for } i = 2, 3{,}4. \tag{5.4}$$

To achieve the desired dynamics in equation 5.3 using the underlying steering relations in equation 5.2, the feedforward control law can be written as

$$\delta_{2,ff} = \frac{(P_{d2}G_{r1\_\delta 1} - G_{r2\_\delta 1})}{G_{r2\_\delta 2}}\delta_1 = G_{21}\delta_1, \tag{5.5}$$

$$\delta_{3,ff} = \frac{(P_{d3}G_{r1\_\delta 1} - G_{r3\_\delta 1} - G_{21}G_{r3\_\delta 2})}{G_{r3\_\delta 3}}\delta_1 = G_{31}\delta_1, \tag{5.6}$$

$$\delta_{4,ff} = \frac{(P_{d4}G_{r1\_\delta 1} - G_{r4\_\delta 1} - G_{21}G_{r4\_\delta 2} - G_{31}G_{r4\_\delta 3})}{G_{r4\_\delta 4}}\delta_1 = G_{41}\delta_1. \tag{5.7}$$

The derived transfer functions for feedforward control, G_{i1} , are based on the linear model of the vehicle combination which is valid for constant speed maneuvering. Thus, in order to implement them on the vehicle, a continuous gain scheduling method was used with vehicle speed as the scheduling variable. In other words, the transfer functions were designed for specified speeds and a linear interpolation was used for operating speeds between the design speeds to prevent discontinuity. This can be expressed mathematically as follows.

Considering the mapping between the design speeds, u^j, and corresponding transfer functions for the feedforward control, G_{21}^j, G_{31}^j and G_{41}^j:

$$\begin{bmatrix} u^1 \\ u^2 \\ \vdots \\ u^n \end{bmatrix} \rightarrow \begin{bmatrix} \{G_{21}^1, G_{31}^1, G_{41}^1\} \\ \{G_{21}^2, G_{31}^2, G_{41}^2\} \\ \vdots \\ \{G_{21}^n, G_{31}^n, G_{41}^n\} \end{bmatrix}, \tag{5.8}$$

for any operating speed u, such that $u^j < u < u^{j+1}$, the transfer functions for the feedforward control can be expressed as

$$\delta_{i,ff} = \beta G_{i1}^{j+1}\delta_1 + (1-\beta)G_{i1}^{j}\delta_1 \;\text{ for }\; i = 2{,}3{,}4, \tag{5.9}$$

where

$$\beta = \frac{u-u^j}{u^{j+1}-u^j}\,. \tag{5.10}$$

5.1.2 Feedback Control

For the feedback part, a decentralized control structure was proposed. It consists of proportional controllers based on the difference between the desired yaw rate (delayed yaw rate of the first unit with the same time lags used in the FF part) and the actual yaw rate of the controlled unit as in

$$\delta_{i,fb} = k_i\left(r_{i_{des}} - r_i\right) = k_i\left(r_1 e^{-\tau_i s} - r_i\right). \tag{5. 11}$$

The proportional feedback gains, k_i, were initially determined using generalized Nyquist's stability criterion for a multi-input-multi-output system, considering a gain margin of two (see Paper C). The achieved gains were then tuned based on the performance and stability of the controller in simulation with the nonlinear model. It should be noted that in the research presented in this thesis, this procedure was only performed for the truck-dolly-semitrailer combination and the same gains were used for all vehicle combinations. In practice, the feedback gains should be tuned for each vehicle combination for better performance.

5.2 Effectiveness of the Lead-Unit-Following Control

The effectiveness of the *lead-unit-following control* was verified by simulations as well as experimental tests, both of which showed significant improvements in the lateral performance of the controlled vehicles. A summary of the obtained results is provided in this section.

5.2.1 Verification for Various Heavy Vehicle Combinations

To verify the controller effectiveness, its performance for all nine heavy vehicle combinations under study was simulated using the nonlinear models of the vehicles. The controller performance for each combination was evaluated in a sine with dwell maneuver at different speeds. To have a fair comparison, the amplitude of the sine with dwell input for each vehicle combination was tuned stepwise so it would result in a lateral displacement of 3 m (a normal lane change) at the front axle of the first unit. The steering input frequency was also adapted for each speed so that the longitudinal displacement during the lane change would be the same at different speeds. The dwell time in steering was half a second for all cases. The achieved reductions in the yaw rate RWA and offtracking are plotted in Figure 5.2 and Figure 5.3, respectively. Significant improvements are attained; the yaw rate RWA is reduced to values below 1.5 for all combinations at all speeds, except for A-double combination at a speed of 100 km/h. The offtracking is decreased to 0.7 m or less for all combinations up to speed of 90 km/h. At a speed of 100 km/h the offtracking is larger than 1 m for some combinations; however, even for these cases more than 60% reduction in offtracking is achieved.

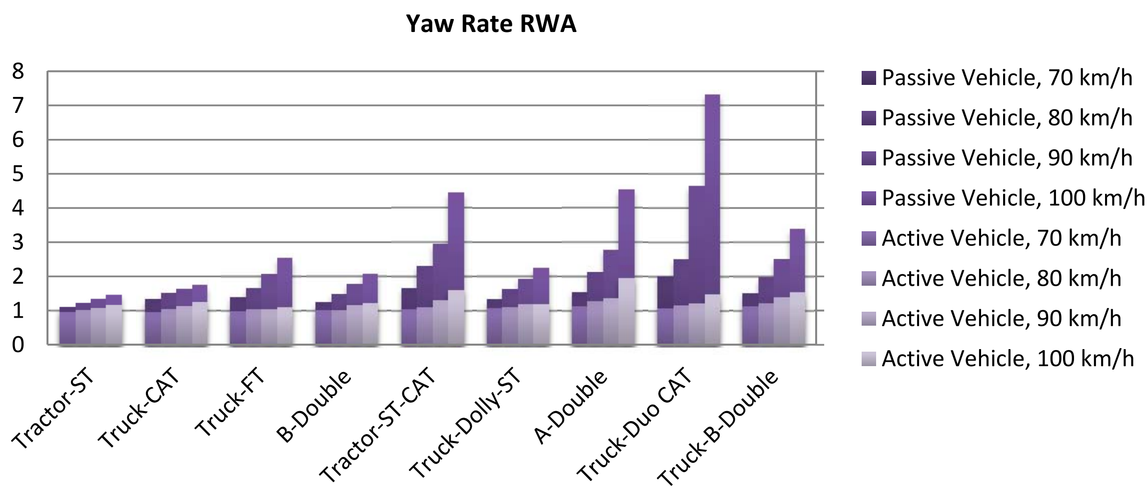


Figure 5.2 Achieved reduction in the yaw rate RWA by the controller in a sine with dwell maneuver

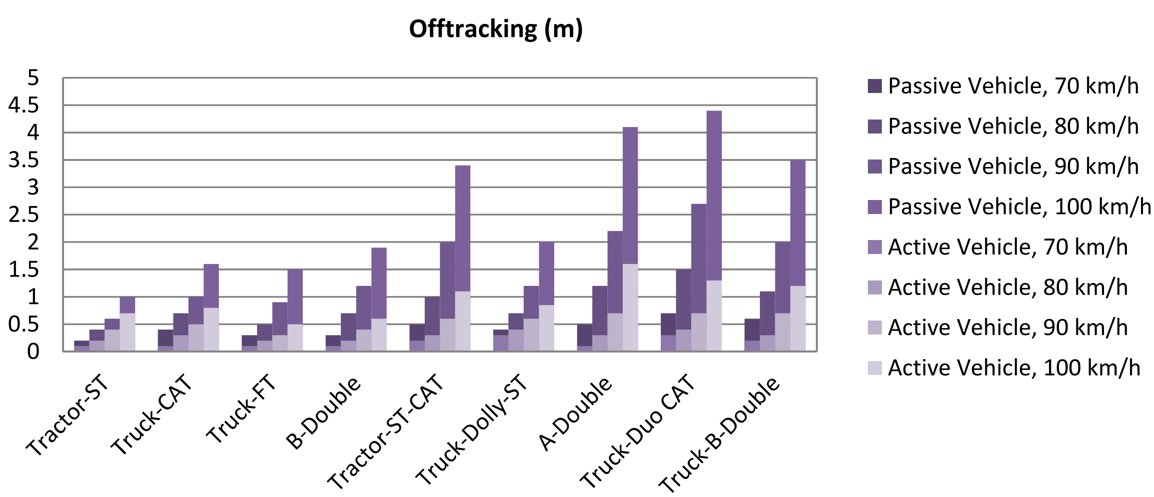


Figure 5.3 Achieved reduction in the offtracking by the controller in a sine with dwell maneuver

In general, the obtained results indicate more reductions in the yaw rate rearward amplification and offtracking at higher speeds, and for the combinations with worse passive performance; this is explainable by the larger potential for improvements for these cases. It should be mentioned that the controller does not deteriorate the responsiveness of the heavy vehicle combinations; in other words, the path of the lead unit is hardly affected by the controller in any of the performed simulations. Furthermore, the controller also reduces the lateral acceleration RWA moderately and therefore decreases the rollover risk. For more information refer to Paper B.

Simulation results provided in this section and in Paper B verify the effectiveness of the *lead-unit-following control* and its viability for various heavy vehicle combinations. For further investigation of the controller efficiency and robustness under various driving conditions and presence of parameter uncertainties, extensive simulations were performed for the main test combination in this thesis, namely the truck-dolly-semitrailer; the outcomes of the investigation are presented in following subsections.

5.2.2 Verification for Various Driving Conditions

The controller effectiveness under various driving conditions was investigated for a truck-dolly-semitrailer. The changing factors were road surface friction, presence of brake intervention and loading condition. Table 5.1 summarizes the considered variations for each factor. It should be noted that in all loading conditions, the controller has the accurate vehicle parameters; the controller sensitivity to parameter uncertainties is discussed in next subsection.

The simulated maneuver was a sine with dwell maneuver with speed of 90 km/h for high friction surface and 70 km/h for low friction surface. The chosen speeds have a 10 km/h difference with the maximum allowed speed for heavy vehicle combination in Sweden, which is 80 km/h. The steering input magnitude was adjusted so it results in a lateral displacement of 3 m. The frequency was chosen based on the frequency response of the linear model of the vehicle combination and close to the frequency at which the yaw rate gain RWA has a peak for each speed; the chosen frequencies were 0.4 and 0.3 Hz for high friction and low friction surface, respectively. The dwell time in steering was half a second.

Table 5.1 Considered variations in the simulated sine with dwell maneuver, representing 20 scenarios

Road Surface Friction	High friction (μ=1) Low friction (μ=0.2)
Brake Intervention	ESC off ESC on
Loading Condition	Fully loaded (13/27 ton)* Half loaded, even distribution (10/10 ton) Half loaded, rear heavy (6.5/13.5 ton) Half loaded, front heavy (13.5/6.5 ton) No load (0/0 ton)

*Payloads of the truck and the semitrailer, respectively

To investigate the effect of presence of brake interventions, a simple Electronic Stability Control (ESC) system was modeled which applies differential braking on the lead unit so that its yaw rate follows a desired value within a certain bound. If the ESC intervenes, it also applies braking on the towed units, so that each unit would decelerate at a rate of 1 m/s^2, unless the available friction does not suffice. The choice of the deceleration magnitude was based on the study presented in [52]. It should be emphasized that the modeled ESC in this study is too simple to be compared by the existing ESC on the heavy vehicles on roads. Nonetheless, the purpose of this study is not to evaluate the efficiency of the ESC, but to evaluate the effects of brake intervention on the effectiveness of the designed controller and the simple modeled ESC gives an insight to this matter. More details on the modeled ESC are provided in Appendix C.

The achieved results are summarized in Table 5.2 and Table 5.3. It can be concluded that:

- Considering the passive vehicle, the fully loaded vehicle has the worst performance in terms of yaw rate RWA and offtracking. The difference between the unladen and half loaded vehicles is not substantial; however, moving the load backwards from the truck to the semitrailer impairs the lateral performance. As expected the situation is more critical on the low surface friction, for which both yaw rate RWA and offtracking are larger.
- ESC intervention decrease the peak yaw rate of each unit but does not affect the yaw rate RWA; however, it reduces the offtracking to some extent. This is due to the fact that ESC reduces the severity and speed of the maneuver.
- The yaw rate RWA is reduced to almost similar levels independent of the loading condition; in other words, the attained reduction in the yaw rate RWA is higher for the cases with worse performance in the passive case. The ESC intervention does not affect the performance of the controller.
- The offtracking is reduced to equal or less than 0.7 m on high friction surface and 1.2 m on low friction surface. The achieved improvement in offtracking is on average about 55% and it is not affected by the loading condition, surface condition or ESC intervention considerably. However, the achieved improvement is slightly less for the low friction surface in comparison with the high friction surface when ESC is on. This can be due to the limited available friction and also the fact that ESC intervention reduces the offtracking to some extent and consequently decreases the improvement potential by the controller; it can be seen that the resulted offtracking is almost the same for ESC-off and ESC-on cases on low friction surface.
- The peak side slip angle of the semitrailer is reduced on average by 76% on high friction surface, the gained improvement is less on low friction surface due to the limited available friction, but is still substantial, on average 59%.

In Table 5.4 the attained reduction in the lateral acceleration RWA is provided which shows a moderate reduction on the high friction surface. Therefore, the rollover risk is decreased by the controller as a byproduct. An insignificant increase in the lateral acceleration RWA occurs for some cases on the low friction surface; however, rollover is not an issue on a low friction surface due to the limited tire lateral forces.

Table 5.2 The controller effectiveness when ESC is off

ESC-off		Yaw Rate RWA			Offtracking			Semitrailer Side Slip Angle		
		Passive	Active	Progress	Passive	Active	Progress	Passive	Active	Progress
High Friction	No Load	1.8	1.14	37%	1.37	0.58	58%	4.68	1.21	74%
	Half Loaded - Even	1.84	1.14	38%	1.43	0.62	57%	4.69	1.13	76%
	Half Loaded – Rear Heavy	1.89	1.15	39%	1.53	0.64	58%	5.11	1.18	77%
	Half Loaded – Front Heavy	1.79	1.13	37%	1.36	0.62	54%	4.37	1.12	74%
	Fully Loaded	1.99	1.18	41%	1.7	0.71	58%	6.01	1.46	76%
Low Friction	No Load	2.31	1.18	49%	1.78	0.83	53%	6.86	2.77	60%
	Half Loaded - Even	2.37	1.19	50%	1.92	0.91	53%	7.71	2.98	61%
	Half Loaded – Rear Heavy	2.72	1.21	56%	2.21	1.03	53%	8.66	3.58	59%
	Half Loaded – Front Heavy	2.06	1.17	43%	1.68	0.83	51%	6.78	2.62	61%
	Fully Loaded	2.97	1.24	58%	2.51	1.17	53%	10.3	4.22	59%

Table 5.3 The controller effectiveness when ESC is on

ESC-on		Yaw Rate RWA			Offtracking			Semitrailer Side Slip Angle		
		Passive	Active	Progress	Passive	Active	Progress	Passive	Active	Progress
High Friction	No Load	1.88	1.1	41%	1.07	0.42	61%	3.64	0.9	75%
	Half Loaded - Even	1.84	1.11	40%	1.15	0.48	58%	3.69	0.85	77%
	Half Loaded – Rear Heavy	1.86	1.1	41%	1.17	0.46	61%	3.8	0.83	78%
	Half Loaded – Front Heavy	1.85	1.1	41%	1.14	0.48	58%	3.67	0.88	76%
	Fully Loaded	2.06	1.14	45%	1.4	0.54	61%	4.84	1.02	79%
Low Friction	No Load	2.3	1.17	49%	1.75	0.87	50%	6.48	2.88	56%
	Half Loaded - Even	2.32	1.19	49%	1.84	0.89	52%	7.31	2.91	60%
	Half Loaded – Rear Heavy	2.6	1.21	53%	2.07	1	52%	8.07	3.44	57%
	Half Loaded – Front Heavy	2.04	1.17	43%	1.62	0.82	49%	6.48	2.58	60%
	Fully Loaded	2.82	1.23	56%	2.34	1.12	52%	9.55	4	58%

Table 5.4 The controller effect on the lateral acceleration RWA

Lateral Acceleration RWA		ESC off			ESC on		
		Passive	Active	Progress	Passive	Active	Progress
High Friction	No Load	1.71	1.34	22%	1.72	1.36	21%
	Half Loaded - Even	1.58	1.33	16%	1.6	1.32	18%
	Half Loaded – Rear Heavy	1.57	1.35	14%	1.61	1.34	17%
	Half Loaded – Front Heavy	1.58	1.31	17%	1.62	1.33	18%
	Fully Loaded	1.53	1.33	13%	1.59	1.38	13%
Low Friction	No Load	1.19	1.19	0%	1.17	1.18	-1%
	Half Loaded - Even	1.16	1.17	-1%	1.16	1.17	-1%
	Half Loaded – Rear Heavy	1.13	1.15	-2%	1.13	1.15	-2%
	Half Loaded – Front Heavy	1.19	1.18	1%	1.19	1.18	1%
	Fully Loaded	1.1	1.12	-2%	1.15	1.12	3%

5.2.3 Sensitivity to Parameter Uncertainties

For further investigation of the controller robustness, its sensitivity to vehicle parameter uncertainties was investigated by simulating the described sine with dwell maneuver for the most severe case: a fully loaded truck-dolly-semitrailer. For this purpose, the vehicle parameters in the controller were varied independently by ± 20% from the actual values; the considered vehicle parameters are mass, moment of inertia and location of center of gravity of the truck and the semitrailer units and cornering stiffness of the tires. It should be noted that the air brake system of heavy vehicles provides air pressure monitoring that can be utilized to estimate the axle loads and consequently the mass and center of gravity of each unit to be used in the controller. Therefore, the moment of inertia of the vehicle units and tire cornering stiffness are the most uncertain vehicle parameters.

Tire cornering stiffness

Figure 5.4 illustrates the effect of uncertainties in tire cornering stiffness on the controller performance and the achieved yaw rate RWA. It can be seen that the yaw rate RWA is reduced significantly on both high and low friction surfaces even with incorrect tire cornering stiffness values in the controller. In Figure 5.5, effect of tire parameter uncertainty on the achieved offtracking is depicted; the controller performance does not vary considerably on high friction. However, the sensitivity to tire parameter uncertainty is higher on low friction surface.

Figure 5.4 Effect of tire cornering stiffness (Cα) uncertainties on the achieved yaw rate RWA

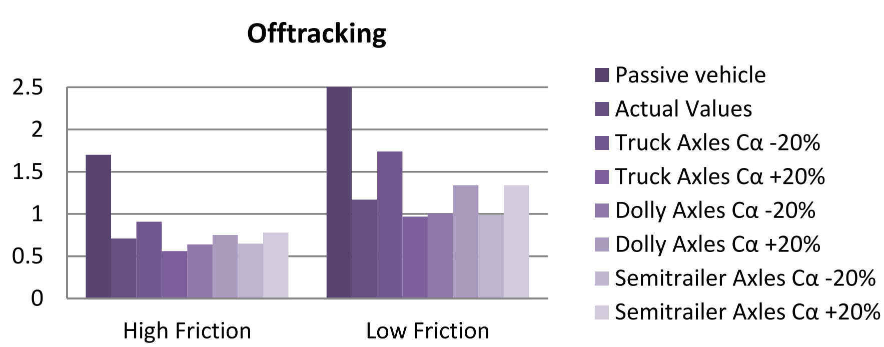


Figure 5.5 Effect of tire cornering stiffness (Cα) uncertainties on the achieved offtracking

Mass

Effects of vehicle mass uncertainties on the controller performance are shown in Figure 5.6 and Figure 5.7. It can be concluded that the controller is quite robust with respect to vehicle mass uncertainties and reduce both yaw rate RWA and offtracking significantly. The achieved improvement in the offtracking varies more with variation in vehicle mass in comparison with the yaw rate RWA.

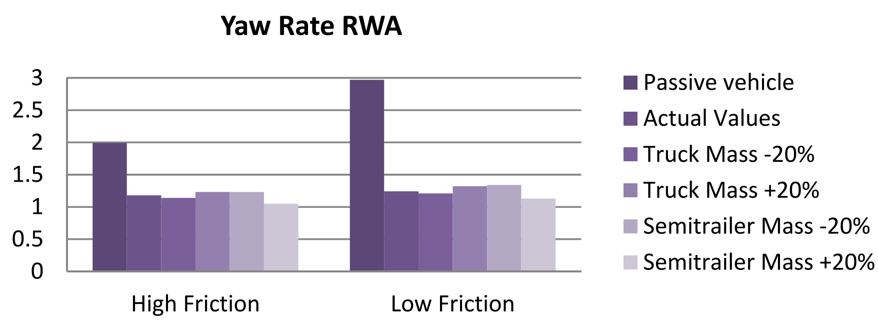


Figure 5.6 Effect of vehicle mass uncertainties on the achieved yaw rate RWA

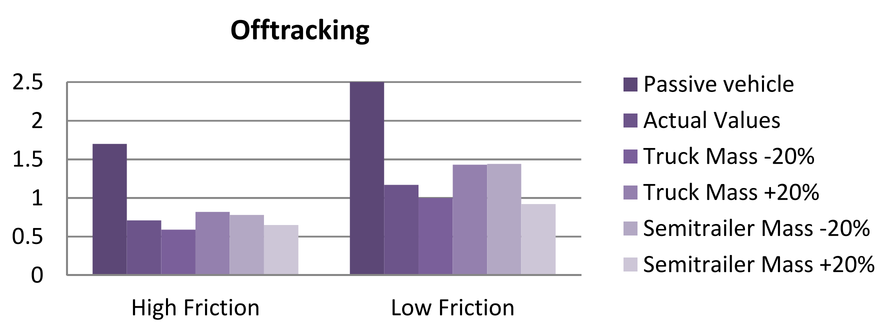


Figure 5.7 Effect of vehicle mass uncertainties on the achieved offtracking

Moment of inertia

The controller sensitivity to uncertainties in vehicle inertia, which is illustrated in Figure 5.8 and Figure 5.9, is the least among the investigated cases. The yaw rate RWA is reduced to 1.3 or less in all cases and the offtracking is less than 0.8 m on high friction surface and 1.3 m on low friction surface.

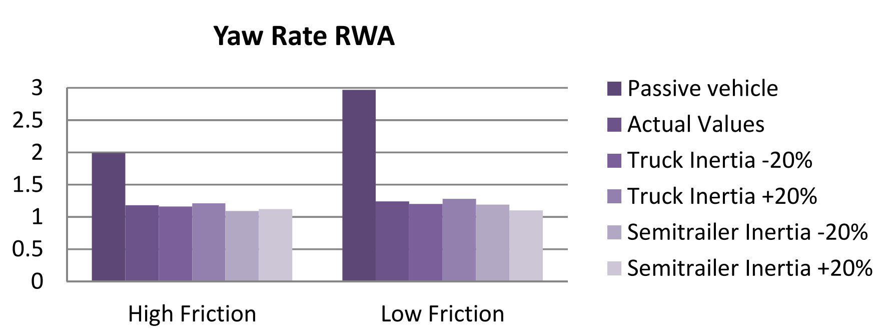


Figure 5.8 Effect of vehicle inertia uncertainties on the achieved yaw rate RWA

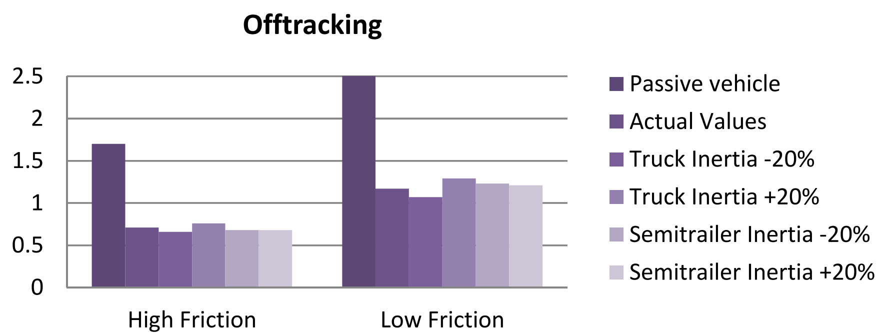


Figure 5.9 Effect of vehicle inertia uncertainties on the achieved offtracking

Location of center of gravity

Considering the differences between the wheelbase length of the truck and the semitrailer, the considered amount of uncertainty for the location of center of gravity (CG) is 0.5 m for the truck and 1 m for the semitrailer. The results of analysis of the controller sensitivity to uncertainties in CG location are shown in Figure 5.10 and Figure 5.11. It can be seen that estimating the truck CG location further rearward than the actual position reduces the controller effectiveness on low friction surface. However, this is not critical considering the fact that the location of center of gravity can be calculated using the measured axle loads by the air brake system in heavy vehicles. Nevertheless, since this sensitivity to uncertain CG location is only in rearward direction, a bias towards forward direction in estimation of the truck center of gravity can be beneficial.

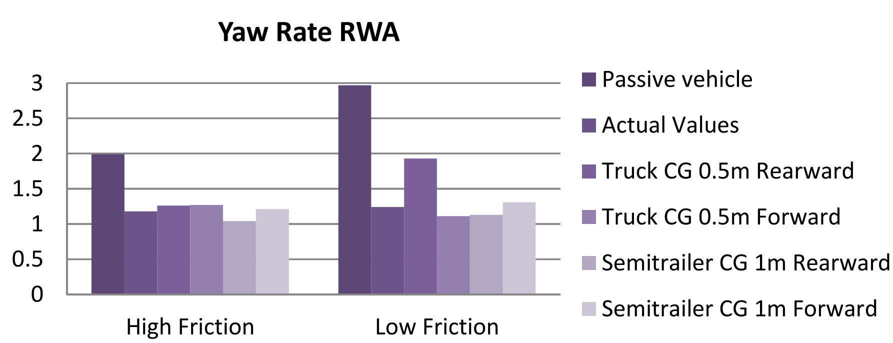


Figure 5.10 Effect of CG location uncertainties on the achieved yaw rate RWA

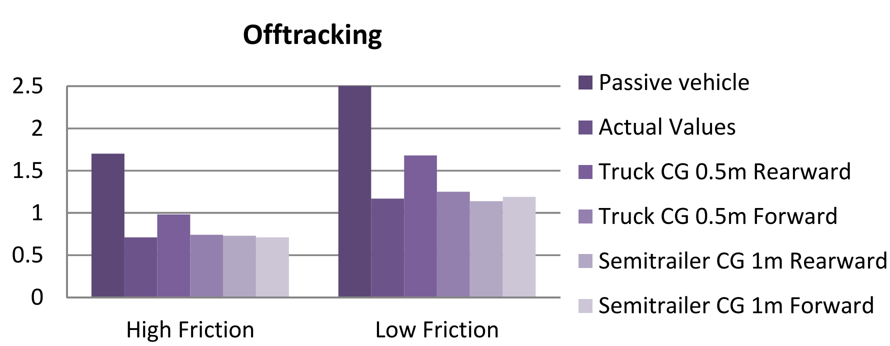


Figure 5.11 Effect of CG location uncertainties on the achieved yaw rate RWA

The presented simulation results in this section confirm that the controller is quite robust and improves the lateral performance of the vehicle considerably even with incorrect vehicle parameters. However, the achievable reduction in offtracking varies more with parameter uncertainties in comparison with achievable reduction in the yaw rate RWA. Furthermore, the controller is more sensitive to vehicle parameter uncertainties on a low friction surface in comparison with the high friction surface.

5.2.4 Verification on Test Track

The controller effectiveness was also verified experimentally with a truck-dolly-semitrailer test vehicle, in collaboration with Volvo Trucks and Cambridge Vehicle Dynamics Consortium. A series of single lane change and double lane change maneuvers on a dry road were performed using a path following steering robot for better repeatability. In addition to the common sensors used in the controller algorithm, such as yaw rate sensors and steer angles, the test vehicle was also equipped with a video-based tracking system which provided a path error measurement. Thus, it was possible to evaluate the controller effect on both yaw rate RWA and offtracking, which confirmed the simulation results. Here some sample test results for the single lane change maneuver are presented, which also illustrate the controller interaction with the existing ESC on the vehicle. More information about the conducted tests can be found in Paper C.

Figure 5.12 shows a sample driver steer input in the single lane change maneuver, which is 55 m long by 3.5 m wide and is performed at a speed of 80 km/h. Since this is a path following maneuver, it includes more steering reversals than the one sinusoidal cycle input used in the simulations. In Figure 5.13 the dynamic response of the test vehicle for the case when both the ESC and the controller are off is illustrated. The steering wheel angle and yaw rates are plotted as they were logged without filtering to show the level of sensor noise and offset that the controller has to deal with, while the lateral acceleration plot shows the filtered accelerometers outputs. It should be noted that the lateral accelerations were not measured at the units' center of gravity, the truck accelerometer was mounted 1.5 m behind the front axle, the dolly accelerometer was mounted 1.69 m ahead of its front axle and in the semitrailer the accelerometer was mounted 1.07 m ahead of its front axle.

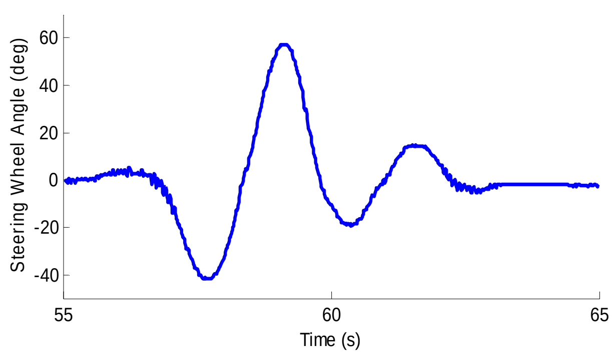


Figure 5.12 Driver steer input for the SLC maneuver

Figure 5.14 shows the test vehicle dynamic response when the ESC is on. ESC intervenes and applies braking forces during the second half of the steering maneuver; however, it is triggered for rollover protection, not yaw control and its function is mostly speed reduction (from 80 to 63 km/h) which results in a drop in RWA as well but does not affect the offtracking. These results demonstrate that the ESC System is not sufficient for suppressing the lateral motions of the towed units in an LCV and the control strategy should be extended to the towed units for enhanced lateral performance.

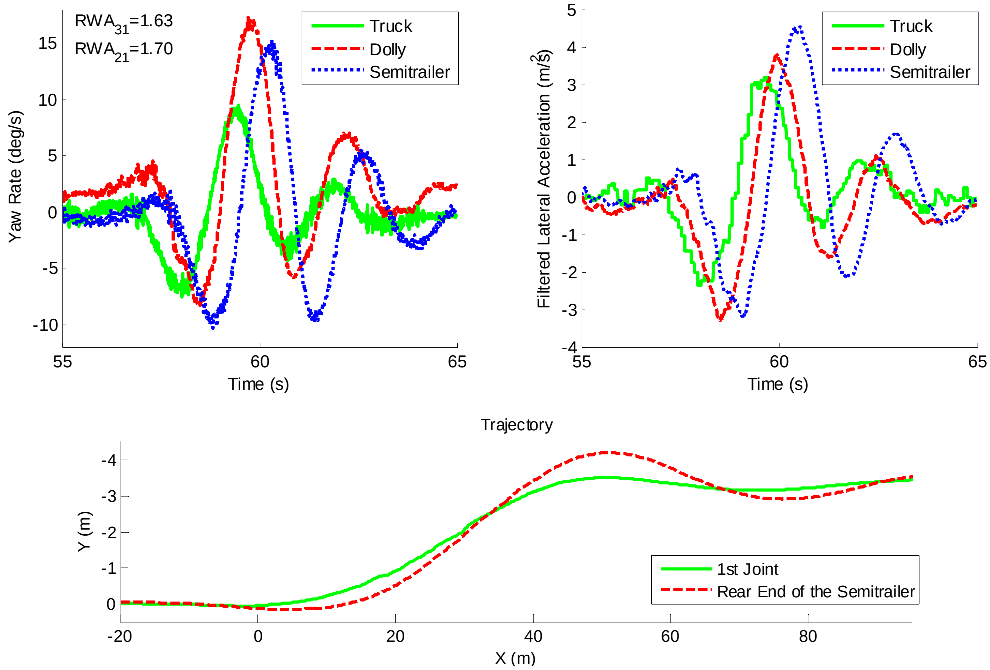


Figure 5.13 Dynamic response of the unsteered test vehicle with ESC off, in the SLC maneuver

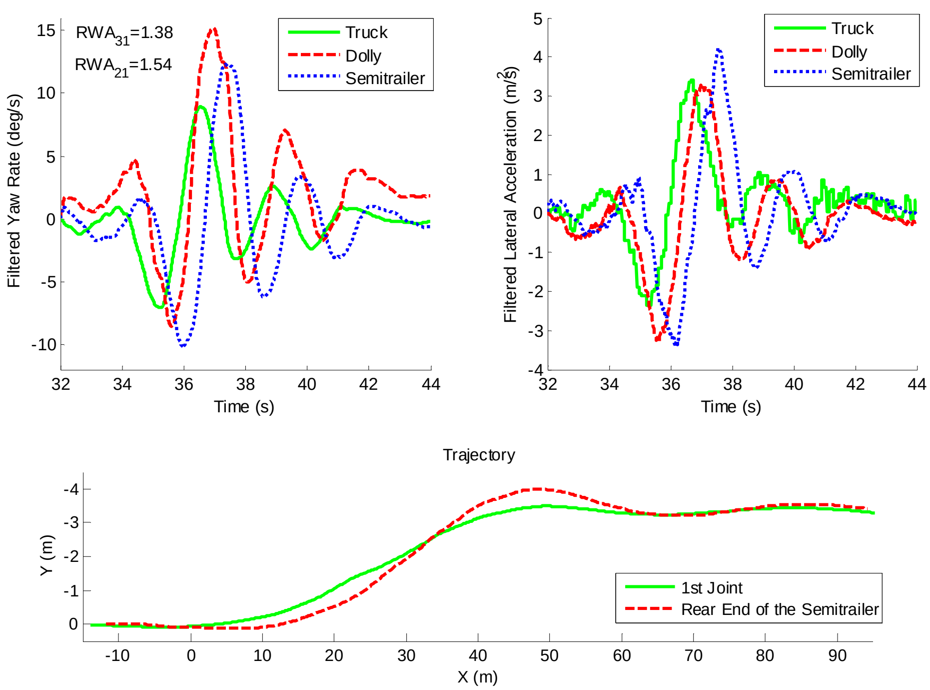


Figure 5.14 Dynamic response of the unsteered vehicle with ESC on, in the SLC maneuver

The dynamic response of the test vehicle with the controller is illustrated in Figure 5.15, which verifies the controller effectiveness; the yaw rate RWA is reduced to 1.06 from 1.63 of the unsteered case and the offtracking is decreased significantly. Unfortunately the cameras were not functioning for any of the test runs for the steered vehicle with ESC off. However, comparison of other vehicle states for two cases of steered vehicle with ESC on and off confirms that although the ESC applies braking forces and reduces the vehicle speed in the second half of the maneuver, it does not affect the overall vehicle response significantly. Thus, the illustrated trajectory for the steered vehicle with ESC on in Figure 5.16 is a fair replacement for the missing data.

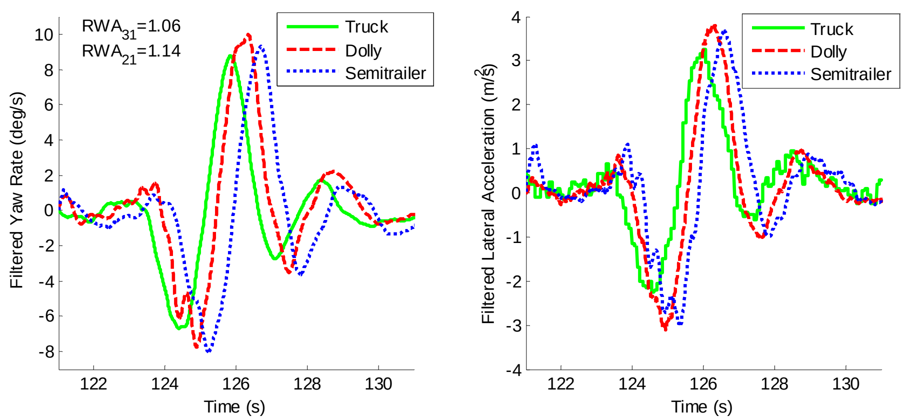


Figure 5.15 Dynamic response of the steered vehicle with ESC off, in the SLC maneuver

The performance of the controller was also tested with ESC on to verify their compatibility; the obtained result is shown in Figure 5.16. As it can be seen, the ESC intervention does not deteriorate the controller functionality and both yaw rate RWA and offtracking are reduced significantly in comparison with the unsteered case.

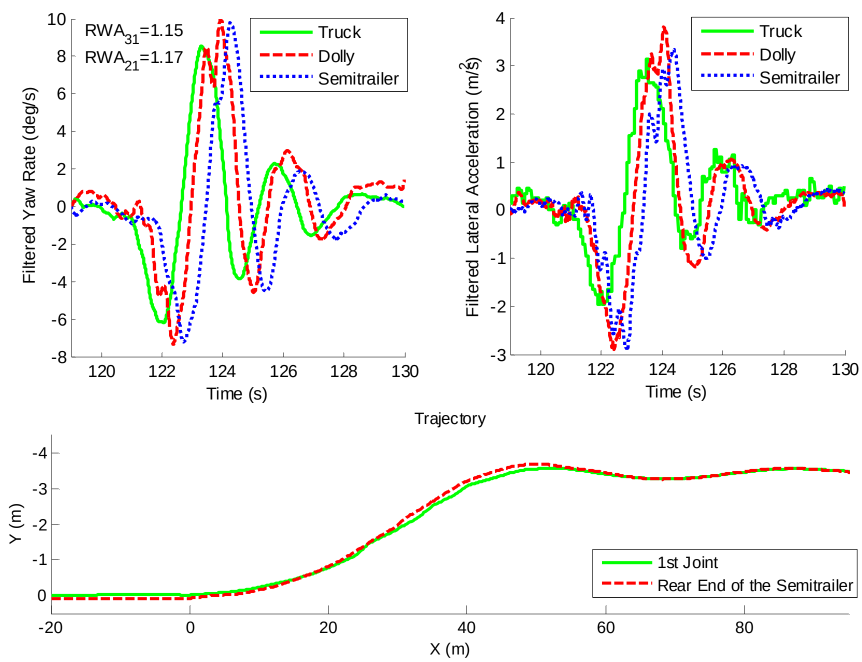


Figure 5.16 Dynamic response of the steered vehicle with ESC on, in the SLC maneuver

The path error for the test cases presented in this section, with functioning cameras, can be compared in Figure 5.17. The term path error is used instead of offtracking due to the fact that the cameras were not mounted at the front and rear axles, but at the first joint and the rear end of the semitrailer. Thus, the lateral deviation between path of the first joint and path of the rear end of the vehicle was measured.

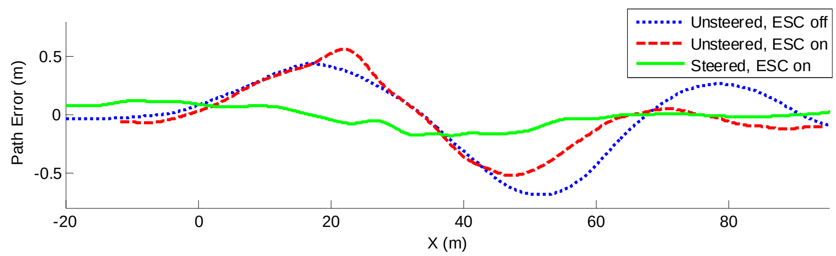


Figure 5.17 Comparison of path error for different test cases (with functioning cameras) of the SLC maneuver

5.3 The Eigenstructure Assignment Control

The feasibility of eigenstructure assignment for the control of lateral performance of a 3-unit LCV, namely the truck-dolly-semitrailer combination, was investigated. The considered control inputs to the vehicle are three steering angles for rear axle of the truck, axles of the dolly and axles of the semitrailer. These three control inputs provide a limited control authority and the whole eigenstructure of the vehicle cannot be assigned arbitrarily. Thus, an alternative approach was used by modifying the key features of the eigenstructure to achieve the desired vehicle performance.

5.3.1 Partial Eigenstructure Assignment

Considering the modal composition of the truck-dolly-semitrailer presented in Chapter 4, it is desirable to increase the damping ratios of the two conjugate modes which show a natural tendency toward amplification of motions in the towed units. However, a simple eigenvalue (pole) placement strategy will also affect the eigenvectors and change the shape of vehicle dynamic response, including the steady state gains, arbitrarily. Therefore, to be able to define and shape the desired steady state gains of the vehicle response with respect to the driver input, the vehicle linear model was expanded to a 7-state model by inclusion of driver steering angle as a state. The expanded model has an additional mode with zero eigenvalue and real eigenvector, which is termed the “zero mode”.

Considering the expanded system model, it was possible to alter the eigenvalues of the system, as well as the eigenvector of the zero mode. The eigenvalues were modified so that the damping ratios of the two troublesome conjugate modes were increased for better lateral performance. The elements of the eigenvector of the zero mode were defined so that the steady state yaw rate gains were kept as the passive vehicle to maintain the vehicle responsiveness, but the steady state side slip gains were decreased to reduce the offtracking.

5.3.2 Controller Effectiveness

The effectiveness of the *eigenstructure-assignment control* was verified by simulation for the truck-dolly-semitrailer and substantial reductions in the yaw rate RWA and offtracking were attained; see Paper D for detailed results. In Figure 5.18 and Figure 5.19, the effectiveness of this control scheme is compared with the *lead-unit-following control* in a 3 m lane change simulated by a since with dwell maneuver at a speed of 80 km/h and steering input frequency of 0.4 Hz. It can be seen that the *eigenstructure-assignment control* has a slightly better performance.

It should be emphasized that the obtained improvements by this control scheme are result of a short study on the feasibility of eigenstructure assignment for the lateral performance control of heavy vehicle combinations; hence, by further development of this control scheme and identification of the ideal eigenstructure, even better performance may be achievable.

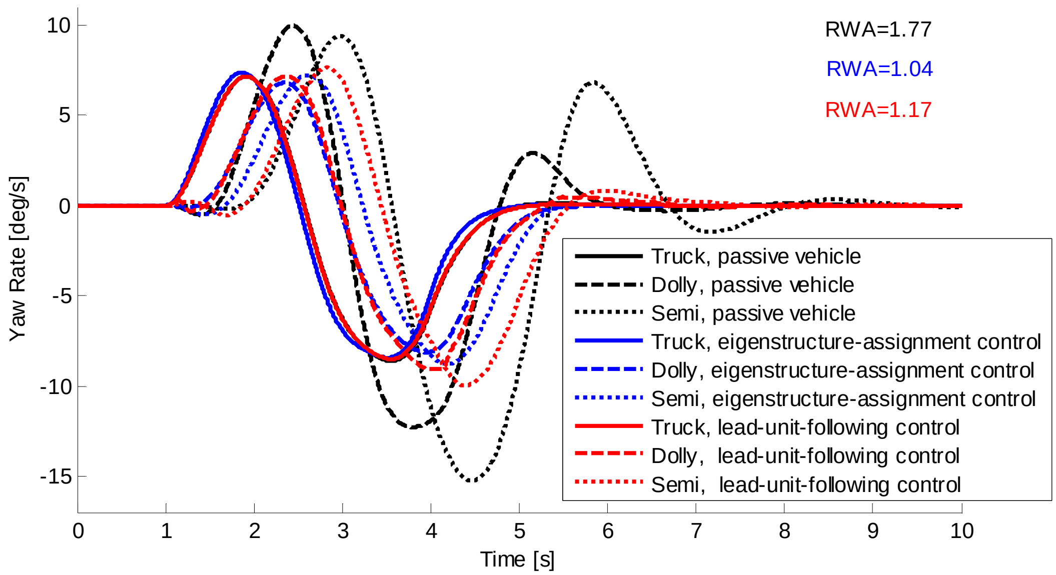


Figure 5.18. Effect of the two developed control schemes on the yaw rate rearward amplification of a truck-dolly-semitrailer in a sine with dwell maneuver

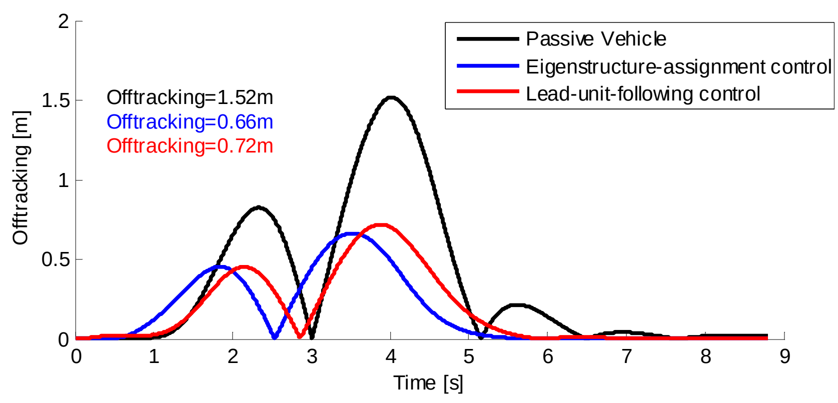


Figure 5.19 Effect of the two developed control schemes on the offtracking of a truck-dolly-semitrailer in a sine with dwell maneuver

Chapter 6

Concluding Remarks

This chapter starts with a description of the scientific contributions of the thesis and appended papers and ends with ideas for potential future work.

6.1 Scientific Contributions

In this thesis, the lateral performance of heavy vehicle combinations with different configuration features are investigated through accident analysis as well as simulation with linear and nonlinear models developed for this purpose. The cause of poor lateral performance of heavy vehicle combinations at high speeds, namely rearward amplification of motions, is identified to be due to the significant time delay between the driver steering and generation of tire lateral forces at the towed units. It is shown that by active steering of the towed units, the timing of generation of lateral forces can be adjusted and the rearward amplification of motions can be suppressed. Accordingly, two control schemes are developed and verified, both of which are significantly effective in reducing the yaw rate RWA and offtracking of the heavy vehicle combinations, specifically LCVs. The effectiveness and robustness of the first control scheme, *lead-unit-following control*, is verified by excessive simulations for various heavy vehicle combinations and driving conditions, as well as implementation on a truck-dolly-semitrailer test vehicle and evaluation on a test track. The second control scheme, *eigenstructure-assignment control*, is result of a short study on feasibility of eigenstructure assignment for the lateral performance control of heavy vehicle combinations and is not investigated as thoroughly as the first one. However, the attained encouraging results confirm the potential of this control scheme. By implementation of the developed controllers on heavy vehicle combinations, the identified relevant accidents can be prevented or mitigated. Furthermore, the illustrated safety benefits of the developed controllers can promote use of LCVs in traffic, which will result in reduction of traffic congestion problem as well as environmental and economic benefits. The outcomes of the research presented in this thesis are published as scientific papers which are included in this thesis; the scientific contributions of each paper are summarized here.

Paper A - Study of Heavy Truck Accidents with Focus on Maneuvers Causing Loss of Control

In this paper, a categorization of critical maneuvers of heavy vehicles which lead to loss of control (yaw instability and turn-over) are proposed and the most critical ones are identified. Furthermore correlations between heavy vehicles loss of control and accident type, vehicle combination type and different road characteristics are investigated. This paper also proposes the sine with dwell maneuver for evaluation of the lateral performance of heavy vehicles based on comparison of relevant existing test maneuvers according to the accident analysis results.

Paper B – A Generic Controller for Improving Lateral Performance of Heavy Vehicle Combinations

This paper starts with a study on the lateral performance of various heavy vehicle combinations. It is shown that the vehicle configuration type plays a more important role than the vehicle length in its lateral performance and there are LCVs that have better performance than some of the conventional combinations. Furthermore, in this paper a generic steering-based controller for improvement of the lateral performance of heavy vehicle combinations is developed (referred to as *lead-unit-following control* in this thesis). The controller effectiveness is verified on various heavy vehicle combinations, which show significant reduction in the yaw rate rearward amplification and offtracking, as well as moderate reduction in lateral acceleration rearward amplification, as a byproduct.

Paper C – Implementation of Active Steering on Longer Combination Vehicles for Enhanced Lateral performance

In this paper the implementation of the developed controller in Paper B on a truck-dolly-semitrailer and its real time performance is presented. The controller is verified in a series of single and double lane changes, which confirm its effectiveness and its compatibility with the existing ESC on the vehicle. The comparison of the controller performance with the ESC demonstrates that the first unit control, as in ESC, is not sufficient for suppression of amplified motions at the towed units

Paper D – Improving Lateral performance of Longer Combination Vehicles – An Approach Based on Eigenstructure Assignment

In this paper the possibility of application of eigenstructure assignment for the lateral performance improvement of LCVs is investigated. This paper presents a basic methodology for designing controllers for this purpose by partial eigenstructure assignment. The effectiveness of the controller is illustrated by simulation, which confirms the potential of this approach.

6.2 Recommendations for Future Work

The focus of the research presented in this thesis is on the lateral performance of heavy vehicle combinations, and although it is shown that the developed controllers could improve the roll stability moderately, the rollover issues of heavy vehicle combinations are not investigated thoroughly. The functionality of the developed controllers can be extended by adding the roll stability enhancement to the objectives. Additionally, the integration of active steering with braking should be investigated for possible further improvements in the lateral performance of heavy vehicle combinations.

In the developed controllers, all the axles of the towed units in a heavy vehicle combination are steered for enhanced lateral performance. The effect of reducing number of steered axles on the achievable improvement should be studied, and the most cost effective set of steerable axles should be identified.

The driver interaction with the developed controllers and the driver perception of the controllers' effectiveness is excluded from the conducted research and should be considered in related future works. Moreover, the input steering pattern of the drivers of heavy vehicle combinations, including steering amplitude and frequency, in normal lane changes and abrupt avoidance maneuvers should be investigated to be used as guidelines for development and verification of the proposed controllers or similar systems. In the followings, specific recommendations for each developed control scheme are provided.

The robustness analysis of the *lead-unit-following control* confirmed that the controller is quite robust and improves the lateral performance of the vehicle considerably even with incorrect vehicle parameters. However, the simulation results show that the controller performance deteriorates to some extent with incorrect tire parameters on low friction surfaces. Therefore, for increased robustness of the controller, an estimator for the tire cornering stiffness should be incorporated in the controller.

The presented *eigenstructure-assignment control* should be further developed by inclusion of the iteration process for selection of the desired eigenstructure in the optimization problem. Furthermore, the robustness and effectiveness of the controller under more diverse driving scenarios and for various heavy vehicle combinations should be studied.

Appendix A

Nomenclature

$a_{y,roll}$: Lateral acceleration threshold for rollover

$\mathbf{A}$: State matrix in the linear state space model

A_i : Amplitude of articulation angle subsequent oscillations

$\widetilde{\mathbf{A}}$: Auxiliary matrix for calculating the state matrix: $A = M^{-1}\tilde{A}$

$\mathbf{B}$: Input matrix in the linear state space model

$\widetilde{\mathbf{B}}$: Auxiliary matrix for calculating the input matrix: $B = M^{-1}\tilde{B}$

$C_{\varphi i}$: Roll damping of the *i*th unit

C_{ij} : Cornering stiffness of the *j*th axle of the *i*th unit

d_{fi} : Longitudinal distance of the front coupling of the *i*th unit, measured forward from its CG

d_{ri} : Longitudinal distance of the rear coupling of the *i*th unit, measured forward from its CG

DR_i : Yaw damping ratio of the *i*th articulation joint

f_d : Damped natural frequency

F_{ri} : Partial derivative of the tire lateral forces of the *i*th unit with respect to its yaw rate

F_{vi} : Partial derivative of the tire lateral forces of the *i*th unit with respect to its lateral velocity

F_{yLij} : Lateral force of the left tire of the *j*th axle of the *i*th unit, expressed in the tire coordinates

F_{yRij} : Lateral force of the right tire of the *j*th axle of the *i*th unit, expressed in the tire coordinates

F_{Ya} : Lateral force at the articulation joint

F_{YLij}: Lateral force of the left tire of the *j*th axle of the *i*th unit, expressed in the vehicle coordinates

F_{YRij} : Lateral force of the right tire of the *j*th axle of the *i*th unit, expressed in the vehicle coordinates

F_{xLij} : Longitudinal force of the left tire of the *j*th axle of the *i*th unit, expressed in the tire coordinates

F_{xRij} : Longitudinal force of the right tire of the *j*th axle of the *i*th unit, expressed in the tire coordinates

F_{XLij} : Longitudinal force of the left tire of the *j*th axle of the *i*th unit, expressed in the vehicle coordinates

F_{XRij} : Longitudinal force of the right tire of the *j*th axle of the *i*th unit, expressed in the vehicle coordinates

g : Gravity constant

G_{i1}^{j} : Transfer function for the controller feedforward part of the *i*th unit, corresponding to the *j*th design speed for the gain scheduling

$G_{ri\_\delta j}$: Transfer function between the yaw rate of the *i*th unit and steer angle of the *j*th unit

h_{si} : Height of CG of the *i*th unit above roll center

I_{zi} : Moment of inertia of the *i*th unit about its *z* axis

I_{xi} : Moment of inertia of the *i*th unit about its *x* axis

k_i : Controller feedback gain for steer angle of the *i*th unit

k_{us} : Understeer coefficient

$K_{\varphi i}$: Roll stiffness of the *i*th unit

l_e : Effective wheelbase

l_{ij} : Longitudinal distance of the *j*th axle of the *i*th unit, measured forward from its CG

m_i : Total mass of the *i*th unit

m_{si} : Sprung mass of the *i*th unit

$\mathbf{M}$: Mass matrix, used for calculation of the states and input matrices

M_{ri} : Partial derivative of generated yaw moment by tires of the *i*th unit with respect to its yaw rate

M_{vi} : Partial derivative of generated yaw moment by tires of the *i*th unit with respect to its lateral velocity

P_{di} : Pade approximation for $e^{-\tau_i s}$

P_f : Peak value of the motion variable of interest for the first unit

P_r : Peak value of the motion variable of interest for the rearmost unit

r_i : Yaw rate of the *i*th unit

r_{ides} : Desired yaw rate of the *i*th unit

u : Vehicle speed

u^j : The *j*th speed design for the gain scheduling

u_i : Longitudinal velocity of the *i*th unit

$\boldsymbol{U}$: Input vector in the linear state space model

v_i : Lateral velocity of the *i*th unit

$\boldsymbol{X}$: State vector

δ_1 : Steer angle of the front axle of the first unit

δ_i : Steer angle of the lumped axles of the *i*th unit, *i*>1

$\delta_{i,ff}$: Steer angle demand by controller feedforward part for the *i*th unit

$\delta_{i,fb}$: Steer angle demand by controller feedback part for the *i*th unit

θ_i : The *i*th articulation angle

λ : Eigenvalue

μ : Friction coefficient

ξ : Damping ratio

τ_i : Time lag of the *i*th unit with respect to the lead unit

φ_i : Roll angle of the *i*th unit

Appendix B

Vehicle Models

From modeling perspective, the heavy vehicle combinations studied in this thesis can be divided into three groups: two unit, three unit and four unit combinations. Here the developed models for a two unit combination are presented, which can be easily extended to combinations with more units.

In the first section, a nonlinear model for combined dynamics is derived using Newton's laws, which include longitudinal, lateral, yaw and roll degrees of freedom and incorporates Pacejka tire model [53]. The vertical load on each tire is calculated based on the roll angle and the lateral and longitudinal accelerations. The considered assumptions for the modeling are as follows: aerodynamic forces are neglected; vehicle units are considered as rigid masses and frame flexibility is neglected; the vehicle units have no bounce; the left and right wheels of each axle have equal steering angle; the roll stiffness and damping of the vehicle suspension systems are constant. In section B.2 the derived model is simplified to a nonlinear model for lateral dynamics. Finally in section B.3 a linear model is presented by considering some extra assumptions.

B.1 Nonlinear Model - Combined dynamics

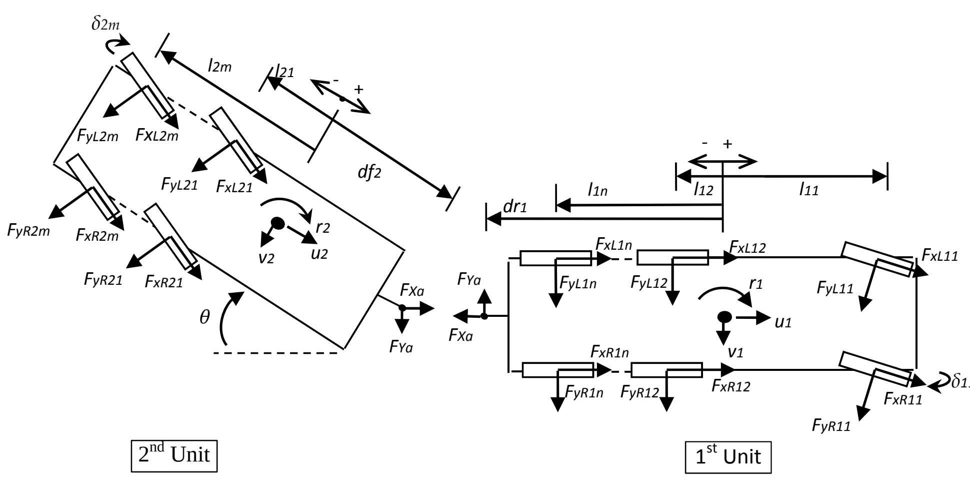


Figure B.1 Top view of a two unit combination

Figure B.1 shows the top view of a two unit combination, in which the tire forces are shown in the tire coordinates: F_{xRij} , F_{xLij}, F_{yRij} , F_{yLij}. However, in order to have orderly equations, the transferred tire forces to the vehicle coordinates, F_{XRij} , F_{XLij}, F_{YRij} , F_{YLij}, are used in the derivation of equations. Equation B.1, presents the rotation matrix for this transformation,

$$\begin{bmatrix} F_{XRij} & F_{XLij} \\ F_{YRij} & F_{YLij} \end{bmatrix} = \begin{bmatrix} \cos\delta_{ij} & -\sin\delta_{ij} \\ \sin\delta_{ij} & \cos\delta_{ij} \end{bmatrix} \begin{bmatrix} F_{xRij} & F_{xLij} \\ F_{yRij} & F_{yLij} \end{bmatrix}. \tag{B.1}$$

The equations of motion for the first unit are:

$$m_1(\dot{u}_1 - v_1\, r_1) \;= \textstyle\sum_{j=1}^{n} F_{XR1j} + F_{XL1j} - F_{Xa}, \tag{B.2}$$

$$m_1(\dot{v}_1 + u_1\, r_1) \;+\; m_{s1}h_{s1}\ddot{\varphi}_1 = \textstyle\sum_{j=1}^{n} F_{YR1j} + F_{YL1j} - F_{Ya}, \tag{B.3}$$

$$I_{z1}\dot{r}_1 = \textstyle\sum_{j=1}^{n}\big(F_{YR1j} + F_{YL1j}\big)l_{1j} + \big(F_{XL1j} - F_{XR1j}\big)t_1 - F_{Ya}d_{r1}, \tag{B.4}$$

$$(m_{s1}h_{s1}^2 + I_{x1})\ddot{\varphi}_1 + C_{\varphi 1}\dot{\varphi}_1 + (K_{\varphi 1} - \; m_{s1}h_{s1}g\;)\varphi_1 + m_{s1}h_{s1}(\dot{v}_1 + u_1\, r_1) = 0. \tag{B.5}$$

Similarly, the equations of motion for the second unit are:

$$m_2(\dot{u}_2 - v_2\, r_2) \;= \textstyle\sum_{j=1}^{m} F_{XR2j} + F_{XL2j} + F_{Xa}\cos\theta + F_{Ya}\sin\theta, \tag{B.6}$$

$$m_2(\dot{v}_2 + u_2\, r_2) + \; m_{s2}h_{s2}\ddot{\varphi}_2 = \textstyle\sum_{j=1}^{m} F_{YR2j} + F_{YL2j} + F_{Ya}\cos\theta - F_{Xa}\sin\theta, \tag{B.7}$$

$$I_{z2}\dot{r}_2 = \textstyle\sum_{j=1}^{m}\big(F_{YR2j} + F_{YL2j}\big)l_{1j} + \big(F_{XL2j} - F_{XR2j}\big)t_2 + (F_{Ya}\cos\theta - F_{Xa}\sin\theta)d_{f2}, \tag{B.8}$$

$$(m_{s2}h_{s2}^2 + I_{x2})\ddot{\varphi}_2 + C_{\varphi 2}\dot{\varphi}_2 + (K_{\varphi 2} - \; m_{s2}h_{s2}g)\varphi_2 + m_{s2}h_{s2}(\dot{v}_2 + u_2\, r_2) \;= 0. \tag{B.9}$$

Combining equations B.2, B.3, B.6 and B.2, B.3, B.7 and B.3, B.4 and B.7, B.8 together, results in:

$$\begin{aligned} m_1(\dot{u}_1 - v_1\, r_1)\cos\theta + m_1(\dot{v}_1 + u_1\, r_1)\sin\theta + m_{s1}h_{s1}\ddot{\varphi}_1\sin\theta + \\ + m_2(\dot{u}_2 - v_2\, r_2) = F_X, \end{aligned} \tag{B.10}$$

$$\begin{aligned} m_1(\dot{v}_1 + u_1\, r_1)\cos\theta + m_{s1}h_{s1}\ddot{\varphi}_1\cos\theta - m_1(\dot{u}_1 - v_1\, r_1)\sin\theta + \\ + m_2(\dot{v}_2 + u_2\, r_2) + \; m_{s2}h_{s2}\ddot{\varphi}_2 = F_Y, \end{aligned} \tag{B.11}$$

$$I_{z1}\dot{r}_1 - m_1 d_{r1}(\dot{v}_1 + u_1\, r_1) - m_{s1}h_{s1}d_{r1}\ddot{\varphi}_1 = M_{Z1}, \tag{B.12}$$

$$I_{z2}\dot{r}_2 - m_2 d_{f2}(\dot{v}_2 + u_2\, r_2) - m_{s2}h_{s2}d_{f2}\ddot{\varphi}_2 = M_{Z2}, \tag{B.13}$$

where

$$F_X = \textstyle\sum_{j=1}^{n}\big(F_{XR1j} + F_{XL1j}\big)\cos\theta + \big(F_{YR1j} + F_{YL1j}\big)\sin\theta + \sum_{j=1}^{m}\big(F_{XR2j} + F_{XL2j}\big), \tag{B.14}$$

$$F_Y = \textstyle\sum_{j=1}^{n}\big(F_{YR1j} + F_{YL1j}\big)\cos\theta - \big(F_{XR1j} + F_{XL1j}\big)\sin\theta + \sum_{j=1}^{m}\big(F_{YR2j} + F_{YL2j}\big), \tag{B.15}$$

$$M_{Z1} = \textstyle\sum_{j=1}^{n}\big(F_{YR1j} + F_{YL1j}\big)(l_{1j} - d_{r1}) + \big(F_{XL1j} - F_{XR1j}\big)t_1, \tag{B.16}$$

$$M_{Z2} = \textstyle\sum_{j=2}^{m}\big(F_{YR2j} + F_{YL2j}\big)(l_{2j} - d_{f2}) + \big(F_{XL2j} - F_{XR2j}\big)t_2. \tag{B.17}$$

The velocity of the articulation point can be expressed based on either the first unit parameters or the second unit parameters, which leads to the equations for the kinematical constraints as

$$(v_1 + d_{r1}r_1)\sin\theta + u_1\;\cos\theta = u_2, \tag{B.18}$$

$$(v_1 + d_{r1}r_1)\cos\theta - u_1\;\sin\theta = v_2 + d_{f2}r_2. \tag{B.19}$$

Differentiating equations B.18 and B.19 results in

$$(\dot{v}_1 + d_{r1}\dot{r}_1)\sin\theta + (v_1 + d_{r1}r_1)(r_2 - r_1)\cos\theta + \dot{u}_1\ \cos\theta - u_1\ (r_2 - r_1)\sin\theta = \dot{u}_2, \quad \text{(B.20)}$$

$$(\dot{v}_1 + d_{r1}\dot{r}_1)\cos\theta - (v_1 + d_{r1}r_1)(r_2 - r_1)\sin\theta - \dot{u}_1\ \sin\theta_1 - \\ -u_1\ (r_2 - r_1)\cos\theta = \dot{v}_2 + d_{f2}\dot{r}_2. \quad \text{(B.21)}$$

Equations B.5, B.9, B.10-B.13 and B.20-B.21 constitute final set of equations of motion.

B.2 Nonlinear Model - Lateral dynamics

Here a simpler nonlinear model for study of the lateral dynamics is derived by assuming constant and equal longitudinal velocity for both units and negligible longitudinal forces; the combined dynamics model is simplified accordingly.

The equations of motion for the first unit are simplified to:

$$m_1(\dot{v}_1 + u\,r_1)\ +\ m_{s1}h_{s1}\ddot{\varphi}_1 = \textstyle\sum_{j=1}^{n} F_{YR1j} + F_{YL1j} - F_{Ya}, \quad \text{(B.22)}$$

$$I_{z1}\dot{r}_1 = \textstyle\sum_{j=1}^{n}\left(F_{YR1j} + F_{YL1j}\right)l_{1j} - F_{Ya}d_{r1}, \quad \text{(B.23)}$$

$$(m_{s1}h_{s1}^2 + I_{x1})\ddot{\varphi}_1 + C_{\varphi1}\dot{\varphi}_1 + (K_{\varphi1} -\ m_{s1}h_{s1}g\ )\varphi_1 + m_{s1}h_{s1}(\dot{v}_1 + u\,r_1) = 0. \quad \text{(B.24)}$$

Similarly, the equations of motion for the second unit are simplified to:

$$m_2(\dot{v}_2 + u\,r_2) +\ m_{s2}h_{s2}\ddot{\varphi}_2 = \textstyle\sum_{j=1}^{m} F_{YR2j} + F_{YL2j} + F_{Ya}\cos\theta, \quad \text{(B.25)}$$

$$I_{z2}\dot{r}_2 = \textstyle\sum_{j=1}^{m}\left(F_{YR2j} + F_{YL2j}\right)l_{1j} + F_{Ya}d_{f2}\cos\theta, \quad \text{(B.26)}$$

$$(m_{s2}h_{s2}^2 + I_{x2})\ddot{\varphi}_2 + C_{\varphi2}\dot{\varphi}_2 + (K_{\varphi2} -\ m_{s2}h_{s2}g)\varphi_2 + m_{s2}h_{s2}(\dot{v}_2 + u\,r_2)\ = 0. \quad \text{(B.27)}$$

Combining equations 21, 24 and 21, 22 and 24, 25 together, results in

$$m_1(\dot{v}_1 + u\,r_1)\cos\theta + m_{s1}h_{s1}\ddot{\varphi}_1\cos\theta + m_2(\dot{v}_2 + u\,r_2) +\ m_{s2}h_{s2}\ddot{\varphi}_2 = F_Y, \quad \text{(B.28)}$$

$$I_{z1}\dot{r}_1 - m_1 d_{r1}(\dot{v}_1 + u\,r_1) - m_{s1}h_{s1}d_{r1}\ddot{\varphi}_1 = M_{Z1}, \quad \text{(B.29)}$$

$$I_{z2}\dot{r}_2 - m_2 d_{f2}(\dot{v}_2 + u\,r_2) - m_{s2}h_{s2}d_{f2}\ddot{\varphi}_2 = M_{Z2}, \quad \text{(B.30)}$$

where

$$F_Y = \textstyle\sum_{j=1}^{n}\left(F_{YR1j} + F_{YL1j}\right)\cos\theta + \sum_{j=1}^{m}\left(F_{YR2j} + F_{YL2j}\right), \quad \text{(B.31)}$$

$$M_{Z1} = \textstyle\sum_{j=1}^{n}\left(F_{YR1j} + F_{YL1j}\right)(l_{1j} - d_{r1}), \quad \text{(B.32)}$$

$$M_{Z2} = \textstyle\sum_{j=2}^{m}\left(F_{YR2j} + F_{YL2j}\right)(l_{2j} - d_{f2}). \quad \text{(B.33)}$$

In this model, only the kinematic constraint in the lateral direction, B.21, is used; it is modified by assuming constant and equal longitudinal speed for both units, as in

$$(\dot{v}_1 + d_{r1}\dot{r}_1)\cos\theta - (v_1 + d_{r1}r_1)(r_2 - r_1)\sin\theta - u\ (r_2 - r_1)\cos\theta = \dot{v}_2 + d_{f2}\dot{r}_2. \quad \text{(B.34)}$$

Equations B.24, B.27, B.28-B.30 and B.34 constitute final set of equations of motion.

B.3 Linear Model

The linear model of a two-unit combination is derived by linearization of the nonlinear model for lateral dynamics about the constant speed straight-line driving condition. The standard assumptions for derivation of a linear vehicle model are used, namely small steering and articulation angles, linear tires and negligible roll dynamics and load transfer.

Assuming linear tires, the lateral forces can be written as

$$F_{Yij} = C_{ij}\left(\delta_{ij} - \frac{v_i + l_{ij} r_i}{u}\right). \tag{B.35}$$

By defining the following terms:

$$F_{vi} = -\sum_j \frac{C_{ij}}{u} \quad , \quad F_{ri} = -\sum_j \frac{C_{ij} l_{ij}}{u} \quad , \; M_{vi} = -\sum_j \frac{C_{ij} l_{ij}}{u} \quad , \quad M_{ri} = -\sum_j \frac{C_{ij} l_{ij}^2}{u} \; , \tag{B.36}$$

the nonlinear equations of motion (equations B.22-B.27) can be linearized to

$$m_1(\dot{v}_1 + u\, r_1) = F_{v1} v_1 + F_{r1}\, r_1 + C_{11}\delta_1 - F_{Ya}, \tag{B.37}$$

$$I_{z1}\dot{r}_1 = M_{v1} v_1 + M_{r1}\, r_1 + C_{11} l_{11}\delta_1 - F_{Ya} d_{r1}, \tag{B.38}$$

$$m_2(\dot{v}_2 + u\, r_2) = F_{v2} v_2 + F_{r2}\, r_2 + \sum_j C_{2j}\, \delta_2 + F_{Ya}, \tag{B.39}$$

$$I_{z2}\dot{r}_2 = M_{v2} v_2 + M_{r2}\, r_2 + \sum_j C_{2j} l_{\,2j}\, \delta_2 + F_{Ya} d_{f2}. \tag{B.40}$$

where δ_1 denotes the steering angle of the front axle of the first unit; rest of the axles of the first unit are assumed to be non-steerable. δ_2 denotes the steering angle of the axles of the second unit, which is assumed to be the same for all axles.

Linearizing and differentiating the equation for kinematical constraint in the lateral direction, B.19, results in

$$\dot{v}_1 + d_{r1}\dot{r}_1 - u\,(r_2 - r_1) = \dot{v}_2 + d_{f2}\dot{r}_2. \tag{B.41}$$

Finally, eliminating the joint force and putting the equations together in state space form, give the linear model of a two unit combination:

$$X = [v_1 \quad r_1 \quad v_2 \quad r_2]^{\mathrm{T}} \quad , \quad U = [\delta_1 \quad \delta_2]^{\mathrm{T}}, \tag{B.42}$$

$$\dot{X} = M^{-1}\tilde{A}X + M^{-1}\tilde{B}U = AX + BU, \tag{B.43}$$

$$M = \begin{bmatrix} d_{r1}m_1 & -I_{z1} & 0 & 0 \\ 0 & 0 & d_{f2}m_2 & -I_{z2} \\ m_1 & 0 & m_2 & 0 \\ -1 & -d_{r1} & 1 & d_{f2} \end{bmatrix}, \tag{B.44}$$

$$\tilde{A} = \begin{bmatrix} F_{v1}d_{r1} - M_{v1} & F_{r1}d_{r1} - M_{r1} - d_{r1}m_1 u & 0 & 0 \\ 0 & 0 & F_{v2}d_{f2} - M_{v2} & F_{r2}d_{f2} - M_{r2} - d_{f2}m_2 u \\ F_{v1} & F_{r1} - m_1 u & F_{v2} & F_{r2} - m_2 u \\ 0 & u & 0 & -u \end{bmatrix}, \tag{B.45}$$

$$\tilde{B} = \begin{bmatrix} C_{11}(d_{r1} - l_{11}) & 0 & C_{11} & 0 \\ 0 & \sum_j C_{2j}\left(d_{f2} - l_{2j}\right) & \sum_j C_{2j} & 0 \end{bmatrix}^{\mathrm{T}}. \tag{B.46}$$

Appendix C

ESC Model

The modeled Electronic Stability Control (ESC) system applies differential braking on the lead unit so that its yaw rate follows a desired value within a certain bound. If the truck oversteers, that is, its yaw rate is higher than the desired value, the ESC applies a constant braking force to the outer front wheel and if the truck understeers, that is, its yaw rate is lower than the desired value, the ESC applies a constant braking force to the inner rear wheel.

The desired yaw rate, $r_{1_{des}}$, for the lead unit is defined based on the driver steering input, δ_1, vehicle speed, u, and vehicle parameters, as in

$$r_{1_{des}} = r_{ss}\frac{\delta_1(s)}{1+\tau s}, \tag{C.1}$$

where

$$r_{ss} = \frac{u}{l_e + k_{us}\frac{u^2}{g}}, \tag{C.2}$$

and l_e, k_{us} are the effective wheelbase and understeer coefficient of the truck [54], respectively. The first order time lag is to prevent oscillation [55], and its time constant, τ, is calculated based on the phase lag between the driver steering input and the truck yaw rate. The desired yaw rate is bounded based on the available friction, μ, and the lateral acceleration threshold for rollover, $a_{y,roll}$ [56], as in

$$\left|r_{1_{des}}\right| \leq \frac{1}{u}\min(0.8\,\mu g, a_{y,roll}). \tag{C.3}$$

The ESC intervenes if the difference between the actual yaw rate of the lead unit and the desired yaw rate is larger than 1 deg/s; the braking force is released when this difference goes back to below 0.5 deg/s. If the ESC intervenes, it also applies braking on the towed units, so that each unit would decelerate at a rate of 1 m/s^2. The choice of the deceleration magnitude is based on the study presented in [52]. The modeled ESC has a simple Antilock Braking System (ABS) which prevents the wheel locking by limiting the applied braking force based on the available friction.

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