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On Traffic Situation Predictions for Automated Driving of Long Vehicle Combinations

by

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To my wonderful girls Elin and Linnea

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Abstract

The introduction of longer vehicle combinations for road transports than are currently allowed is an important viable option for achieving the environmental goals on transported goods in Sweden and Europe by the year 2030. This thesis addresses how driver assistance functionality for high-speed manoeuvring can be designed and realized for prospective long vehicle combinations. The main focus is the derivation and usage of traffic situation predictions in order to provide driver support functionalities with a high driver acceptance. The traffic situation predictions are of a tactical character and include a time horizon of up to 10 s.

Data collection of manual and automated driving with an A-double combination was carried out in a moving-base driving simulator. The driving scenario was comprised of a relatively curvy and hilly single-lane Swedish county road (180). The driving trajectories were analysed and complemented with results from optimization. Based on observations of utilized accelerations it was proposed that the combined steering and braking should prioritize a smooth and comfortable driving experience.

It was hypothesized that high driver acceptance of driver assistance functionality including automated steering and propulsion/braking, can be realized by utilizing driver models inspired by human cognition as an integrated part in the generation of traffic situation predictions. A longitudinal and lateral driver model based on optic information was proposed for lane-change manoeuvring. The driver model was implemented in a real-time framework for automated driving of an A-double combination on a multiple lane one-way road. Simulations showed that the framework gave reasonable results for maintain lane and lane change manoeuvres at constant and varying longitudinal velocities.

Keywords: long vehicle combination, vehicle dynamics, active safety, driver behaviour, heavy trucks, steering, braking, prediction, automated driving, advanced driver assistance systems

List of Included Papers

Paper 1: P. Nilsson, L. Laine, and B. Jacobson, "Performance Characteristics for Automated Driving of Long Heavy Vehicle Combinations Evaluated in Motion Simulator," in *Intelligent Vehicles Symposium Proceedings, 8-11 June 2014. Dearborn, MI*, 2014, pp. 362-369.

Paper 2: J. Sandin and P. Nilsson, "Drivers' assessment of driving a 32 meter A-double with and without full automation in a moving simulator base simulator," in *13th International Heavy Vehicle Transport Technology Symposium, San Luis, Argentina*, 2014

Paper 3: P. Nilsson, L. Laine, O. Benderius, and B. Jacobson, "A Driver Model Using Optic Information for Longitudinal and Lateral Control of a Long Vehicle Combination," in *IEEE 17th International Conference on Intelligent Transportation Systems (ITSC), October 8-11, 2014. Qingdao, China*, 2014, pp. 1456-1461.

Paper 4: P. Nilsson, L. Laine, B. Jacobson and N. van Duijkeren, "Driver Model Based Automated Driving of Long Vehicle Combinations in Emulated Highway Traffic," *Submitted to IEEE 18th International Conference on Intelligent Transportation Systems (ITSC), September 15-18, 2015. Las Palmas, Spain*, 2015.

The author of this thesis had the main responsibility for the implementation of models, carrying out numerical simulations, analysis and writing of Papers 1, 3 and 4. For Paper 3, the author collaborated with Benderius in the parametrization of the driver model and writing. For Paper 2, the author collaborated with Sandin in extracting the data set and the analysis. The writing was mainly done by Sandin. For Paper 4, the author collaborated with Duijkeren in the the implementation of models.

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Peter Nilsson
Göteborg, March 2015

Contents

Chapter 1

Introduction

This chapter presents the background, motivation and objectives of the thesis. It also gives the limitations of scope and outline of the thesis.

1.1 Background

Sweden has a long history of long vehicle combination usage within road transports in order to improve productivity. Further development of how future modular long vehicle combinations for road transport can be designed and controlled, with respect to both energy efficiency and safety, are currently ongoing within the transport community in Sweden and Europe. It is foreseen that longer combinations than those agreed upon in EU directive 96/53, European modular system, can be one way to meet upcoming environmental goals and emission legislations on transported goods. The predicted combinations typically range between 27-34 m in length and have at least two articulated joints. In a pilot project [1] conducted in Sweden for long vehicle combination (LVC) utilization, the productivity of a 90 t vehicle combination of 30 m in length, was seen to improve by approximately 20 percent when compared to a conventional Nordic 60 t combination of 25.25 m in length.

1.2 Motivation

The motivation for this project is to investigate how driver assistance functionality for high-speed manoeuvring, ranging from 0-90, km/h could be designed and realized for LVCs. It is anticipated that traffic situation predictions (TSPs), including vehicle models that can represent the planar vehicle dynamics of LVCs, will play an important part in the development of automated driving functionalities.

1.3 Objective

The overall objective of this work is to increase the knowledge of TSPs in order to provide driver support functionalities including automated steering and propulsion/braking for high-speed manoeuvring of LVCs. Firstly, due to amplified planar vehicle dynamics, the TSP generation for LVCs incorporates many dynamic motion constraints. When considering the usage of vehicle models to express the motion constraints, the vehicle models need to be quantified with respect to complexity level in order to secure reliable and computationally efficient TSP algorithms. Secondly, the driving behaviour of the LVC drivers are coupled to the TSP generation and is important for the overall driver acceptance of the envisioned support functionalities.

1.4 Limitations

The research presented in this thesis assumes the usage of an envisioned reference architecture for vehicle motion functionality developed at Volvo GTT, illustrated in Figure 1.1. The architecture is partitioned into a hierarchical structure to separate motion functionality into long term, mid term, and short term planning, execution, and tracking. The functionality domain component vehicle motion management (VMM), includes a time horizon of up to 1 s and has a reactive and coordinative character rather than predictive and arbitrative. The core functionality of VMM is vehicle stability. The traffic situation management (TSM) includes a time horizon of up to 10 s and the prediction has a tactical character rather than reactive. The functionality domains of strategical character have a time horizon larger than 10 s and are omitted from Figure 1.1. The targeted traffic situation predictions are within the functionality domain TSM.

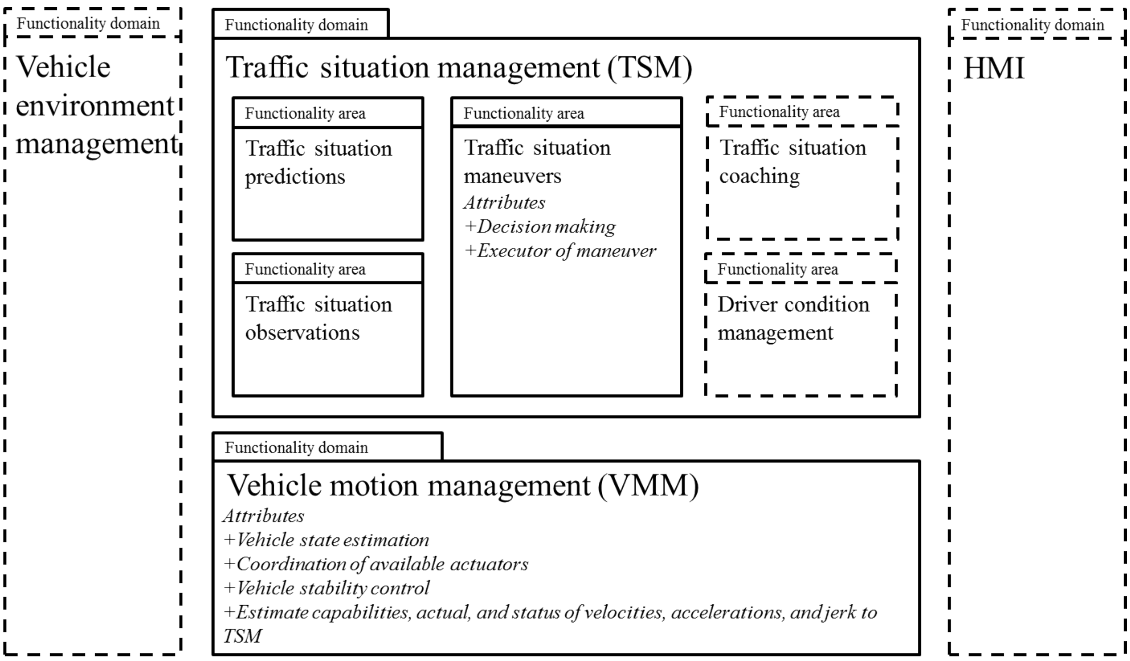

Figure 1.1: Part of the envisioned reference architecture for vehicle motion functionality developed at Volvo GTT for year 2020 and beyond.

The primary application of the research is LVCs based on the modular concept however, only the A-double combination has been considered in this work. The developed functionality can be simplified to a tractor semi-trailer combination which is the most

common vehicle combination within Europe.
Sensor fusion of vehicle environment sensors, as well as road conditions including low friction, are excluded from this work.

1.5 Thesis Outline

The thesis is structured as follows. Chapter 2 introduces basic knowledge about LVCs. In Chapter 3 typical road usage and relevant driving scenarios for LVCs are discussed. In Chapter 4 vehicle planar dynamics of LVCs are introduced and the vehicle models are derived and analysed. In Chapter 5 envisioned driver assistance systems for LVCs are discussed and the main conclusions from Papers 1-4 are presented. Finally, in Chapter 6 the main methodology is discussed and future priorities are proposed. Notations used follow ISO 8855 [2], and units are SI unless otherwise stated.

Chapter 2

Long vehicle combinations

Long vehicle combinations refer to modular road vehicles that are longer and heavier than the currently permitted dimensions in Sweden. LVCs typically include at least two articulated joints and their length typically varies between 27-34 m. The main motivating factors for LVC utilization are increased transport productivity and reduced environmental impact. However, road infrastructure and traffic safety are also important considerations in the potential introduction of LVCs.

2.1 Background in Sweden and Europe

The following background is based on Aurell and Wadman [3]. Prior to 1968, there were no limits on the total length of truck combinations in Sweden and quite a few were 30 m and longer. The most common vehicle combination was a truck with a full trailer and the normal length was 24 m. During a transition period between 1968 and 1972, the maximum authorized total length was set to 24 m. The first European directive on weights and dimensions for articulated vehicles was established in 1985, directive 85/3 EEC. This limited the maximum length of vehicles in international traffic to 18 m, but did not regulate the domestic road transport solutions. In 1989 and again in 1991, amendments to directive 85/3 EEC were added that clarified the maximum allowed vehicle load length. In 1996, an updated version of the existing directive for the weights and dimensions was approved, directive 96/53/EC [4], as is described in next section.

2.2 Current situation in Europe

For all heavy goods vehicles within the European Union, the maximum authorized weight and length dimensions are regulated by the Council directive 96/53/EC. The directive states that the maximum lengths of tractor semi-trailer combinations and rigid truck-trailer combinations are 16.5 m and 18.75 m, respectively. The maximum weight is restricted to 40 t with the exception of domestic transports including 40-foot ISO containers in a combined transport operation, where the maximum weight is 44 t. However, the directive gives each member country the possibility to use longer and heavier vehicle combinations within its territory provided that the combinations are based on the so-called

European modular system (EMS) [3], or do not significantly affect international competition in the transport sector. In Sweden, longer and heavier vehicle combinations based on the EMS concept were introduced in 1997. There are three types of allowed combinations, each carrying one short module and one long module, as illustrated in Figure 2.1. This gives a total vehicle length of 25.25 m and an allowed maximum weight of 60 t. Beyond Sweden, longer vehicle combinations based on EMS are currently utilized in Finland and experiments in other countries such as the Netherlands and Denmark have provided examples of good practice.

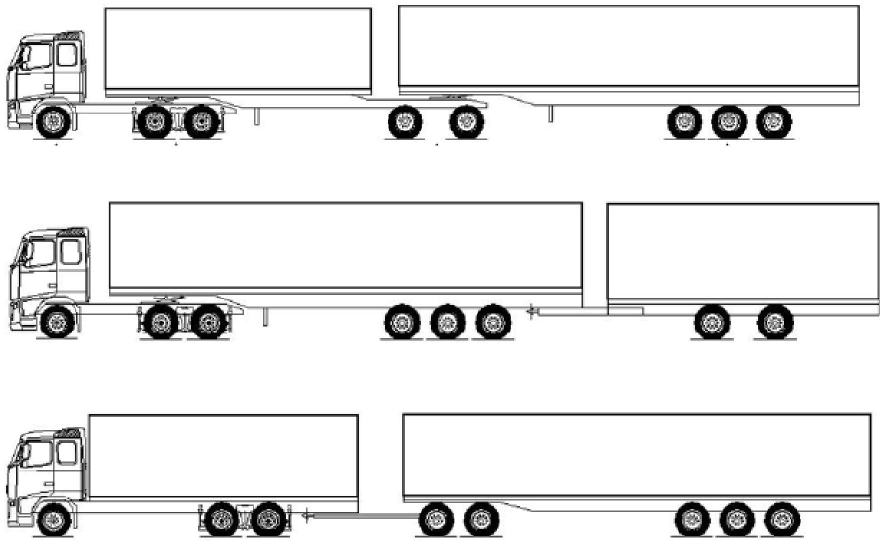

Figure 2.1: Illustration of vehicle combinations longer than 18.75 m currently allowed in Sweden: B-double (top), tractor semi-trailer and centre-axle trailer (middle), and truck dolly and semi-trailer (bottom). The figure is based on [3].

In Sweden, there is a growing interest within society and the transport community to study heavy goods vehicles that exhibit a reduced environmental impact and have the potential to be more productive than existing vehicles. Based on this criteria the most promising vehicles are the modular LVCs, exemplified in Figure 2.2. Asides from the pilot studies of LVC usage mentioned in Section 1.1, initiatives for a general introduction of LVCs in traffic are being carried out [5, 6]. In addition, investigations on performance based standards (PBSs) suitable for Swedish conditions are ongoing [7].

The main concerns regarding an introduction of LVCs relate to the road infrastructure and traffic safety. An introduction of LVCs in Sweden would most likely require additions to the existing classification of road types. There are also several studies concerning the traffic safety impact of longer and heavier vehicles such as LVCs [8, 9, 10]. However it is not clear if and how traffic safety would be affected by an introduction of LVCs.

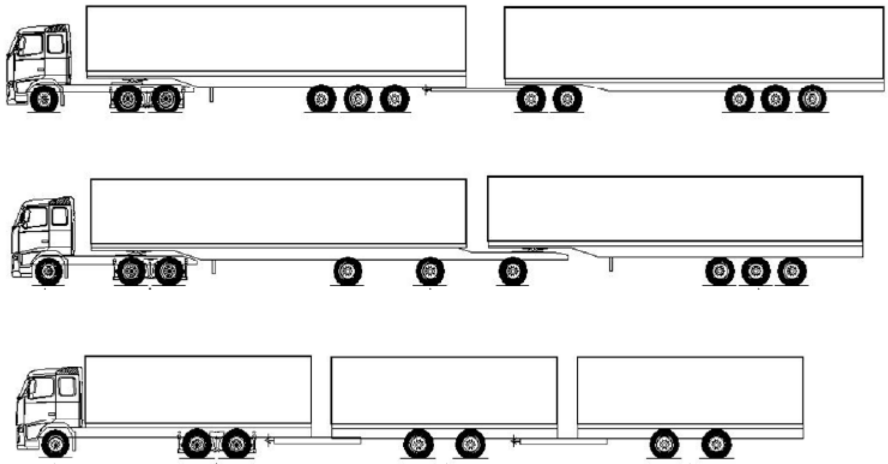

Figure 2.2: Illustration of LVCs based on the European modular system (EMS). A-double (top), B-double (middle) and double centre-axle trailer (bottom). The figure is based on [3].

2.3 Current situation outside Europe

Varying utilizations of LVCs exist in Australia, Brazil, Canada, Mexico, New Zeeland and USA [11]. Common for all countries is that LVCs are not allowed to operate on all types of roads. The most common vehicle combinations are A-double and B-double, but C-double and truck-full trailer are also present.

In Australia, PBSs for heavier and longer vehicles were introduced in 2007. PBSs provide regulations concerning the vehicle performance instead of utilizing a common legislation for the length and weight dimensions. The intention of PBS is to improve road traffic safety and increase transport productivity. The maximum length and weight dimensions in Australia, for example, are 53.5 m and 125.2 t respectively. The legislation for LVCs in the USA varies between the different states where the state of Colorado allows the highest maximum vehicle length of 35.2 m.

Chapter 3

Driving scenarios for LVCs

3.1 Typical road type usage

In Sweden, the public road network is divided into 3 weight classes: BK1, BK2 and BK3. The classes restrict maximum gross combinations of weight and static load per axle and axle groups and minimize the distance between axle groups. BK1 roads, on which the highest weights are permitted, cover about 95 % of the public road network. Today's heavy goods vehicles are allowed on all BK1 roads. However, LVCs are not expected to be driven on all existing BK1 roads and a further specification of the road weight classes in some form is envisioned and proposed [6] as part of a general introduction of LVCs. The intention is to allow modular LVCs mainly on roads with the highest weight class. Before driving on other roads they can be decoupled into shorter conventional combinations. For example, the A-double combination, which consists of a tractor unit, semi-trailer, dolly-converter and a second semi-trailer, can be converted to a standard tractor semi-trailer when approaching city areas.
Besides the weight restrictions, other road infrastructure limitations e.g. oncoming traffic and types of intersections, may be taken into consideration when determining the road usage in a general LVC introduction. One traffic situation that has been specifically studied is overtaking situations in oncoming traffic [8]. Even though no significant increased accident risk associated with overtaking was found, one-way multiple lane roads are considered in this thesis to be the primary application for LVCs.

3.2 Typical driving scenarios

Identifying typical driving scenarios that commonly occur or are safety critical is vital to the development of driver support systems for LVCs, including the automation of steering and braking/propulsion. Based on the assumed typical road usage of one-way multiple lane roads, the majority of the driving is expected to consist of maintain lane manoeuvring. Naturally, other driving situations or manoeuvres such as lane changes, roadway departures and overtaking also occur. Investigation of existing European heavy truck accidents shows that human error is a significant factor in many accidents [12]. When the truck contributes to the accident, the most common cause is limited visibility due to blind spots. A typical blind spot accident is a lane change to right in right-hand traffic [13], exemplified in Figure 3.1 (top). When increasing the vehicle combination length it is hypothesized that a corresponding accident for the combination rear end, both left and right hand side, will occur more frequently, as exemplified in Figure 3.1 (bottom). However, this is only a hypothesis and cannot be proven by accident statistics since LVCs are not yet in general use.

Based on the discussion above, the development of driver support functionality included in this thesis focuses on maintain lane and lane change manoeuvring. Maintain lane manoeuvring is investigated in Papers 1 and 2 and lane change manoeuvres are discussed in Papers 3 and 4.

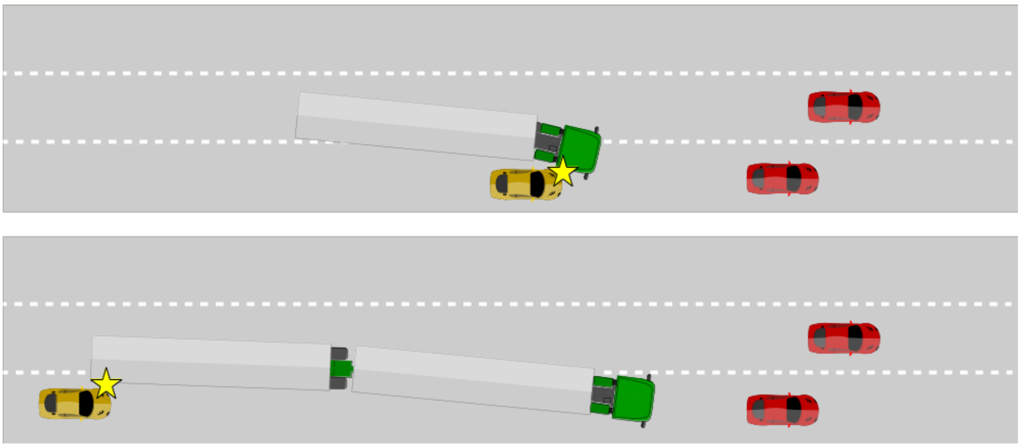

Figure 3.1: Top panel: Typical blind spot accident involving an existing vehicle combination, together with a lane change to the right in right hand traffic. Bottom panel: Hypothesized lane change accident including LVCs.

Chapter 4

Vehicle planar dynamics of LVCs

Vehicle planar dynamics refer to the vehicle motion in the longitudinal, lateral and yaw directions. In this chapter vehicle models for planar motion analysis of LVCs are introduced and analysed.

4.1 Vehicle modelling

A vehicle model is here defined as the system of equations needed to describe the dynamics of a vehicle during acceleration, braking and/or cornering manoeuvring. The vehicle model is often expressed as a system of non-linear differential-algebraic equations (DAEs). The DAEs are based on kinematic and inertial properties of the vehicle as well as constitutive relations between motion and force quantities. One example of a commonly used constitutive relation is a tyre model where the tyre force is related to the tyre velocity.

In order to satisfy the requirements stated in vehicle models for usage in the generation of traffic situation predictions, rigid body two-track and one-track models are considered. In a two-track model, each wheel is modelled separately and can accordingly exhibit varying cornering stiffness. Moreover, yaw moment produced by non-symmetrical wheel torque interventions are inherently represented. In a one-track model (also known as a single-track model or bicycle model), the effects of all tyres on an axle are combined into one virtual tyre. The concept of an equivalent wheel-base is also introduced and used, which means that groups of axles (e.g a boogie or tridem) are collapsed into one axle. The included motion degrees of freedom for both model types are the longitudinal, lateral, and yaw motion of each vehicle unit. The general form of both the two-track and the one-track models can be simplified by introducing assumptions related to the vehicle driving conditions. A common simplification is linearisation where the assumptions are: constant longitudinal velocity, small steering and articulation angles, small side slip angles, and linear tyre constitution. Examples of linearised two-track and one-track models can be found in [14] and [15], respectively. In addition to linearisation, manoeuvre dependent assumptions are possible and commonly utilized. Examples of such assumptions are steady-state conditions and low speed manoeuvring, which are mathematically represented by neglecting terms of the type inertia multiplied by acceleration.
There are several methods for generating the dynamic relations for planar motion [16].

All methods generate equivalent descriptions but different forms of the equations may be better suited for intended analysis. Two main methods, the *Newtonian formalism* and the *Lagrangian formalism*, are considered here. In Newtonian formalism [17], the second law is used to express the equations of motions for all individual vehicle units. In Newtonian mechanics, external forces, e.g. tyre forces, are acting on the studied system. When dividing the studied system into sub-systems, e.g. vehicle units, internal forces between the sub-systems are added. This means that the constraints between the vehicle units need to be defined for both the motion and force variables. The couplings between the vehicle units can be assumed to be ideal constraints, but can also be modelled as springs and dampers, in which case they are instead modelled with constitutive equations.
In Lagrangian formalism [18], the first step is to define a set of coordinates that describes the system's state uniquely with respect to an inertial frame. Any set of coordinates having this property is called a set of generalized coordinates. Secondly, the kinetic and potential energies are written in terms of these coordinates and the force components of the forces acting on the system are computed along these coordinates. These forces are referred to as generalized forces. Finally, the substitution of these quantities in Lagrange's equations results in the final formulation. The main benefit of using the Lagrange formalism is that the coupling forces between the vehicle units are inherently represented and the number of equations are correspondingly fewer. The approach also has the advantage of requiring only the kinetic and potential energies of the system to be formulated and hence tends to be less prone to error than summing together the coupling forces. On the other hand, the Newtonian approach is more adapted for modularization, which can be efficiently used in the modelling of LVCs, especially when combined with tools for symbolic computation [19].

4.2 Tyre mechanics

The vehicle motion accomplished during acceleration, braking and/or cornering, is a response to imposed forces. In order to understand the vehicle dynamics it is therefore important to grasp how and why these forces occur. The most dominant forces in vehicle dynamics behaviour are produced by the relative motion of the tyre to the ground and are denoted as tyre forces in the road plane. The characteristics of the tyres produces forces are mainly affected by the tyre material, construction, and shape, tyre vertical load, tyre air pressure, and ground/road condition [20].
A lateral tyre force, i.e. a force directed perpendicular to the wheel rotation plane, is generated when the travelling direction of the wheel hub is out of line with the wheel rotational plane. This situation can occur when the vehicle has lateral and/or yaw motion or in straight line driving if the wheel has a steering angle. The wheel is then said to have a side slip where the angle between the wheel travelling direction and the wheel rotational plane is called the side-slip angle. A longitudinal tyre force, i.e. a force directed tangential to the wheel rotation plane, is generated by producing a difference between the tyre circumferential speed and its translational speed. The relative motion of the tyre and road surfaces is measured as a longitudinal tyre slip, which is defined as the ratio of the relative speed between the tyre and the ground and a reference speed. The reference speed can be

the translational speed or the circumferential speed of the tyre, or a combination of both. The force/slip relation of a typical non-driven truck front tyre is depicted in Figure 4.1. Due to its importance in vehicle engineering, the subject of tyre mechanics has been extensively studied. There exists a vast number of tyre models [21], both empirical and theoretical, that are commonly used within vehicle dynamics modelling.

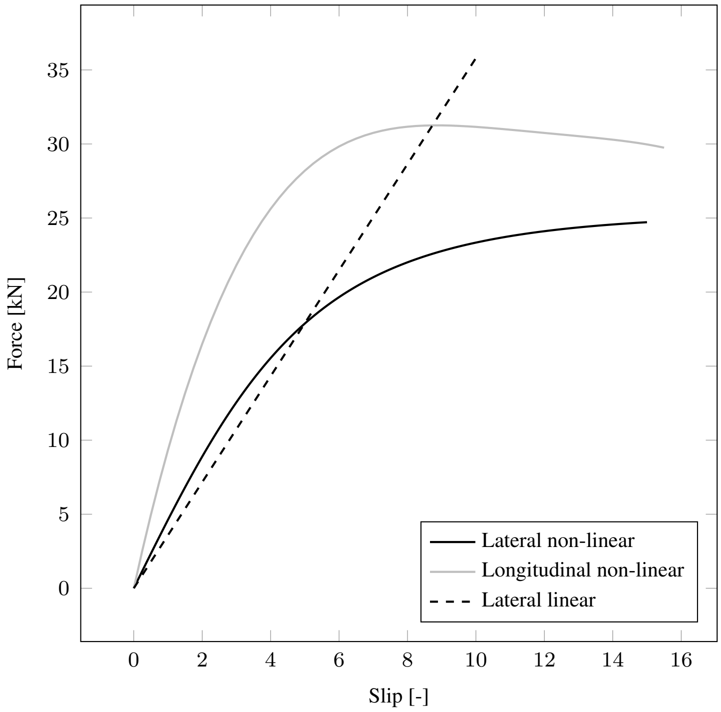

Figure 4.1: Typical non-linear(solid) and linear (dashed) tyre force/slip relations for a non-driven truck front tyre.

4.2.1 Longitudinal forces

Longitudinal tyre forces are of importance when studying vehicle propulsion and braking behaviour. A common physical model for describing the generation of the longitudinal tyre force is the brush model [21]. By using the brush model a longitudinal tyre stiffness, C_{X}, can be calculated as

$$C_{\mathrm{X}} = \left(\frac{\partial F_{\mathrm{XT}}}{\partial S_{\mathrm{X}}} \right)_{S_{\mathrm{X}}=0} \tag{4.1}$$

where F_{XT} is the longitudinal tyre force and S_{X} is the longitudinal slip.
Other commonly used models for describing the longitudinal tyre force are the empirical Magic formula tyre model [21] and the linear tyre model. The linear longitudinal tyre model is defined as

$$F_{\mathrm{XT}} = C_{\mathrm{X}} \cdot S_{\mathrm{X}} \tag{4.2}$$

4.2.2 Lateral forces

As stated above, lateral tyre forces are generated when the travelling direction of the wheel hub is out of line with the wheel rotational plane. This behaviour typically occurs in vehicle cornering. According to the brush model, a lateral tyre slip, S_Y, is defined as the ratio of the wheel hub sliding speed in lateral direction and a reference speed [22]. If no longitudinal slip exists the lateral slip is calculated as the ratio between the sliding speed in lateral direction and the longitudinal speed of the wheel hub and can be expressed by using the side-slip angle, α, as

$$S_Y = \tan(\alpha) \tag{4.3}$$

The brush model can describe the generation of lateral tyre forces, F_{YT}, and define a lateral tyre stiffness, C_Y. If considering small slip angles, the lateral slip can be approximated as the side-slip angle and the lateral tyre stiffness are equal to the cornering stiffness, C_α. The lateral stiffness is calculated as

$$C_{\text{Y}} = -\left(\frac{\partial F_{\text{YT}}}{\partial S_{\text{Y}}}\right)_{S_{\text{Y}}=0} \tag{4.4}$$

Other commonly used models for describing the lateral tyre force are the empirical Magic formula tyre model [21] and the linear tyre model. The linear lateral tyre model is defined as

$$F_{\text{YT}} = C_{\text{Y}} \cdot S_{\text{Y}} \tag{4.5}$$

4.2.3 Combination of longitudinal and lateral forces

Vehicle manoeuvring often involves a combination of steering and propulsion or braking. During such conditions the lateral and longitudinal forces deviate from the values derived under independent conditions. Introduction of longitudinal slip generally tends to reduce the lateral force at a given slip angle and conversely the application of a slip angle reduces the longitudinal force under a given propulsion or braking condition. This behaviour is commonly explained using the friction circle concept [22] which can be expressed as

$$\left(\frac{F_{\text{YT}}}{F_{\text{ZT}}}\right)^2 + \left(\frac{F_{\text{XT}}}{F_{\text{ZT}}}\right)^2 \leq \mu^2 \tag{4.6}$$

where F_{ZT} is the normal force on the tyre and μ is the friction level. In equation (4.6), the friction is assumed to be isotropic, i.e. equal in both the longitudinal and lateral directions. The tyre performance in combined slip condition can be modelled using the empirical Magic formula tyre model [21].

4.3 Vehicle models

4.3.1 Two-track model

In this section the Newtonian formalism is used to derive the dynamic equations of a two-track model representing an A-double combination. The left panel in Figure 4.2 illustrates the spatial parameters of the vehicle model and the right panel illustrates the included motion variables and tyre forces. The considered motion variables v_{Xvi}, v_{Yvi} and $\dot{\psi}_i$ are the longitudinal, lateral and yaw velocities respectively, where $i = 1, ..., n$ with n as the number of vehicle units.

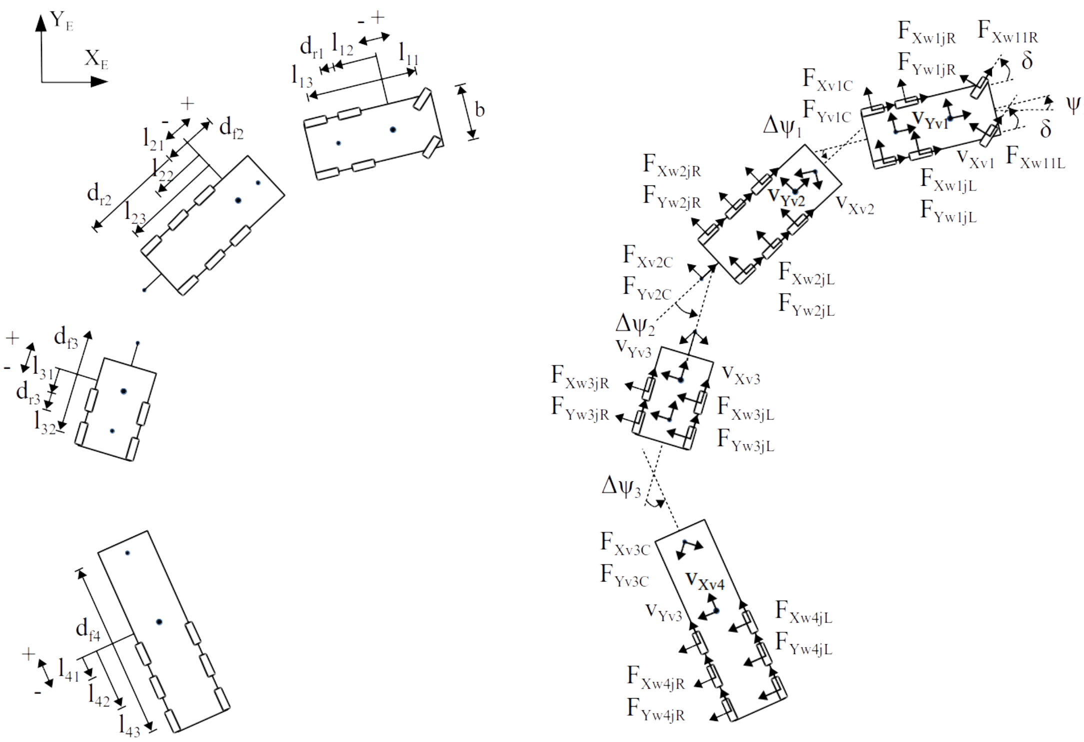

Figure 4.2: Illustration of a two-track model of an A-double combination. The left panel illustrates the spatial parameters of the vehicle model and the right panel illustrates the included motion variables and tyre forces. The spatial parameters are defined as positive if they are in front of the unit's CoG and negative if they are behind the CoG.

In Figure 4.2, the tyre forces, $F_{\mathrm{Xw}ij\mathrm{L}}, F_{\mathrm{Xw}ij\mathrm{R}}, F_{\mathrm{Yw}ij\mathrm{L}}$ and $F_{\mathrm{Yw}ij\mathrm{R}}$, where $j = 1, ..., p_i$ with p_i as the number of axles for unit i and L/R indicating left and right, are shown in the wheel coordinate frames. In order to simplify the expressions of the dynamic equations, the tyre forces are translated to the body-fixed coordinate frames of the respective vehicle unit, $F_{\mathrm{Xv}ij\mathrm{L}}, F_{\mathrm{Xv}ij\mathrm{R}}, F_{\mathrm{Yv}ij\mathrm{L}}$ and $F_{\mathrm{Yv}ij\mathrm{R}}$. The translation from the wheel coordinate frames to the vehicle unit coordinate frames is stated as

$$\begin{pmatrix} F_{\mathrm{Xv}ij\mathrm{R}} & F_{\mathrm{Xv}ij\mathrm{L}} \\ F_{\mathrm{Yv}ij\mathrm{R}} & F_{\mathrm{Yv}ij\mathrm{L}} \end{pmatrix} = \begin{pmatrix} \cos(\delta_{ij}) & -\sin(\delta_{ij}) \\ \sin(\delta_{ij}) & \cos(\delta_{ij}) \end{pmatrix} \cdot \begin{pmatrix} F_{\mathrm{Xw}ij\mathrm{R}} & F_{\mathrm{Xw}ij\mathrm{L}} \\ F_{\mathrm{Yw}ij\mathrm{R}} & F_{\mathrm{Yw}ij\mathrm{L}} \end{pmatrix} \tag{4.7}$$

where δ_{ij} is the steering angle for vehicle unit i and axle j, assuming that the steering angle for the left and right wheels are the same. For the considered A-double combination the steering angles are zero for all axles except for the front axle of the tractor unit.

The equations of motion, expressed for each vehicle unit in a body-fixed coordinate frame, are stated as

$$m_i\left(\dot{v}_{Xvi} - v_{Yvi}\cdot\dot{\psi}_i\right) = \sum_{j=1}^{n} F_{\mathrm{Xv}ij\mathrm{R}} + F_{\mathrm{Xv}ij\mathrm{L}} + F_{\mathrm{Xv}i\mathrm{C}} - F_{\mathrm{Xv}(i-1)\mathrm{C}}\cdot\cos\left(\Delta\psi_{(i-1)}\right) + F_{\mathrm{Yv}(i-1)\mathrm{C}}\cdot\sin\left(\Delta\psi_{(i-1)}\right) \tag{4.8}$$

$$m_i\left(\dot{v}_{Yvi} + v_{Xvi}\cdot\dot{\psi}_i\right) = \sum_{j=1}^{n} F_{\mathrm{Yv}ij\mathrm{R}} + F_{\mathrm{Yv}ij\mathrm{L}} + F_{\mathrm{Yv}i\mathrm{C}} - F_{\mathrm{Xv}(i-1)\mathrm{C}}\cdot\sin\left(\Delta\psi_{(i-1)}\right) - F_{\mathrm{Yv}(i-1)\mathrm{C}}\cdot\cos\left(\Delta\psi_{(i-1)}\right) \tag{4.9}$$

$$J_{\mathrm{Zv}i}\cdot\ddot{\psi}_i = \sum_{j=1}^{n}\left(F_{\mathrm{Yv}ij\mathrm{R}} + F_{\mathrm{Yv}ij\mathrm{L}}\right)\cdot l_{ij} + \left(F_{\mathrm{Xv}ij\mathrm{R}} - F_{\mathrm{Xv}ij\mathrm{L}}\right)\cdot\frac{b}{2} + F_{\mathrm{Yv}i\mathrm{C}}\cdot d_{\mathrm{r}i} - \left(F_{\mathrm{Yv}(i-1)\mathrm{C}}\cdot\cos\left(\Delta\psi_{(i-1)}\right) + F_{\mathrm{Xv}(i-1)\mathrm{C}}\cdot\sin\left(\Delta\psi_{(i-1)}\right)\right)\cdot d_{\mathrm{f}(i-1)} \tag{4.10}$$

where the parameters m_i and $J_{\mathrm{Zv}i}$ are the mass and the yaw mass moment of inertia of unit i, respectively. The articulation angles between the vehicle units are $\Delta\psi_{(i)}$ for the coupling between units i and $i+1$. The coupling forces in longitudinal and lateral direction $F_{\mathrm{Xv}i\mathrm{C}}$ and $F_{\mathrm{Yv}i\mathrm{C}}$, respectively, are defined in a body-fixed coordinate frame of unit i. The spatial parameters are defined as positive if they are in front of the unit's CoG and negative if they are behind the CoG.

Equations (4.8)- (4.10) constitute a $3n$ component system of equations including $3n$ motion variables and $2\,(n-1)$ coupling forces. By combining (4.8) and (4.9) the coupling forces can be eliminated, resulting in a $n+2$ component system of equations. In order to solve the resulting system of equations for the considered motion variables another $2\,(n-1)$ equation related to the couplings between the vehicle units are required. It is noted that the velocity components in the articulation points can be expressed either based on the motion variables of the towing or the trailing unit. The following $(n-1)$ motion constraints are stated as

$$v_{Xvi} = v_{Xv(i+1)}\cdot\cos\left(\Delta\psi_i\right) - \left(v_{Yv(i+1)} + \dot{\psi}_{(i+1)}\cdot d_{\mathrm{r}(i+1)}\right)\cdot\sin\left(\Delta\psi_i\right) \tag{4.11}$$

$$v_{Yvi} + \dot{\psi}_i\cdot d_{\mathrm{r}i} = v_{Xv(i+1)}\cdot\sin\left(\Delta\psi_i\right) + \left(v_{Yv(i+1)} + \dot{\psi}_{(i+1)}\cdot d_{\mathrm{r}(i+1)}\right)\cdot\cos\left(\Delta\psi_i\right) \tag{4.12}$$

Another $(n-1)$ equation is given by differentiating (4.11) and (4.12) with respect to time. The differentiation results in

$$\dot{v}_{Xvi} = \left(\dot{v}_{Xv(i+1)} - v_{Yv(i+1)} \cdot \Delta\dot{\psi}_i - \dot{\psi}_{(i+1)} \cdot d_{\mathrm{r}(i+1)} \cdot \Delta\dot{\psi}_i\right) \cos\left(\Delta\psi_i\right) \tag{4.13}$$
$$- \left(v_{Xv(i+1)} \cdot \Delta\dot{\psi}_i - \dot{v}_{Yv(i+1)} - \ddot{\psi}_i \cdot d_{\mathrm{r}(i+1)}\right) \cdot \sin\left(\Delta\psi_i\right)$$
$$\dot{v}_{Yvi} + \ddot{\psi}_i \cdot d_{\mathrm{r}i} = \left(v_{Xv(i+1)} \cdot \Delta\dot{\psi}_i + \dot{v}_{Yv(i+1)} + \ddot{\psi}_{(i+1)} \cdot d_{\mathrm{r}(i+1)}\right) \cos\left(\Delta\psi_i\right) \tag{4.14}$$
$$+ \left(\dot{v}_{Xv(i+1)} - v_{Yv(i+1)} \cdot \Delta\dot{\psi}_i - \dot{\psi}_{(i+1)} \cdot d_{\mathrm{r}(i+1)} \cdot \Delta\dot{\psi}_i\right) \cdot \sin\left(\Delta\psi_i\right)$$

The dynamic equations of the vehicle model representing inertial and kinematic vehicle properties are given by combining (4.8)- (4.14). In order to express the final vehicle model, constitutive relations for the tyre forces, out of plane equilibria and constitutive relations for the axle suspension are required. This part has been omitted.

4.3.2 One-track model

In this section a one-track model of an A-double combination is derived using the Lagrangian formalism. The left panel in Figure 4.3 illustrates the spatial parameters of the vehicle model and the right panel illustrates the included motion variables and tyre forces.

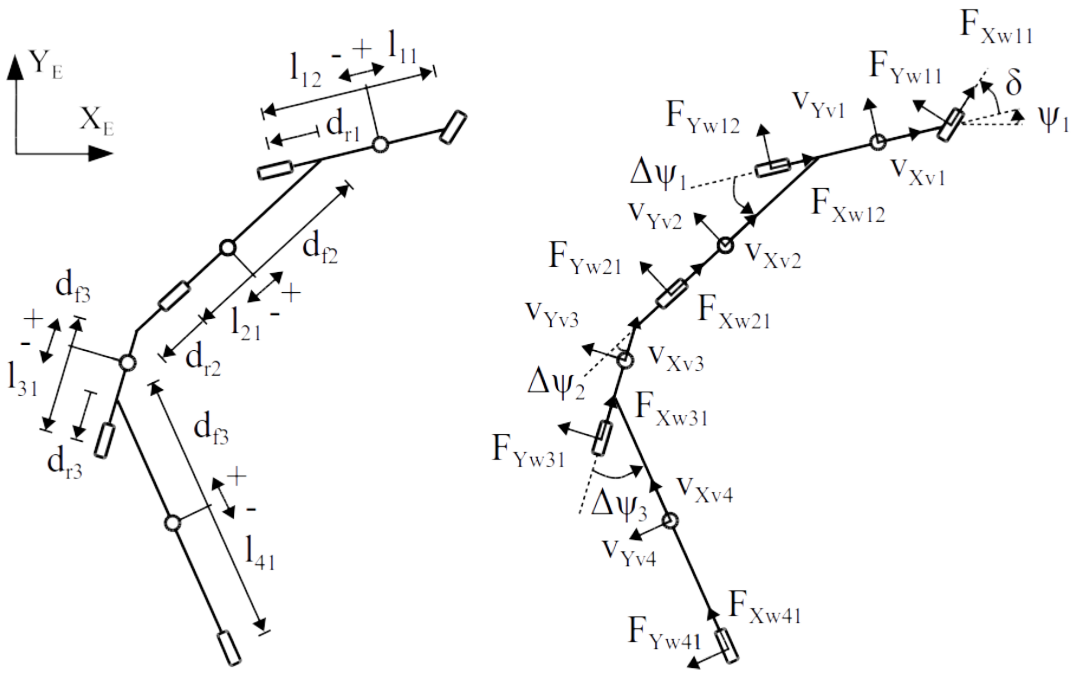

Figure 4.3: Illustration of a one-track model of an A-double combination. The left panel illustrates the spatial parameters of the vehicle model and the right panel illustrates the included motion variables and tyre forces. The spatial parameters are defined as positive if they are in front of the unit's CoG and negative if they are behind the CoG.

Besides the introduction of a virtual tyre for each vehicle axle, the concept of equivalent wheel-base [23], meaning that groups of axles are collapsed into one axle, is introduced and utilized. By combining these concepts the resulting vehicle model consists of a tractor unit including two axles with one tyre per axle and trailing units including one axle and one tyre. The equivalent wheel-base of a multi-axle vehicle is the wheel-base of a two-axle vehicle with similar steady-state turning behaviour as the multi-axle vehicle. Assuming linear lateral tyre forces, the equivalent wheelbase can be calculated as

$$l_{\text{eq}} = L_{wb} \cdot \left(1 + \frac{T_f}{L^2} \cdot \left(1 + \frac{C_{\alpha\text{R}}}{C_{\alpha\text{F}}}\right)\right) \tag{4.15}$$

where L_{wb} is the wheelbase calculated as the distance from the front axle to the longitudinal position and the moments generated by vertical loads of the rear axles add up to zero. $C_{\alpha\text{F}}$ and $C_{\alpha\text{R}}$ are front axle cornering stiffness and sum of rear axle cornering stiffness's respectively. The tandem factor T_f is calculated as

$$T_f = \frac{\sum_{k=1}^{M} \Delta_k^2}{M} \tag{4.16}$$

where M is the number of rear axles and Δ_k is the longitudinal distance from the rear end of L_{wb} to the k^{th} rear axle.

The Lagrangian equations are defined as

$$\frac{d}{dt}\frac{\partial T}{\partial \dot{q}_i} - \frac{\partial T}{\partial q_i} + \frac{\partial V}{\partial q_i} = Q_i \tag{4.17}$$

where $i = 1, ..., N$ with N as the number of generalized coordinates. The generalized coordinates q_i are the dependent variables, T is the kinetic energy, V is the potential energy and Q_i are the generalized forces. Due to the fact that only planar motion is to be considered the potential energy, V is zero.

For the A-double combination the following set of generalized coordinates are chosen

$$q = \left[\bar{X}_1,\ \bar{Y}_1,\ \psi_1,\ \Delta\psi_1,\ \Delta\psi_2,\ \Delta\psi_3\right] \tag{4.18}$$

where $\bar{X}_1$ and $\bar{Y}_1$ are the longitudinal and lateral position of the CoG for the tractor unit expressed in an inertial coordinate frame. ψ_1 is the heading angle of the tractor unit and $\Delta\psi_1, \Delta\psi_2$ and $\Delta\psi_3$ are the articulation angles of the trailing vehicle units.
To improve the usability of the model, the velocities of the tractor unit are expressed relative to a body-fixed coordinate frame. However, the longitudinal and lateral velocities, v_{Xv1} and v_{Yv1} respectively, are not derivatives of generalized coordinates. They can however be regarded as derivatives of quasi-coordinates and therefore can be introduced in the Lagrange equations. In order to derive the Lagrange equations using the quasi-coordinates the approach presented in [21] can be followed.

The generalized forces represent the external forces, F_k, applied to the system in terms of components along the generalized coordinates. The generalized forces can be calculated as

$$Q_i = \sum_{k=1}^{p} F_k \cdot \frac{\partial r_{tyre,k}}{\partial q_i} \tag{4.19}$$

where p is the number of vehicle axles, $i = 1, ..., N$ with N as the number of generalized coordinates, and $r_{tyre,k}$ are the positions of the tyres.

The kinetic energy of the system is calculated as

$$T = \frac{1}{2} \cdot \sum_{i=1}^{n} m_i \cdot v_i^2 + \cdot J_i \cdot \dot{\psi}_i^2 \tag{4.20}$$

where n is the number of vehicle units, v_i and $\dot{\psi}_i$ are the translational and rotational velocities of unit i, m_i are the masses and J_{Zi} are the yaw moments of inertia of unit i. The translational velocity components in the CoG of vehicle units are calculated using the corresponding position vectors as a starting point. The position vectors $\vec{r}_i$, are expressed relative the CoG of the tractor unit as

$$\vec{r}_1 = \bar{X}_1 \cdot \vec{e}_{\mathrm{X_E}} + \bar{Y}_1 \cdot \vec{e}_{\mathrm{Y_E}} \tag{4.21}$$

$$\vec{r}_2 = \vec{r}_1 + R(\psi_1) \cdot \begin{pmatrix} l_{12} + d_{\mathrm{r1}} \\ 0 \end{pmatrix} + R(\psi_2) \cdot \begin{pmatrix} d_{\mathrm{f2}} \\ 0 \end{pmatrix} \tag{4.22}$$

$$\vec{r}_3 = \vec{r}_2 + R(\psi_2) \cdot \begin{pmatrix} l_{21} + d_{\mathrm{r2}} \\ 0 \end{pmatrix} + R(\psi_3) \cdot \begin{pmatrix} d_{\mathrm{f3}} \\ 0 \end{pmatrix} \tag{4.23}$$

$$\vec{r}_4 = \vec{r}_3 + R(\psi_3) \cdot \begin{pmatrix} l_{31} + d_{\mathrm{r3}} \\ 0 \end{pmatrix} + R(\psi_4) \cdot \begin{pmatrix} d_{\mathrm{f4}} \\ 0 \end{pmatrix} \tag{4.24}$$

where $i = 1, ..., n$ with n as the number of vehicle units, and $\vec{e}_{\mathrm{X_E}}$ and $\vec{e}_{\mathrm{X_E}}$ are unit vectors. $R(\psi_{\mathrm{i}})$ are the rotation matrices in 2D Euclidean space defined as

$$R(\psi_{\mathrm{i}}) = \begin{pmatrix} \cos(\psi_{\mathrm{i}}) & -\sin(\psi_{\mathrm{i}}) \\ \sin(\psi_{\mathrm{i}}) & \cos(\psi_{\mathrm{i}}) \end{pmatrix} \tag{4.25}$$

Starting from (4.21)- (4.24), the translational velocity vectors of the CoG of the vehicle units are calculated as

$$\vec{v}_{\mathrm{i}} = \frac{d}{dt}\vec{r}_{\mathrm{i}} + R(\psi_{\mathrm{i}}) \cdot \vec{r}_{\mathrm{i}} \tag{4.26}$$

where $i = 1, ..., n$, with n as the number of vehicle units.

The dynamic equations of the one-track model representing inertial and kinematic vehicle properties are given by (4.17)- (4.26). In order to express the final vehicle model, constitutive relations for the tyre forces are also required.

4.4 Performance based characteristics

To evaluate the performance of LVCs, fourteen performance based characteristics (PBCs) for the longitudinal and lateral directions have been defined [24]. The PBCs for longitudinal direction are: startability, gradeability, acceleration capability, stopping distance, and down-grade holding capability. The PBCs for lateral direction are: rearward amplification (RWA), swept path width (SPW), high speed transient off-tracking (HSTO), high speed steady-state off-tracking (HSSO), yaw damping coefficient (YDC), straight line off-tracking (SLO), lateral clearance time (LCT), steady-state rollover threshold (SRT), and deceleration capability in a turn. The most important lateral characteristics for high speed manoeuvring are RWA, HSSO, HSTO and YDC, which are described as follows:

- RWA is the relationship between the maximum motion of the first and last vehicle units during a specified steering manoeuvre [25] and vehicle speed and is usually given in the metrics lateral acceleration gain or yaw velocity gain. It expresses the increased risk for a last unit roll-over or swing-out, respectively. Such a problem can typically occur if a sudden steering manoeuvre is performed. An example of the RWA in a lane change manoeuvre is illustrated in Figure 4.4.

- The off-tracking characteristics, HSSO and HSTO, both describe the lateral deviation between the path of the front axle and the path of the most severely off-tracking axle of the last unit. These measures express the additional space needed for the last unit in a specific steering manoeuvre and vehicle speed. An example of the HSTO in a lane change manoeuvre is illustrated in Figure 4.4.

- The YDC is the damping ratio of the least damped articulation joint's angle during free yaw oscillations of the vehicle combination after a specific steering manoeuvre and vehicle speed. A longer decay time might result in higher driver workload and increased safety risk for other road users.

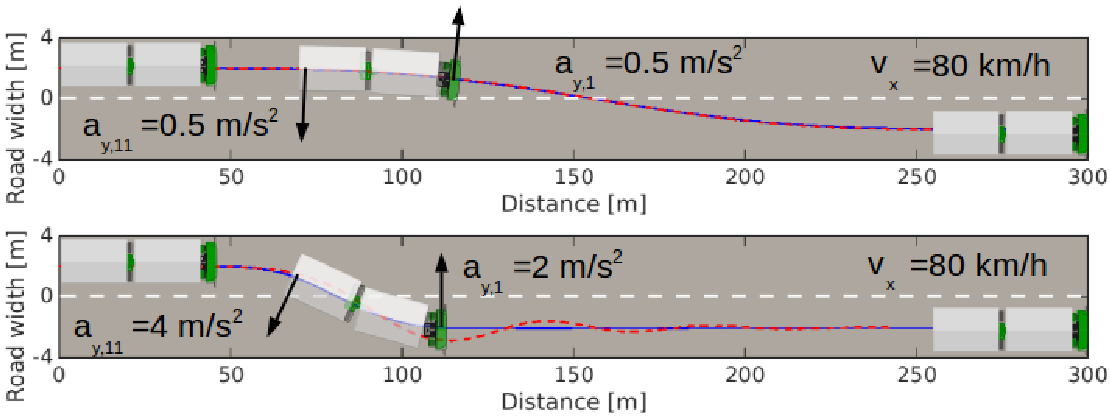

Figure 4.4: Illustration of performance characteristics for a LVC in a normal (top) and critical (bottom) highway lane change manoeuvre at 80 km/h with a focus on the lateral acceleration rearward amplification and lateral off-tracking. The rearward amplification is approximately 1 in the normal manoeuvre and 2 in the critical manoeuvre. The lateral off-tracking between the first and last axles in the vehicle combination is about 0.1 m in the normal manoeuvre and 1 m in the critical manoeuvre. The solid blue line and the dashed red line illustrate the path of the first and last vehicle axles, respectively.

4.5 LVC prediction models

4.5.1 Required validity range

All vehicle models have a validity range that is coupled to the studied vehicle manoeuvring. During a high-speed dynamic manoeuvre, such as a lane change, the RWA increases the risk of reaching the roll-over threshold of the vehicle combination. The lateral vehicle dynamics are in the frequency range of 0-1 Hz. The SRT in combination with RWA is thus the most important PBC with regards to the validity range of LVC prediction models, see Section 4.4.

As mentioned, the TSP manoeuvres generated for LVC automation must avoid vehicle roll-over. Therefore, the validity range in lateral acceleration is typically below $0.35g$, see Figure 4.5. Figure 4.5 illustrates that within the heavy vehicle fleet in New Zeeland, vehicles with a SRT of $0.3g$ or less have more than 3 times the crash rate in comparison to the rest of the fleet. The recommended minimum SRT in New Zeeland is $0.35g$. A two-track model, see Section 4.3.1, with load transfer, can predict roll-over which is beyond the foreseen usage of TSPs. In addition, the two-track model requires the actuator configuration to be on wheel level which generates a computationally demanding model with many states and multiple inputs. Instead, a one-track model, see Section 4.3.2, with simplified actuation requests of steering and longitudinal acceleration is predicted to be sufficient for the considered lateral acceleration range. The lateral acceleration range also minimizes the usage of non-linear tyre forces. Therefore linear tyre modelling, see Figure 4.1, should be sufficient in combination with a one-track model for TSP generation of LVCs in normal manoeuvring for the frequency range of 0-1 Hz. Simplifying the one-track model further by using zero slip tyre modelling, i.e. a kinematic model has shown to be insufficient, see Section II, Paper 1.

4.5.2 Motion constraints

The PBCs discussed in Section 4.4 in combination with requirement limits, see Section 2.2, are intended to be used in the regulations for road approval of LVCs and are not directly coupled to the prediction models. However, the PBCs show important motion requirements of the LVCs that must also be considered in the TSP generation. The motion constraint of lateral acceleration on the last vehicle unit is important for capturing the effects of RWA and minimizing the risk of roll-over. To maintain the vehicle combination within the desired lane, another motion constraint is defined, the perpendicular distance offset between center of the axle and the road center-line which in PBC is represented by HSTO. These and other motion constraints have been used, see Section III, Paper 4.

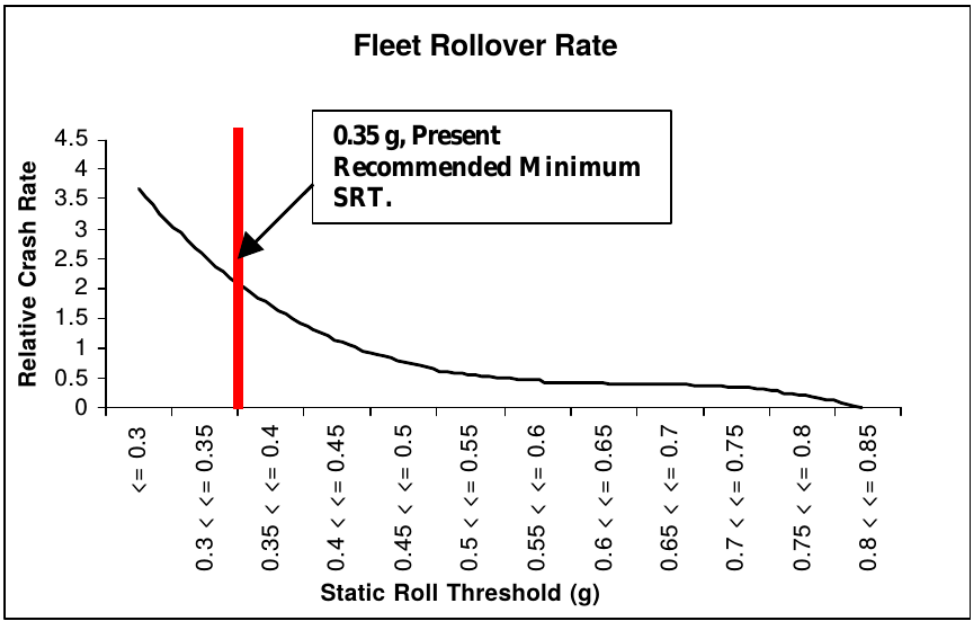

Figure 4.5: Illustration of relative crash rate versus static roll-over threshold for heavy vehicles in New Zeeland. The recommended minimum static roll-over threshold (SRT) in New Zeeland is $0.35g$ (units of g). Figure from Mueller et al. [26].

Chapter 5

Automated driving of LVCs

The future of automated driving of LVCs will most likely include supervision from a human driver and is therefore also described as a driver support system. Driver support systems for LVCs with full or partial automation of steering and propulsion/braking, are envisioned for the driving scenarios described in Chapter 3 in order to enhance traffic safety, transport efficiency and productivity, and the driver working environment. However, even though there has been an intense evolution within the field of intelligent vehicles and driver support systems in the last decades [27, 28], there is still much work that remains to be done before such systems can be commercially available.

Perhaps the most important question within the development of driver support systems is driver acceptance. This because it has a high impact on the system's practical efficiency [29]. Driver acceptance is however a highly complex question which comprises both engineering and behavioural disciplines.

The research in this thesis focuses on understanding the necessary basis for generating TSPs in order to provide driver support systems with a high driver acceptance. Papers 1 and 2 studied the driver acceptance in automated driving by comparing two different ways of generating actuation requests for steering and braking/propulsion. Manual driving was also investigated during cornering and compared with results from post-experiment optimizations. In Papers 3 and 4, a driver model based algorithm for TSP generation was proposed. The utilized driver model was based on human perception and cognition and was envisioned to generate TSPs with high driver acceptance.

5.1 Traffic situation predictions

In order to realize the envisioned architecture for motion functionality illustrated in Chapter 1, traffic situation predictions (TSPs) including the subject and surrounding vehicles are needed for a time horizon of up to 10 s. The TSPs have a tactical character and include the motion variables of the vehicle combination that define a relevant driving manoeuvre. TSPs in a multiple lane one-way road scenario are illustrated in Figure 5.1. The main requirements on a TSP algorithm for LVCs are that

- it can generate feasible collision-free paths.
- it is sufficiently computationally efficient.

- it includes the main motion constraints related to the vehicle dynamics of LVCs. For example, the lateral dynamics of LVCs in high speed manoeuvres can be highly amplified compared to current vehicle combinations and, if ignored, can result in a roll-over accident.

There are several ways of generating TSPs [30], whereof some are more relevant for including a high level of motion constraints. Regardless of which algorithm is used, it is questionable to what extent the overall driver acceptance of the automated driving functionality is affected by the TSP generation. For example, many interesting TSP algorithms are based on optimization [31] and the TSPs are then a result of the formulation of the cost function and the constraints. Relevant in this case is how these should be formulated in order to generate TSPs that result in a high driver acceptance.

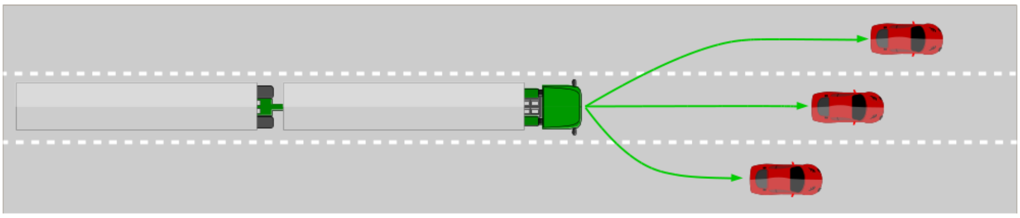

Figure 5.1: Illustration of traffic situation predictions (TSPs) in a multiple lane one-way road scenario.

5.2 Single lane road driving (Papers 1 and 2)

Papers 1 and 2 are mainly based on results of a driving simulator study in which manual and automated driving of an A-double LVC were studied. The driving scenario was comprised of a relatively curvy and hilly single-lane Swedish county road (180), without additional road users and safety critical events. Two automated driving strategies for steering, propulsion and braking were formulated, whereof one of the steering strategies included resulted from an optimal control based receding horizon approach [32]. Based on subjective ratings and comments, both automated driving strategies were appreciated for their lane positioning and driving performance, with a slight preference for the strategy based on optimal control. However, for both strategies reoccurring comments referred to harsh decelerations before curves.
Furthermore, results from post-experiment optimization, where a one-track model was simulated in a cornering situation, were used in the evaluation. The cost functions were then formulated as a minimization of the final time (opt 1), a minimization of the acceleration vector of the truck front axle (opt 2), and a minimization of the corresponding jerk vector (opt 3). When analysing manual driving trajectories from cornering, it was observed that the utilized accelerations had a round shape. A similar shape was found when using an objective function which included minimizing the resultant jerk vector, as illustrated in Figure 5.2. This suggests that the combined steering and braking should prioritize a smooth and comfortable driving experience. This result was implemented in

a real-time model predictive control based framework for trajectory generation [31]. The framework was successfully used in a moving-base driving simulator study where lane change gap acceptance and combined steering and braking in lane changes were investigated. The results from the driving simulator study have not yet been fully analysed and are omitted in this thesis.

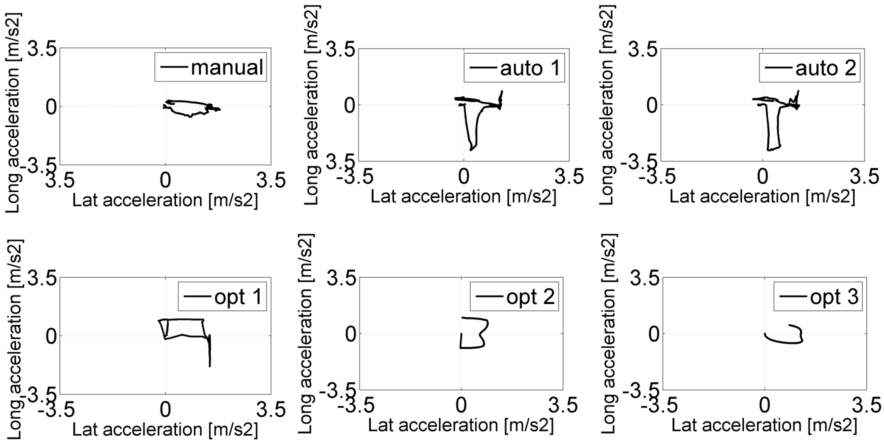

Figure 5.2: G-g diagram for a left curve with minimum radius of 150 m. The manual drivers had a mean speed of approximately 60 km/h during cornering. Mean values of manual driving (top left), automated driving 1 (top center), automated driving 2 (top right), opt 1 (bottom left), opt 2 (bottom center) and opt 3 (bottom right).

5.3 Multiple lane one way road driving (Papers 3 and 4)

Papers 3 and 4 studied lane change manoeuvring, including an A-double, on relatively straight and flat multiple lane one-way roads with surrounding traffic. In Paper 3, a combined longitudinal and lateral driver model based on the approaches in [33, 34], combined with a one-track prediction model, was proposed for the generation of TSPs. The parametrization of the lateral driver model was based on data collection of lane change manoeuvres during an actual transport mission carried out within the DUO^2 project [35]. The main frequency of the steering behaviour in a lane change at 80 km/h was found to be approximately 0.1 Hz, which is well below the critical vehicle yaw mode frequencies. The steering wheel angle and the driver model fit for one of the measured lane change events are shown in Figure 5.3. Aside from a good correlation of the main steering frequency and amplitude it was also observed that the lane change included a second peak when the vehicle was straightened. Paper 3 hypothesised that a high driver acceptance can be reached if the TSPs would exhibit a human-like control behaviour, both regarding heuristics and safety thresholds.

In Paper 4, the combined longitudinal and lateral driver model proposed in Paper 3, was utilized in a real-time framework for automated highway driving. The included driving manoeuvres were maintain lane, lane change to right and left, abort lane change to right and left, and emergency brake. The TSPs, generated in a receding horizon fashion, were used for actuation request and the evaluation of dynamic constraints related to

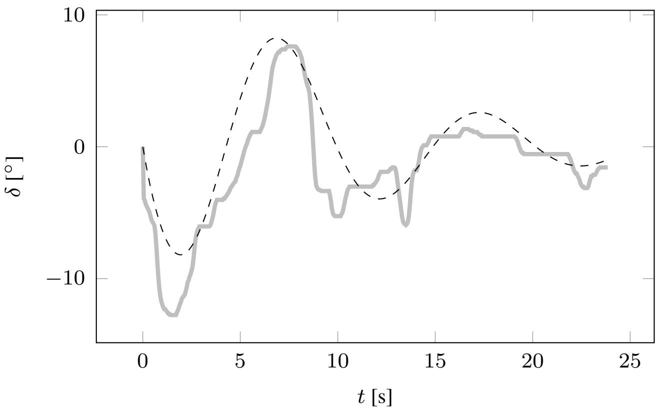

Figure 5.3: Steering wheel angle (solid) and the driver model fit (dashed) for one of the measured lane change events.

the subject vehicle dynamics, road boundaries and distance to surrounding objects. The framework was evaluated for lane changes at varying constant velocities and during braking and was later successfully used in the moving-base driving simulator study mentioned above in Section 5.2.

Chapter 6

Discussion and prospective future work

6.1 Discussion

In order to study and design TSPs envisioned in automated driving functionalities for LVCs, performance characteristics of LVC usage and driver acceptance have been analysed and characterized using both objective and subjective data from a moving-base driving simulator study. The methodology of using driving simulators includes both advantages and disadvantages when compared to physical testing.
The main advantages in the considered situation are firstly that the studied vehicle combinations are not in general traffic usage. Physical testing would therefore require test-track areas or special traffic permissions. Secondly, vehicle environment sensors and vehicle state estimation are currently not commercially available for the considered vehicle combinations. When using driving simulators focus can be placed directly on the functionality of the motion control. Thirdly, driving simulators offer high controllability, reproducibility and the possibility of encountering dangerous driving conditions without being physically at risk. On the other hand, the disadvantages with driving simulators are limited physical, perceptual, and behavioural fidelity. To accomplish satisfactory realism in regards to truck vehicle dynamics, extensive subjective testing has been carried out by experienced truck drivers. Real road environments have also been utilized to increase the degree of recognition and thereby increase the perceptual fidelity. To summarize, the use of driving simulator experiments have enabled studies on driver acceptance when developing automated driving functionalities for LVCs. One hypothesis to enable automated functionalities for LVCs is that automated and manual driving behaviour must be treated simultaneously in order to find suitable solutions. However, the results derived in a driving simulator must be interpreted with caution because of the limited fidelity.

6.2 Prospective future work

In continuation of this research, subjective and objective data from a completed moving-base driving simulator study will be analysed. The objectives of the simulator study were to investigate lane change gap acceptance and utilization of combined braking and steering in lane change manoeuvres. The gap acceptance study was carried out using a tractor semi-trailer combination and an A-double combination whereas the combined steering and braking study was carried out using the A-double only. In addition to manual driving, two different methods for generating TSP for the driving manoeuvres maintain lane and lane change, were implemented in a driver assist functionality including combined automated steering and propulsion/braking. The driver acceptance of the functionality and its connection to the TSP generation will be analysed and discussed.

The methods used for TSP generation in the driving simulator study will be objectively compared and analysed in off-line simulations of maintain lane and lane change manoeuvring. This is in order to increase the understanding of the limitations of the respective methods and the situations in which each method is most suitable.

The driver model developed and utilized in Papers 3 and 4, will be further developed with respect to a hierarchical structure for different driving modes e.g. maintain lane, lane changes or collision avoidance. Also, real-time optimization of the driver model parameters can be considered as one possible way to increase the feasibility when used in TSP generation.

The one-track vehicle model used in the TSP generation in Papers 3 and 4 will be further analysed with respect to its validity range. The analysis will first consider normal manoeuvring to investigate if simplifications e.g. steady-state assumptions, can be made in order to reduce the demands on computational resources, and secondly consider, safety critical manoeuvres e.g. automatic emergency braking to study effects from combined tyre slip modelling. The adaptation of vehicle models towards more advanced motion actuation is also a topic of special interest for LVCs, e.g. including steering and propulsion on more axles than the traditional ones.

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