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Roll dynamics and tyre relaxation in heavy combination vehicle models for transient lateral manoeuvres

Master's thesis in Vehicle Dynamics

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Photo from Volvo GTT/Niklas Fröjd. Volvo's 90-ton dispensation vehicle ETT, in a lane change

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Abstract

Long combination vehicles are the future of road transportation. They have much better energy and transport efficiency compared to traditional transport vehicles. There is a huge interest in using long combination vehicles for transporting goods in Sweden. In Finland, long combination vehicles (34.5 m, 76 ton) were introduced to the roads in the end of January 2019.

In Sweden, performance based standards, or PBSs, are becoming an accepted tool for evaluating LCV's safety and transport efficiency. The goal of this thesis is to produce the simplest possible vehicle models, for the performance based standards that concern lateral dynamics. These models will be used to evaluate those performance based standards, with a minimum number of parameters. The models should include lateral dynamics, roll dynamics due to high load and tyre relaxation. Non-linear tyre model will also be implemented to the simulation models. In simulations, single lane change manoeuvre was used to evaluate the performance of the created vehicle models.

The finalized models were validated against high-fidelity models from Volvo GTT. The models were created in Modelica language and implemented to OpenPBS library.

Key words: *Roll dynamics, Tyre relaxation, Vehicle Dynamics, Modelica, Performance based Standards*

Contents

Preface

This master's thesis project was conducted during a time period of 6 months from February 2019 to July 2019. Thesis was done at Chalmers University of technology with support from Volvo GTT and University of Oulu.

I would like to thank my examiners and supervisors for all their help during this thesis work. Special thanks for my examiner, professor, Bengt Jacobson and my supervisor, Vehicle dynamics specialist, Niklas Fröjd for giving me the chance to work on this project in Sweden and their continuous help during this six month period. I would also like to thank my supervisors in Finland, PhD-student Miro-Tommi Tuutijärvi and professor emeritus Mauri Haataja, who have done an amazing work teaching automotive engineering in University of Oulu.

Finally, I would like to thank my amazing family, who have supported me during my studies in University of Oulu and have always encouraged me to keep chasing my dreams.

Göteborg 2019-07-15

Ville Santahuhta

Notations and Abbreviations

Abbreviations

CAT	Centre Axle Trailer
COG	Centre Of Gravity
DR	Damping Ratio
$D.O.F$	Degree Of Freedom
FMI	Functional Mock-up Interface
HCT	High Capacity Transport
$HSTO$	High Speed Transient Off tracking
LCV	Long Combination Vehicle
LLT	Lateral Load Transfer
LTR	Load Transfer Ratio
OEM	Original Equipment Manufacturer
PBS	Performance Based Standards
RWA	Rearward Amplification
SRT	Static Roll Threshold
YD	Yaw Damping

Roman upper-case letters

A	Front coupling position from first axle of the unit
A_{cog}	Front coupling position from CoG of unit
B	Rear coupling position from first axle of the unit
B_{cog}	Rear coupling position from CoG of unit
C	Shape factor, non-linear tyre model
Cc	Cornering coefficient
CC_y	Cornering coefficient, non-linear tyre model
$CC_{y,0}$	Cornering coefficient at nominal tyre normal force, non-linear tyre model
C_{st}	Axle cornering stiffness
F_{cx}	Longitudinal coupling force
F_{cy}	Lateral coupling force
F_{cz}	Vertical coupling force

FOH	Front overhang of unit
F_x	Longitudinal force
F_{xw}	Longitudinal tyre force
F_y	Lateral force
$F_{YT,SS}$	Lateral tyre force, steady state, non-linear tyre model
F_{yw}	Lateral tyre force
F_z	Vertical force
F_{zl}	Load on unit's left side tyres
F_{zr}	Load on unit's right side tyres
F_{ZT}	Tyre's normal force, non-linear tyre model
$F_{ZT,0}$	Tyre's nominal normal force, non-linear tyre model
I_{xs}	Unit's inertia in roll plane
I_z	Unit's inertia in yaw plane
L	Axle positions
L_{cog}	Distance from unit's CoG to unit's axles
L_{C-cog}	Distance from coupling to CoG of unit
L_r	Tyre relaxation length
M_x	Roll moment in roll centre
M_{xd}	Torque created by dampers
M_{xs}	Torque created by springs
ROH	Rear overhang of unit
W	Track width of unit
X	CoG position from the first axle of the unit

Roman lower-case letters

ccg_y	Maximum cornering coefficient gradient, non-linear tyre model
c_{roll}	Roll stiffness
c_s	Roll damping
d_{roll}	Roll damping
g	Fall acceleration
h	CoG height of unit
h_C	Coupling height
h_{RC}	Roll centre height of unit
i	Unit

k_s	Roll stiffness
m	Mass of unit
na	Number of axles in unit
nu	Number of units in combination vehicle
p_x	Roll angle of unit
t	Time
$u_{g,y}$	Maximum lateral force gradient, non-linear tyre model
u_y	Maximum lateral force coefficient, non-linear tyre model
$u_{y,0}$	Maximum lateral force coefficient at nominal vertical tyre force, non-linear tyre model
u_2	Slide friction ratio, non-linear tyre model
v_x	Longitudinal velocity of unit
v_y	Lateral velocity of unit
v_{yc}	Lateral velocity of coupling
v_{ys}	Lateral velocity of sprung mass
v_{yu}	Lateral velocity of unsprung mass
x_1	Measured value 1 for calculating yaw damping
x_2	Measured value 2 for calculating yaw damping

Greek lower-case letters

α	Tyre's lateral slip angle
α_y	Tyre's lateral slip angle, non-linear tyre model
$\acute{\alpha}_y$	Tyre's relaxed lateral slip angle, non-linear tyre model
δ	Steering angle
θ	Articulation angle
ω_x	Roll rate of unit
ω_z	Yaw rate of unit
$\omega_{z,i}$	Yaw rate of unit i
$\omega_{z,1}$	Yaw rate of unit 1

SI units and radians used where not else is stated

1 Introduction

This chapter contains basic introduction to this thesis. It presents thesis' background, problem motivating the project, envisioned solution and thesis' research questions. In the end it goes through deliverables of this project and the limitations in this thesis.

1.1 Background

This thesis is a part of a larger project called "Performance Based Standards II" involving Chalmers university of Technology, Volvo Group, Swedish Transport Administration, Scania, The Swedish National Road and Transport Research institute, Parator Industri, Transportstyrelsen, Nokian Tyres and University of Oulu (Chalmers University of technology, 2019). Project's goal is to produce performance based standards for High Capacity Transport (HCT) vehicles that correspond better to Swedish and Nordic conditions. Performance based standards (PBS) are a set of regulations that specify the vehicle performance required to operate safely and transport efficiently on public roads. The PBS approach on vehicle legislation will allow the development of cost effective and eco-friendly HCT vehicles, without negative side effects on traffic safety or infrastructure.

One part of this project is to develop an "OpenPBS tool". This tool should make possible to easily evaluate the stability and other PBS measures of HCT vehicles in an understandable and repeatable way. The first beta version of the tool is already published, and it is open and free. The tool is available from *GitHub*. The OpenPBS tool uses Modelica format, and most of this thesis' simulation is done in open source software called OpenModelica. (OpenModelica, 2019) The goal of this tool is to be able to virtually verify the PBSs and implement them in a computer tool. (Jacobson, et al., 2017, p. 7) The future version of this tool could be used as a part of approval of individual combination vehicles in a web-application, which could be an update of the web service by Transport Styrelsen (Transport Styrelsen, 2019). It can also be used by vehicle manufacturers and transport operators to develop combinations that are compliant with the legislation. There are multiple different PBS measures defined currently. Some of these are for example: Startability, tail swing, steady state rollover and rearward amplification.

In this thesis the primary stress is in the HCT vehicles and their yaw and rollover stability. HCT vehicles are capable to transport larger amount of goods because of their larger payload capacity. Typical maximum weight of HCT vehicle is 74 tons and it was introduced in Sweden in April 2018. Larger payload capacity makes HCT vehicles very efficient and cost-effective alternative to run on the road. Larger environmental and economic benefits will be obtained with Long Combination Vehicles (LCV), where the maximum length of the combination can be as high as 35 meters instead of current maximum of 25.25 meters. These long combination vehicles were allowed in the road in Finland in January of 2019. Larger payload capacities will have a negative effect on the stability of these vehicles. The difference comes mostly of their higher centre of gravity height and additional vehicle units.

1.2 Problem motivating the project

The Open PBS tool must use as simple vehicle model as possible. This way the number of parameters stays low, compared to full vehicle 3D-models for example in Adams/Car, Truckmaker/Trucksim or other non-public models used in automotive industry. The model still must be complex enough to represent influence of the important parameters. Current version of the OpenPBS tool does not satisfy these demands well enough. The lack of roll influence and tyre relaxations influence makes the model too simple, and thus, it doesn't give information that is accurate enough for PBS measures in situations with high lateral acceleration. Examples of these PBS measures are: rearward amplification (RWA), High Speed Transient Off tracking (HSTO), Load Transfer Ratio (LTR) and Damping Ratio (DR), which can differ up to 50 % from high fidelity models. Influence of centre of gravity height and tyre relaxation are identified as likely causes for the difference. (Islam, et al., 2019).

1.3 Envisioned solution and objective

In a response to the previously mentioned challenge, some improvements to the OpenPBS tool needs to be made. The envisioned solution is to add the important physical phenomena to the 2D one-track model, which results in a model which is advanced enough to represent the influence of the centre of gravity height and tyre relaxation.

This thesis answers to these two questions:

- Which phenomena needs to be modelled to better capture the influence of centre of gravity height and tyre relaxation for PBS measures for heavy combination vehicles?
- How to model these phenomena, i.e. which new/changed equations and parameters are needed?

1.4 Deliverables

There are multiple deliverables in this thesis project, which create the framework for the thesis work. The deliverables are listed below:

- Vehicle model on Modelica format, with roll dynamics and tyre relaxation in in-road-plane motion model.
- Model integrated in OpenPBS tool and released on Gitlab.
- New tyre models implemented in OpenPBS.
- Model validated versus high fidelity models.

1.5 Limitations

There are some limitations in this thesis, that will affect the accuracy of the simulation models and thesis work. These limitations are listed below:

- Only a few example vehicles will be used for model validation.
- Model validation will only be versus to high-fidelity modelling library, not real vehicle tests.

- In the simulation models only steering on first axle of towing vehicle will be used.
- Influence from flexible frames was not included in the targeted model, but a small investigation was included, see appendix 1.

1.6 Research method

Timewise, this thesis is divided into multiple stages. It's done so that the accuracy of the model should increase, when nearing the end of this thesis project. The thesis work consists of:

- Creating multiple different candidate phenomenon, with additional elements compared to traditional 2D-one-track -model.
    - Roll influence on 2D one-track model.
    - Tyre relaxation on 2D one-track model.
    - Non-linear tyre model in 2D one-track model
- Selecting the best and most comprehensive phenomena.
    - The selection has to do the balance between as few and easily standardized parameters as possible and agreement with more advanced models
    - Comparison against previously done research.
- Validate the selected phenomenon versus high fidelity vehicle models.
    - High fidelity models from VTM library.
- Implement the selected phenomenon to Open PBS.

2 High capacity transport vehicles

High capacity transport vehicles or HCT vehicles are a new approach to transport demands on road. These combination vehicles are heavier and/or longer than normal truck combinations on the roads currently. (Traficom, 2019) In Finland, HCT vehicles have been allowed on the road with special permissions from 2013 and after January 2019 Finland allowed long combinations on the roads. These long combinations can be up to 34,5 meters long and can weight up to 76 tonnes. (Ministry of transports and communications of Finland, 2019). Longer combinations can give advantages especially in container transport and transporting goods, that benefit from the increased payload capacity without increasing the maximum gross combination weight. HCT vehicles will create savings and they're eco-friendlier. With larger payloads, vehicles will obtain better energy consumption in relation to the amount of goods transported. In this way, the emissions and costs are lowered. The decrease in number of vehicles on the road will have a positive effect on traffic safety. (Traficom, 2019)

In Finland in 1.1.2019 there were 93 HCT combination vehicles on the road with special permission. Most of the vehicles have maximum weight of 76 tons. Currently there are 20 combinations that exceed 76 tons on Finnish roads. (Traficom, 2019)

HCT combinations are most attractive in situations where there is a demand to transport large volumes of either, volume- or weight limited cargo, on a predetermined section of road. This road section can be for example a travel between two warehouses, or a warehouse and a factory. (ITF, 2019, p. 24)

HCT vehicles can be used to decarbonise freight transport. Based on research by ITF, carbon reduction can be anywhere between 10-20 %, depending on a vehicle configuration. Some studies even show possible carbon reduction up to 35 % for some combination vehicles. (ITF, 2019, p. 26). Similar results have also been reported from DUO2 project, where there have been savings up to 20 % in CO_2 and in fuel consumption. (DUO2 project, 2018)

The increase in weight and length of combination vehicles will impose some challenges to the current infrastructure. There are some things to consider, when increasing the size of the combination vehicles, like keeping the axle load the same or possibly even lower than currently. HCT vehicle with similar or lower axle loads will not have large negative effect on roads compared to standard vehicles. Some studies still suggest that HCT vehicles with longer axle span are less likely to affect the road in negative way. The most severe load on the road from HCT combination is the polishing wear introduced to the road by the lateral forces of the tyres. The wear is dependent of the combination vehicles length, articulation and the geometry of the road. (ITF, 2019, pp. 32-35). The increased vehicle gross weight is of importance for bridges, since in most cases the whole vehicle will be on the bridge at the same time.

Current studies show that HCT vehicles can have an improved safety performance than standard vehicles. This is because of the development of vehicle dynamics, stricter policies with additional safety technology and better management. Also, normally drivers of HCT combinations are more experienced and most of the driving is done on large main roads. (ITF, 2019, p. 79) In figure 1 the most commonly used HCT combination vehicles are presented.

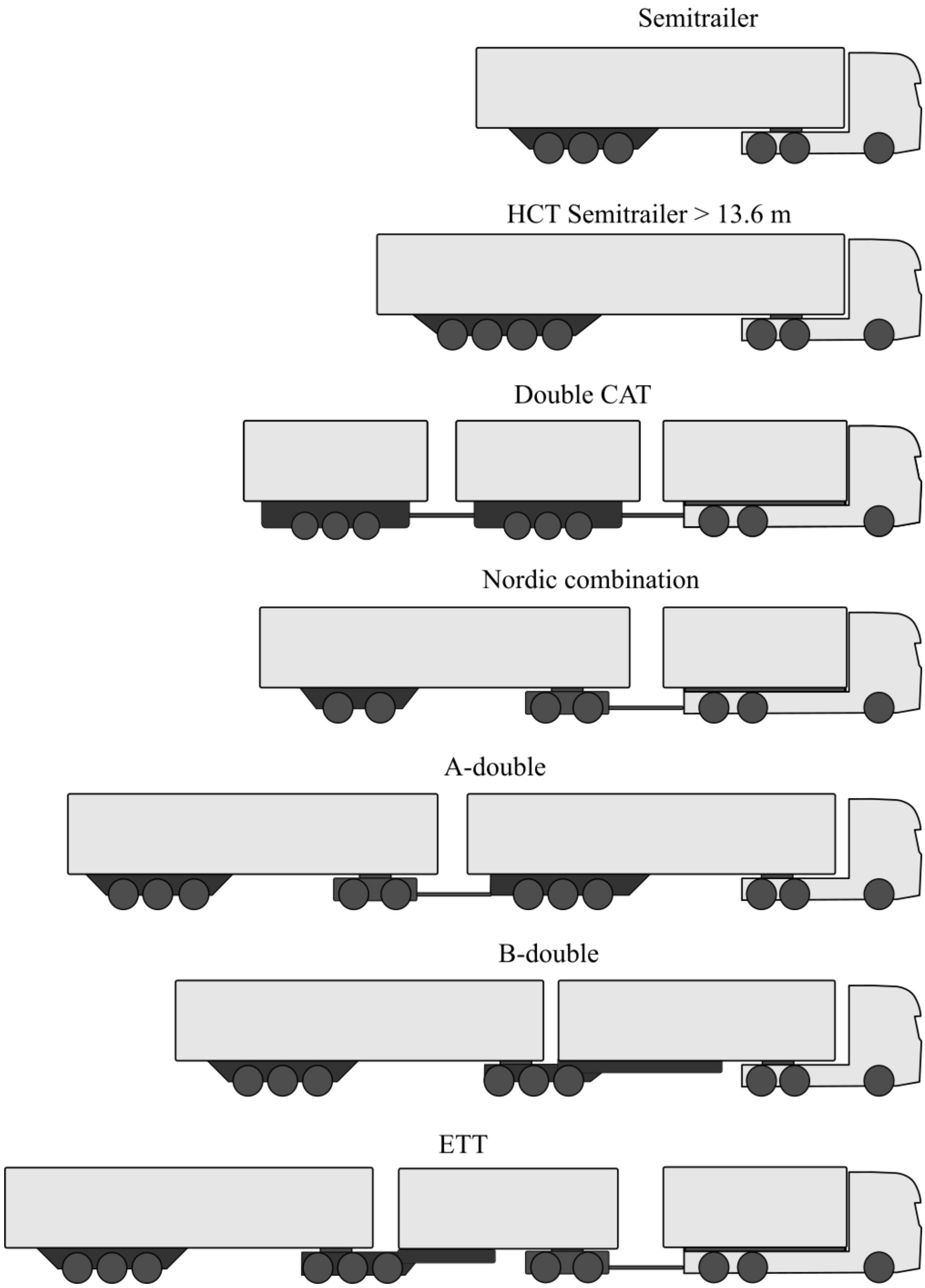

Figure 1. Mostly used HCT combination vehicles, Semitrailer combination as a reference. List from (ISO 18868:2013) updated with combinations used in research projects in Sweden and on the road in Finland.

3 Performance based standards

Performance based standards (PBS) is a new way to define vehicle restrictions. They are a set of measures designed to evaluate the vehicle stability and capabilities on road. They were first introduced in Canada in the 1980’s. The idea is that, instead of prescribing vehicle design, vehicles performance characteristics would be used to evaluate vehicle safety and their transport efficiency on the road. The reason for using PBS instead of regular prescriptive limits are its capabilities. Using performance based standards will lead to better transport efficiency and safer vehicles on the road. It will also encourage innovation, because of all the possibilities performance based standards offer. Growing freight tasks will make PBS more attractive alternative to ensure that vehicles on the road are safe and efficient. (OECD, 2005)

One of the performance based standards strong suits is its way to handle problems and jurisdictions. It describes outcomes instead of dictating how one should get to those outcomes. (OECD, 2005, p. 15)

Performance based standards can be used in multiple ways in legislation. PBS can be used as a base for prescriptive limits. For example, simulations can be used to define how vehicle performance changes with changes in vehicles mass, dimension and vehicle configuration. From there the prescriptive limits can be defined to encourage safe and economical solutions. The other way is to use PBS with the prescriptive limits. This method is already used quite widely in the world, for example in New Zealand and in Australia. In this method some performance regulations are introduced to the legislation to make the vehicle performance better. For example, in New Zealand Static Roll Threshold (SRT) is used to make sure that vehicles perform as expected. The combination vehicles that don’t fulfil these performance demands, can for example be limited to lower payloads to ensure good performance all around. The last method is to use just PBS as a limiting factor in vehicle legislation. This way just the essential vehicle limitations, like vehicle width, would be defined by the prescriptive limits. All other vehicle

parameters could be optimized to pass the performance based standards. From OECD’s perspective, this would advance sustainability, infrastructure, productivity and compliance.
(OECD, 2005, pp. 10-11)

3.1 Performance based standards around the world

Currently multiple countries are using prescriptive standards in their vehicle legislation. Prescriptive standards are made in a way that they limit road stress and vehicle dimensions. Road stress is limited with maximum gross weight and maximum axle weight. Vehicle dimensions are limited to make sure that the vehicles can run on public roads with no problems. Restricting vehicle dimension one can assure that the vehicles can for example clear intersections. Problem with these limitations are that they do not directly affect the vehicle stability. For example, vehicle height is limited, but the effect it has on vehicle stability basically comes from the centre of gravity height, which tends to change with the vehicle height. (OECD, 2005, pp. 29-32)

PBS regulations for heavy vehicles have already been implemented in Australia, Canada and New Zealand. Out of those three, Australia has made the most progress in PBS regulations. In Australia, the PBS scheme is divided into two parts: Four infrastructure standards and 16 safety standards. For each standard, four performance levels have been defined, which regulate the road network access for the heavy combination vehicles. In Canada, PBSs have been used to develop the general vehicle layout used in the road. (Kharrazi, et al., 2015, pp. 13-14)

In New Zealand, some form of PBS has been used since about 1989. The set of PBS were last reviewed in 2002 and static rollover threshold or SRT was added to the PBS. In New Zealand, the maximum length of combination vehicle is 20 meters and it has a maximum weight of 44 tonnes. In 2010 rule was amended to allow HCT vehicles to operate on routes that can support them. In New Zealand the PBS follows mostly Australian model with some additions and variations. There are added performance measures, like dynamic load transfer in single lane change and high speed steady-state off tracking at a lateral acceleration of 0.2 g. (Kharrazi, et al., 2015, p. 19)

In Canada, major research study was conducted in 1987, to identify HCT vehicles with minimal impact on infrastructure and good dynamic performance. This research led to use of seven performance based standards for evaluating vehicle performance. These include: Static rollover threshold, dynamic load transfer ratio, friction demand in tight turn, braking efficiency, low speed off tracking, high speed off tracking and transient high speed off tracking. These PBS measures were then used to create "vehicle envelopes", a set of general vehicle layouts that perform well enough. Canada's approach to HCT vehicle regulations is then a prescriptive approach based on performance based standards. (Kharrazi, et al., 2015, pp. 19-20)

Of the countries that use PBS, Australia has the most comprehensive existing PBS approach to regulation of HCT vehicles. In Australia, the process started in 1999 and the PBS scheme was taken into operation in 2007. There, using PBS is voluntary and it is used as an alternative for prescriptive regulations. Australia's approach is that it allows the use of vehicles that do not comply with the prescriptive regulations, if their performance is satisfactory in safety, manoeuvrability and effect to infrastructure. Australian PBS scheme consists of 20 different PBS values, including values like Startability, acceleration capability, frontal swing, tail swing, rearward amplification, static rollover threshold, pavement vertical loading and bridge loading. All these performance based standards have 4 different levels and vehicles performance in those standards define how large road network the vehicle can use. (Kharrazi, et al., 2015, pp. 21-23)

3.2 PBS in Nordic countries

Currently the ongoing "Performance Based Standards II" -project is working on performance based standards and their applicability in Nordic countries. (Chalmers University of technology, 2019). There are some things that need to be taken into consideration when translating performance based standards for example from Australia to Nordic circumstances. The snowy and icy road conditions provide additional challenge to the vehicles. (Kharrazi, et al., 2014)

Previously PBS has been partially used in Sweden for regulation of double combinations. It also has been used as a basis for modular combinations and granting dispensations in previous years. The performance based standards project started in Sweden in 2013 as an answer for increased interest in HCT vehicles. (Kharrazi, et al., 2015, p. 18) Currently Sweden is doing research regarding 16 different PBS measures. These measures include rearward amplification, high speed transient off tracking, front and rear swing, friction demand on drive tyres and braking stability in turn. (Jacobson, et al., 2017, p. 8)

Finland allowed long combination vehicles on roads in January 2019. In Finland, there is interest in using performance based standards to evaluate the performance of HCT combination vehicles. Currently the only PBS measure used in Finland is the braking capability of the combination vehicle. (Lahti, 2018)

In the future, possible approach determined by Traficom for combination vehicles is as follows (Lahti, 2018):

- Braking performance must be in line with the mass of the combination
- Coupling devices must withstand the forces between the vehicle units
- Axle loads cannot exceed the limits of the road network
- Combination vehicles velocity must remain quite unchanged driving uphill
- Combination vehicle must have enough traction to start moving
- Combination vehicle is capable of manoeuvring in intersections and tight spaces in its route

- Combination vehicle is stable and safe, even in sudden manoeuvres while driving

Currently, Finland uses a set of equations to evaluate the performance of HCT vehicles. These calculations include calculation of turning radius, rear swing and vehicle stability. These equations were developed using simulations as a way to create those equations. (Lahti, 2019)

3.3 Formats of PBS definitions

One part the “performance based standards” -projects here in Sweden is trying to change is the openness of the PBS measures. (Chalmers University of technology, 2019). One goal of the project is to create an open source tool, OpenPBS, that is available to anyone to use. This way all the parties involved, will have access to all the same information, insight and equations. The parties involved are the authority, transport operators and vehicle manufacturers.

Similar steps have also been taken in Finland, where simple spreadsheet calculations, instead of vehicle simulations, could be used to evaluate the lateral stability of combination vehicles. This approach has been taken to ensure that all the parties involved can evaluate the possible combination vehicles in the future, and cost will not be an issue for transportation companies. In this approach, main dimensioning parameters, such as trailer wheelbase, coupling locations and drawbar length are used to evaluate the lateral stability of a combination vehicle. (Tuutijärvi, et al., 2019)

These two approaches differ significantly of the approach used in Australia for example. Even though the PBS-scheme in Australia is the most comprehensive one, there the evaluation of the vehicle stability and safety is done by PBS Assessors that run the simulations regarding the vehicle safety. Thus, transport companies must contact those PBS assessors to see if their vehicle concept is applicable. (NHVR, 2019) The certified PBS assessors will use their own experience and knowledge to evaluate the combination vehicles by testing, numerical modelling or with calculations. This way the information is not widely available, and each combination vehicle will be tested/simulated differently. (NHVR, 2017)

4 Modelica and FMI

Modelica is a language used to model complex physical systems that are for example mechanical, hydraulic or electric. The language is non-proprietary, object oriented, and it's based on equations. Open source Modelica libraries currently have about 1600 model components and 1350 functions. Modelica simulation environments are widely available commercially and free of charge. Widely used commercial software using Modelica language is Dymola. Most of this thesis is on the other hand done using a free software called OpenModelica. Industry uses Modelica language and Modelica libraries in a model-based development. Modelica is in use especially in many automotive companies, such as Audi, BMW and Toyota. (Modelica Association 1, 2019)

Functional Mock-up Interface, or FMI is a standard that supports model exchange and co-simulation of dynamic models. FMI uses a combination of xml-files and compiled C-code. FMI development was initiated by Daimler AG and first version of FMI was released in 2010. The goal for FMI was to improve the exchange of simulation models between suppliers and OEMs. Currently the development continues, with participants out of 16 companies and research institutes. Development is run by Modelica Association and currently FMI is supported by over 100 tools. It's used in automotive and non-automotive applications throughout Europe, Asia and North America. (Modelica Association 2, 2019)

4.1 OpenModelica

OpenModelica is an open-source Modelica based modelling and simulation environment. It's intended for industrial and academic usage. Its development is supported by a non-profit organization, the Open Source Modelica Consortium (OSMC). From OpenModelica webpage: "*The goal with the OpenModelica effort is to create a comprehensive Open Source Modelica modelling, compilation and simulation environment based on free software distributed in binary and source code form for research, teaching, and industrial usage. We invite researchers and* *students, or any interested developer to participate in the project and cooperate around OpenModelica, tools, and applications."* (OpenModelica, 2019)

Because the goals of this thesis, Modelica language and OpenModelica simulation software were selected to create simulation program, that is available for everyone as open source. Using open-source software and creating an open source simulation package gives a lot of possibilities, as the program is widely available. This way of working also gives for example a state of transparency for this project.

4.2 OpenPBS

The goal for the OpenPBS tool is to be able to evaluate and simulate different PBSs for different vehicles or combination vehicles. It's capable of doing virtual verification of PBSs. Swedish transport agency has worked on a web-application that is used for approval of individual combination vehicles. In the future, this OpenPBS tool could maybe be used as a part of this web application. (Jacobson, et al., 2017, p. 7)

The OpenPBS is an open assessment tool and it is based on standard formats for dynamic models. The OpenPBS tool uses the simplest possible vehicle models to keep the required parameter amount as low as possible, still while giving out results that represent the vehicle behaviour in relevant accuracy. The PBS measures/manoeuvre models also must be as simple as possible to keep the number of needed parameters low and to keep the models understandable. (Jacobson, et al., 2017, pp. 7-9)

OpenPBS is built so that in the package all simulation parameters can be changed individually. This means that **vehicle model** is individual from **vehicle definition** and vehicle definition is individual from **PBS measures/manoeuvres.** This way all parts of the simulation model can be changed and improved in the future. (Jacobson, et al., 2017, p. 9) The OpenPBS structure is presented in figure 2.

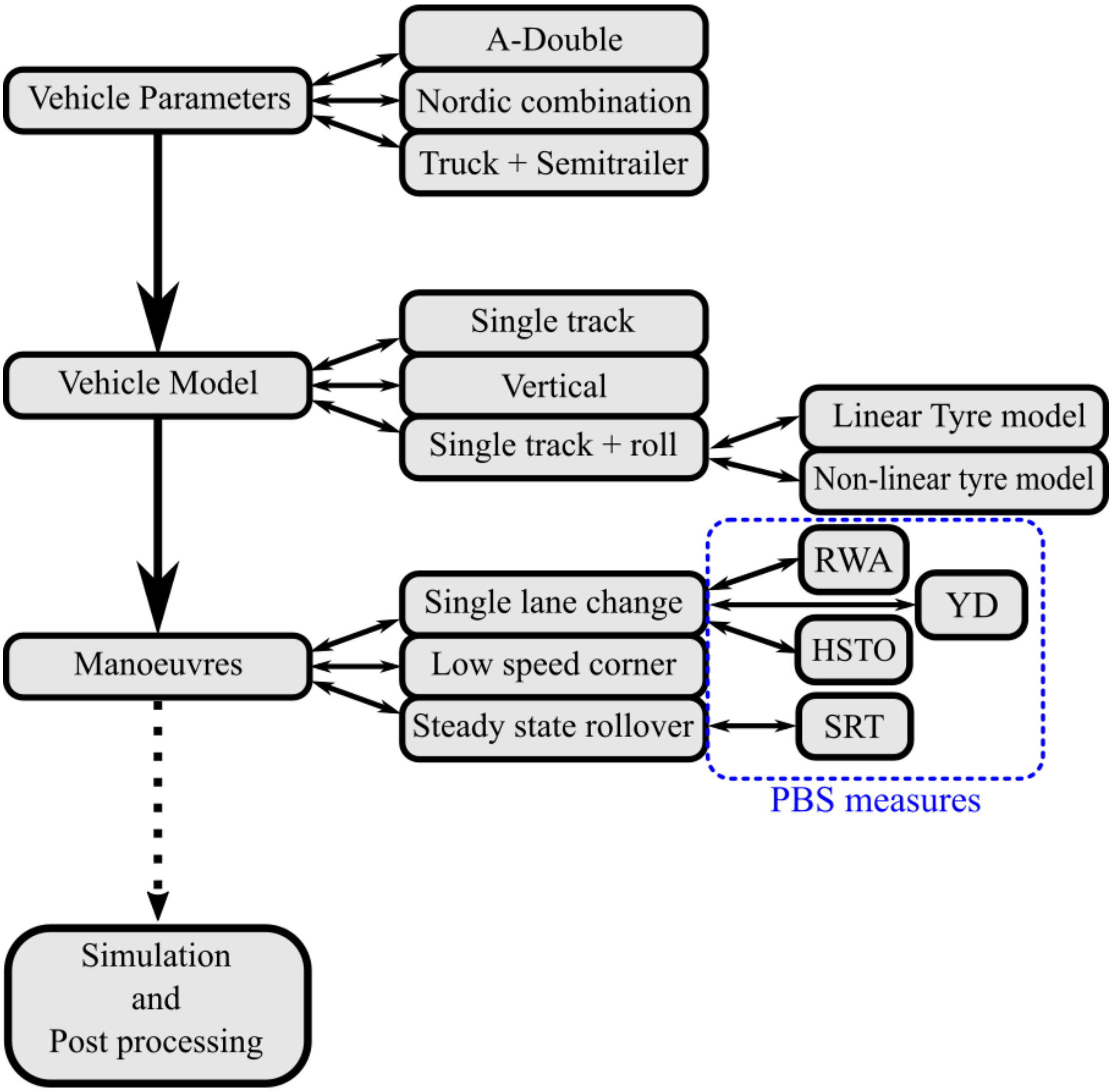

Figure 2. Overview of how OpenPBS is structured.

The basic structure of OpenPBS consists of Vehicle parameters, Vehicle models and Manoeuvres. Vehicle parameters contains parameters for different vehicle types. For example, parameters for A-double contain information of vehicle mass, its dimensions and inertias. Vehicle models describe different vehicle models, that use vehicle parameters defined previously. For example, Single track vehicle model describes a full single-track model. This model is then instantiated in the Manoeuvres, where the different PBS manoeuvres are described. This construction of simulation model allows the user to add vehicles, redefine the simulation models or add different manoeuvres without affecting each other.

5 Vehicle dynamics

Normally vehicle handling is described as vehicle's reaction to steering input and vehicle's reaction to forces affecting on the vehicle body, like wind or input from the road. These forces and inputs try to affect on the vehicle's direction of motion. Two basic issues regarding vehicle handling are the vehicle's capability to follow steering input and the vehicle's reaction to outside forces. (Wong, 2001, p. 335).

A single-unit vehicle has only one body which has six degrees of freedom around its centre of gravity. These degrees of freedom or d.o.f are the translational movements in x, y and z -axis and the roll around these axes. These translational d.o.f can be described as the vehicle longitudinal, lateral and vertical movements. As for rotational d.o.f these can be described as Yaw, Roll and Pitch. (Wong, 2001, p. 335) Vehicle axis system is presented in figure 3.

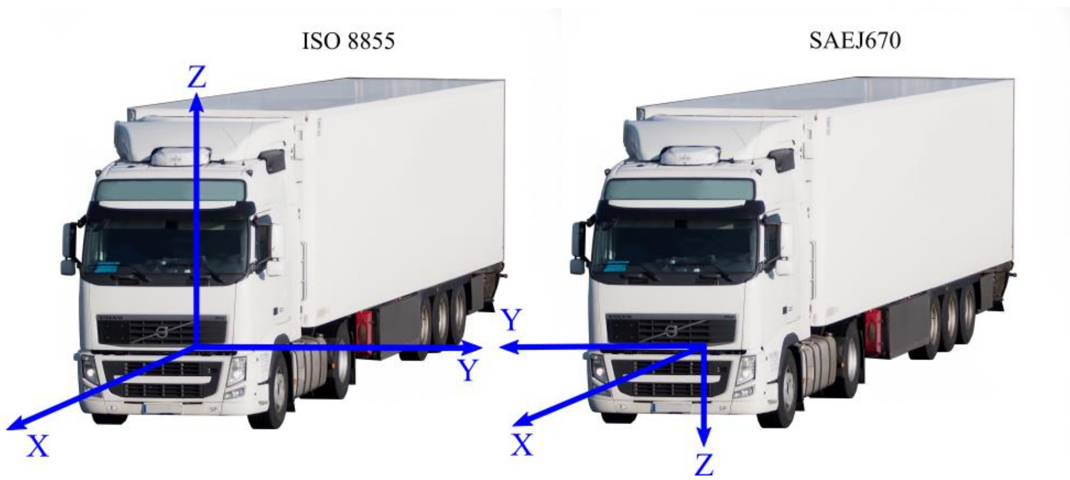

Figure 3. Vehicle axis system ISO 8855 and SAEJ670. ISO-standard used in this thesis. Figure is based on (Wikimedia Commons, 2016).

Time between steering input and the steady state motion after that is called transient state. In this state the vehicles response to transient situation is studied. Vehicles handling characteristics are dependent on this transient behaviour. At its best, the vehicle reacts fast to steering input with minimal oscillation while approaching the steady state motion. In transient state vehicle's inertia properties must be taken into consideration, because the vehicle moves both in translation and in rotation. (Wong, 2001, p. 359).

In this thesis the roll dynamics and lateral dynamics are emphasized. Vehicle's lateral dynamics describes mainly the vehicle's motions that effect the vehicle's dynamic stability, cornering and road holding. (Heißing & Ersoy, 2011, p. 35). Whereas, roll dynamics effects to vehicle's vertical and lateral dynamics. Forces created through lateral acceleration in the bodies will affect the vehicle dynamics through the springs, dampers and sway bars. (Heißing & Ersoy, 2011, pp. 67,77-78)

A combination vehicle consists of several units, so the number of degrees becomes more than 6. For each unit that is added with an articulation coupling, there is at least one in-road-plane d.o.f added, the yaw rotation. Combination vehicle's stability can be evaluated with multiple different values. These values are for example rearward amplification, yaw damping, lateral load transfer and high speed transient off tracking.

Rearward amplification describes the amplification of vehicle's last unit's movement compared to the first unit. When the rearward amplification is larger than one, the movement from the truck amplifies to the last unit. Rearward amplification can be calculated for each unit behind the towing vehicle, but normally the rearward amplification is calculated from the difference between first and the last unit of the combination vehicle. Rearward amplification can be calculated either from lateral acceleration in some point at each of the vehicle units or from the yaw rate of the vehicle units. Rearward amplification is calculated from the maximum values of the selected variable during for example a single lane change. (ISO 14791: 2000 (E)). The rearward amplification describes how unstable the combination vehicle is in a lane change. Livelier the vehicle, the more prone the combination vehicle is to deviate laterally from lane or even to rollover. Typical maximum acceptable value for rearward amplification is 2.0. (ITF, 2019). Example of rearward amplifications definition in figure 4.

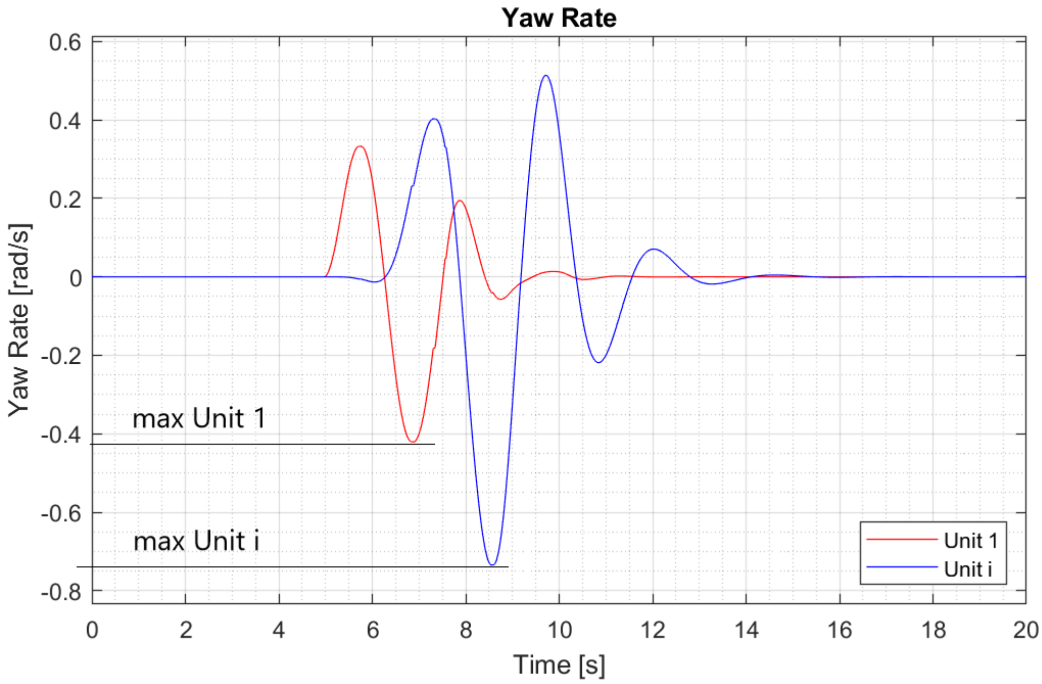

Figure 4. Yaw rate values used for definition of rearward amplification.

Rearward amplification from yaw rate can be defined as follows:

$$RWA = max_i\left(\frac{max_t(|\omega_{z,i}(t)|)}{max_t(|\omega_{z,1}(t)|)}\right) \tag{1}$$

Where RWA is the rearward amplification, $\omega_{z,i}$ is the yaw rate for the desired unit behind the first unit and $\omega_{z,1}$ is the yaw rate for the first unit.

Yaw damping describes how fast the lateral sway, introduced to the combination vehicle by steer input, decays over time. Calculation of yaw damping can be done from either yaw rate, articulation angle or articulation angular velocity. The minimum value for acceptable level of damping is 0.15. The base for yaw damping calculations is presented in figure 5 (ITF, 2019). If YD is less than 0 the swaying increases, when YD is 0 there is no damping and when YD is greater than 0 the swaying decays over time. Yaw damping can be defined as follows:

$$YD = \frac{\ln \frac{|x_1|}{|x_2|}}{\sqrt{4\pi^2 + \left(\ln \frac{|x_1|}{|x_2|}\right)^2}} \tag{2}$$

Where x_1 is the first measurement point and x_2 is the second measurement point. For example, when using articulation angle to calculate the yaw damping, x_1 would be the first articulation angle and x_2 would be an articulation angle after selected number of peaks in the articulation angle. The first selected peak can be any of the peaks that happen after the manoeuvre, when the vehicle units are swaying freely.

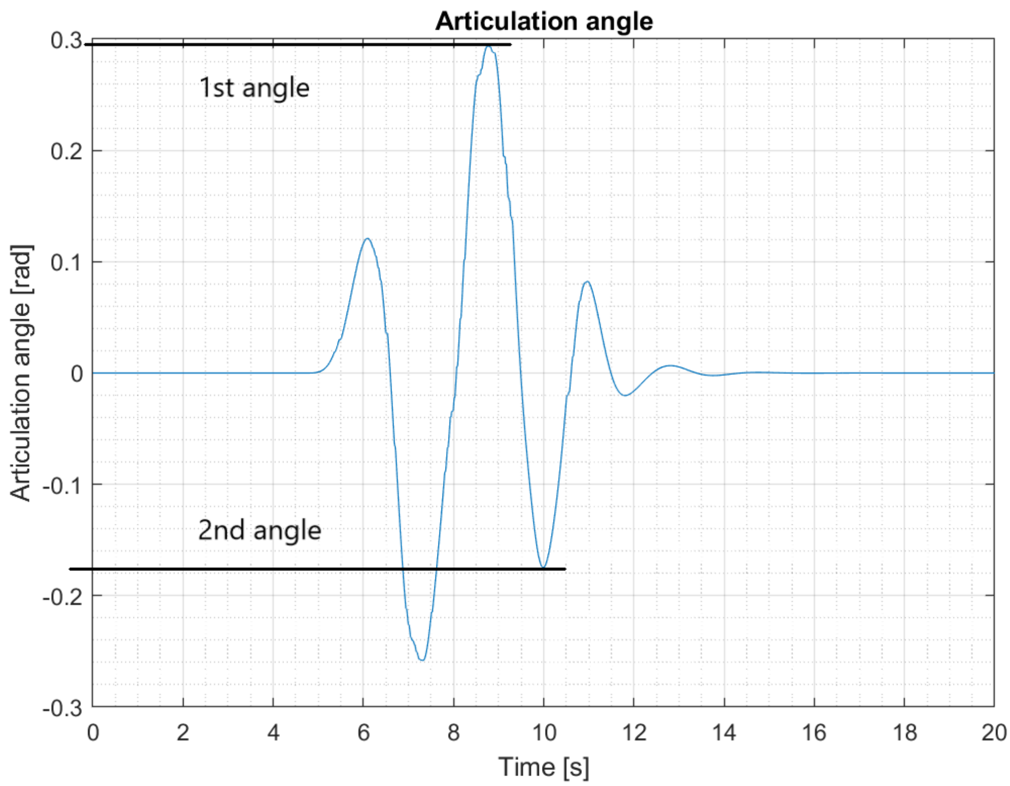

Figure 5. Articulation angle used for definition of the yaw damping. First selected peak is the first peak after the manoeuvre.

Lateral load transfer describes the change in load on vehicles either side during a transient manoeuvre. The value describes unit's rollover stability. Lateral load transfer is calculated in a single lane change manoeuvre. Lateral load transfer or LLT can be defined as follows:

$$LLT = max_i \left(max_t \left(\frac{F_{zl} - F_{zr}}{F_{zl} + F_{zr}} \right) \right) \quad (3)$$

Where *LLT* is the lateral load transfer, F_{zl} is the load on the left side wheels and F_{zr} is the load on the right-side wheels.

When the lateral load transfer gets a value of 1, tyres in the other side of the vehicle lift off from the ground. Typical limit for the lateral load transfer is 0.6. (ITF, 2019) Values used for calculating the lateral load transfer are described in the figure 6.

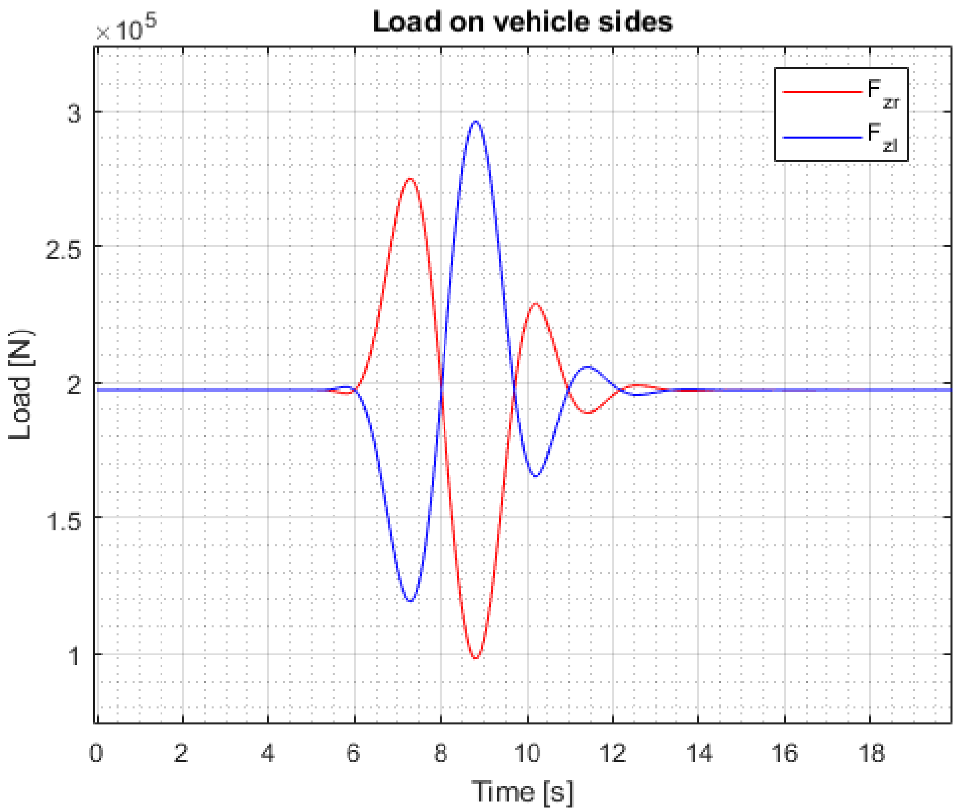

Figure 6. Load on sides of vehicle unit. Used for calculation of lateral load transfer.

Last key value used for analysing combination vehicles stability is the high speed transient off tracking. It's an important value that defines, if the combination vehicle stays on the road during a single lane change manoeuvre. In Canada, reference value for the high-speed transient off tracking is 0.8 meters, and in Australia it varies from 0.6 to 0.8 based on the road conditions. (ITF, 2019) High speed transient off tracking is defined as a difference in the trajectories of the first and the last axle of the combination vehicle. In single lane change, HSTO is defined as follows

$$HSTO = \max\,(position_{last}) - \max\,(position_{first}) \tag{4}$$

Where $position_{last}$ and $position_{first}$ describe the y-coordinate of the vehicle axles

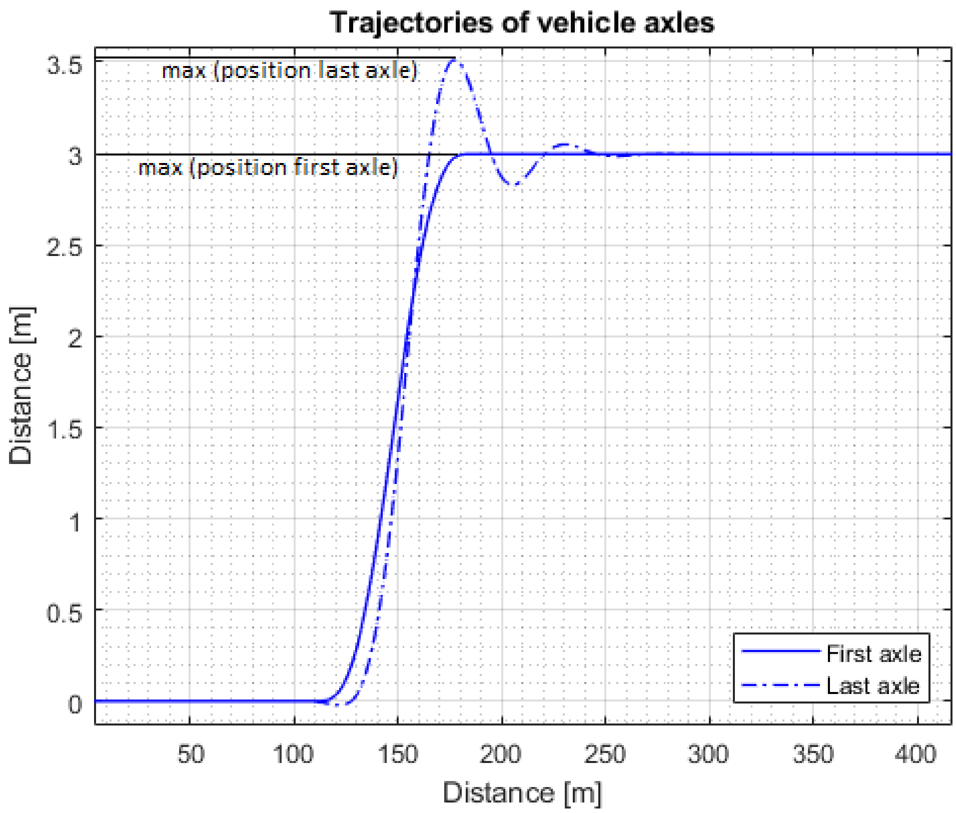

Figure 7. Axle trajectories in a single lane change, for calculating high speed transient off tracking of combination vehicle.

These key values will be used for evaluating the simulation models and combination vehicle's stability.

5.1 Lateral dynamics and single-track model

Vehicle's lateral dynamics describes mainly the vehicle's motions that effect the vehicle's dynamic stability, cornering and road holding. These motions have a significant role nowadays regarding the development of driver assistance systems and vehicle dynamics control systems. (Heißing & Ersoy, 2011, p. 35)

In a single-track model, two track vehicle is assumed to be one track. Instead of two tires per axles, in single track model the vehicle is simplified, so that it only has one virtual tire per axle. Single track model is a simple way to describe vehicles lateral dynamics. The simple single-track or "bicycle" model was first introduced in 1940. The model is still used in many applications due to its simplicity. In the single-track model, there are multiple important omissions and simplifications. With these simplifications the model's degrees of freedom are greatly reduced and thus, make the simulation of these models much easier. The single-track model is still capable of capturing the vehicle's dynamic behaviour in linear range. (Heißing & Ersoy, 2011, p. 89)

In the original single-track model, some critical simplifications were made. These simplifications are (Heißing & Ersoy, 2011, p. 89):

- Height of the vehicle's centre of gravity is assumed to be at the level of road surface. This way, vertical tire forces at inner and outer wheels remain the same during cornering.
- The equations in single track model are linearized. This is true for both angle functions and tire behaviour.

The single-track model's idea is presented in figure 8.

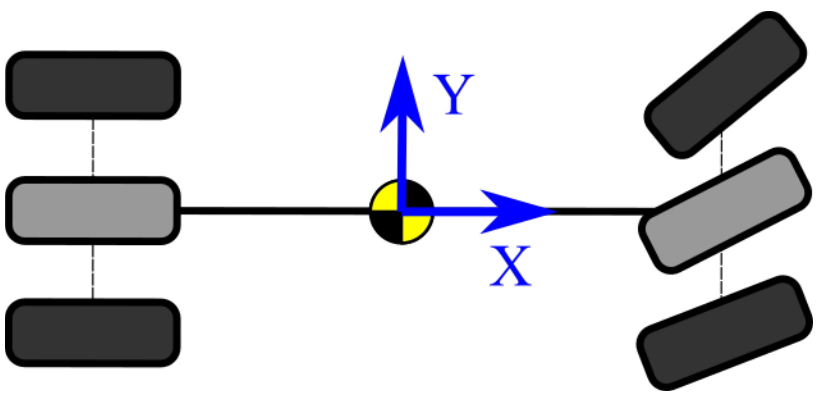

Figure 8. Base of single track model. Two tires in single axle are combined to a single virtual tyre per axle.

In this thesis, most of the simulation is done using single track models. The base for thesis' single track model comes from the conference paper "Vectorized single-track model in Modelica for articulated vehicles with arbitrary number of units and axles" by Sundströn, Jacobson and Laine (Sundström, et al., 2014).

Sundström's model is vectorized and parametrized so that the model has the capability to simulate different vehicles, with just a change in vehicle parameters. The change of vehicle does not create any problems for the simulation model. With vectorized and parametrized model you can simulate for example truck-semitrailer combination, Nordic combination and a A-double combination back to back with the same simulation program. The parametrization principle of single-track model is described in figure 9.

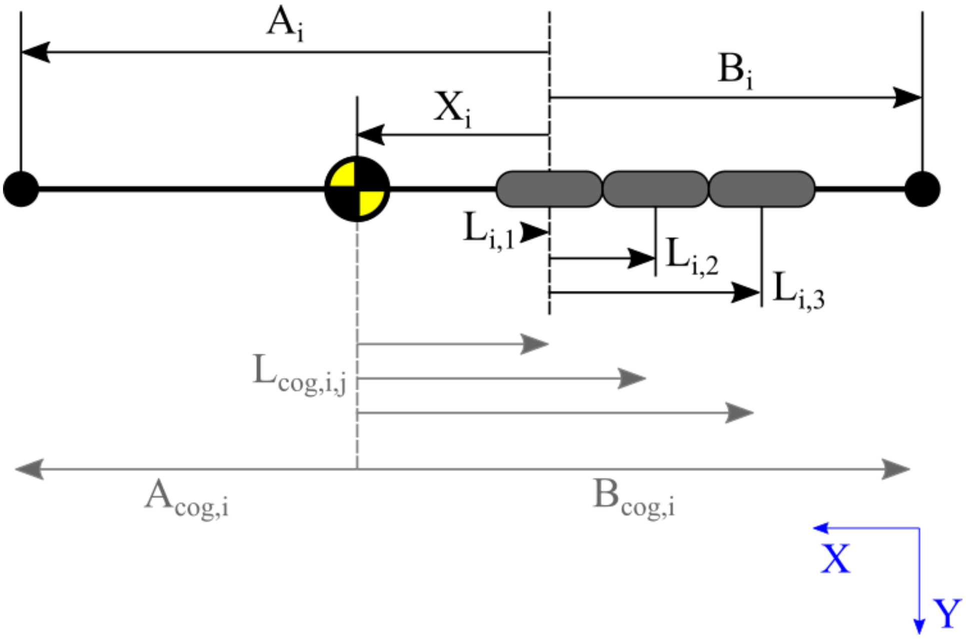

Figure 9. Parametrization of single track model. Figure is based on (Sundström, et al., 2014).

In Sundström's model, unit's first axle is used as a "zero" point for all the measurements. In the figure, *i* represents unit's number. From the unit's first axle we can take the following measurements. $L_{i,2}$ and $L_{i,3}$ are distances from the first axle to following axles. A_i and B_i represent distances from the first axle to unit's coupling points. X_i is the distance from the first axle to the centre of gravity. These measurements are used to calculate the other distances, that represent the distance from the unit's centre of gravity. $L_{cog,i,j}$ is the axle distance from centre of gravity. $A_{cog,i}$ and $B_{cog,i}$ are the coupling distances from the centre of gravity. In this representation, coordinate system is presented on the right lower corner of the figure 9. So, measurements going to the left are positive and measurements going to the right are negative.

The following equations describe the single track model used in OpenPBS and created by Sundstöm. It consists of equations that describe slip angle, tyre forces, coupling restrictions and longitudinal, lateral and yaw equilibriums. In this parametrized model the tire forces can be defined as follows (Sundström, et al., 2014):

Slip angle relation to vehicle motion and steering angle

$$\alpha = \frac{v_y + L_{cog} \cdot \omega_z}{v_x} - \delta \tag{5}$$

Where L_{cog} is the distance from centre of gravity to each axle, v_y and ω_z are the lateral velocity and yaw rate of the vehicle units. The δ represents axles steering angle.

Lateral tire force is defined as follows,

$$F_{yw} = -C_{st} \cdot \tan(\alpha) \rightarrow for\ small\ angles{:}\ F_{yw} = -C_{st} \cdot\ \alpha \tag{6}$$

Where C_{st} is the axle cornering stiffness.

Tire forces can be transformed to the vehicle coordinate system with the steering angle δ as follows,

$$F_x = F_{xw} \cdot \cos(\delta) - F_{yw} \cdot \sin(\delta) \tag{7}$$

$$F_y = F_{xw} \cdot \sin(\delta) + F_{yw} \cdot \cos(\delta) \tag{8}$$

Where F_x is the longitudinal force on the vehicles coordinate system, F_y is the lateral force on the vehicles coordinate system and F_{xw} is axles longitudinal force.

The longitudinal force in the axles in this model is defined by an *if*-function. It gives us information of the needed drive force to maintain the input velocity. So, if the axle is driven axle, the force on the axle is F_{xd}, otherwise it's 0.

For the articulated vehicle model to work, some coupling constraints must be added to the simulation model. Free body diagram of the coupling is presented in the figure 10.

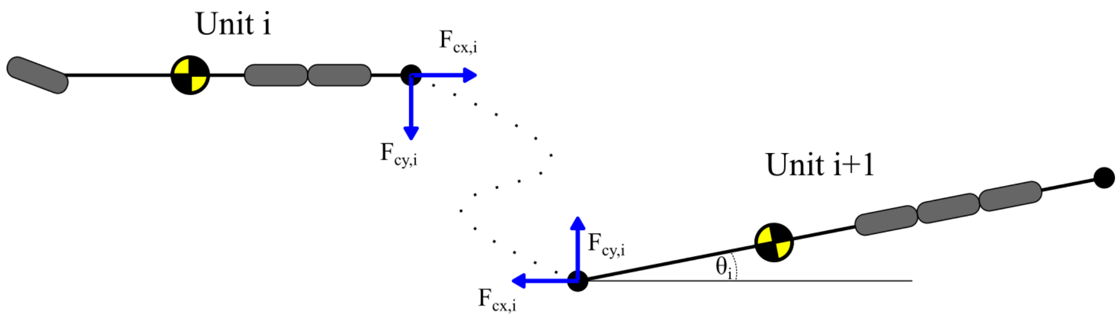

Figure 10. Coupling restrictions in Sundströms vectorized single track model. Figure based on (Sundström, et al., 2014).

For coupling the velocity restriction can be described as

$$v_x[i+1] = v_x[i] \cdot \cos(\theta[i]) - \left(v_y[i] + B_{cog}[i] \cdot \omega_z[i]\right) \cdot \sin(\theta[i]) \tag{9}$$

$$v_y[i+1] + A_{cog}[i+1] \cdot \omega_z[i+1] = \left(v_y[i] + B_{cog}[i] \cdot \omega_z[i]\right) \cdot \cos(\theta[i]) + v_x[i] + \cdot \sin(\theta[i]) \tag{10}$$

Where v_x is the longitudinal velocity of unit, i is units' number and θ is the angle between the units.

For the single-track model to be functional, some other equations must also be determined. These equations are the vehicle equilibriums in yaw plane. (Volvo Group Trucks Technology, 2001)

Longitudinal equilibrium for the single-track model is

$$m \cdot \left(\dot{v_x} - v_y \cdot \omega_z\right) = F_x + F_{cx} \tag{11}$$

Where F_{cx} is the couplings longitudinal force, which is dependable upon the unit and the angle between the two units.

Lateral equilibrium is described as

$$m \cdot \left(\dot{v_y} + v_x \cdot \omega_z\right) = F_y + F_{cy} \tag{12}$$

Where F_{cy} is the couplings lateral force, which is dependable upon the unit and the angle between the two units.

Yaw equilibrium can be described

$$I_z \cdot \dot{\omega_z} = L_{cog} \cdot F_y \pm L_{C-cog} \cdot F_{cy} \tag{13}$$

Where I_z is the unit's inertia in yaw plane, L_{cog} is the distance from axles to centre of gravity and $L_{C\text{-}cog}$ is the distance from the coupling to the centre of gravity

In figure 11 the single-track model for semitrailer combination is presented.

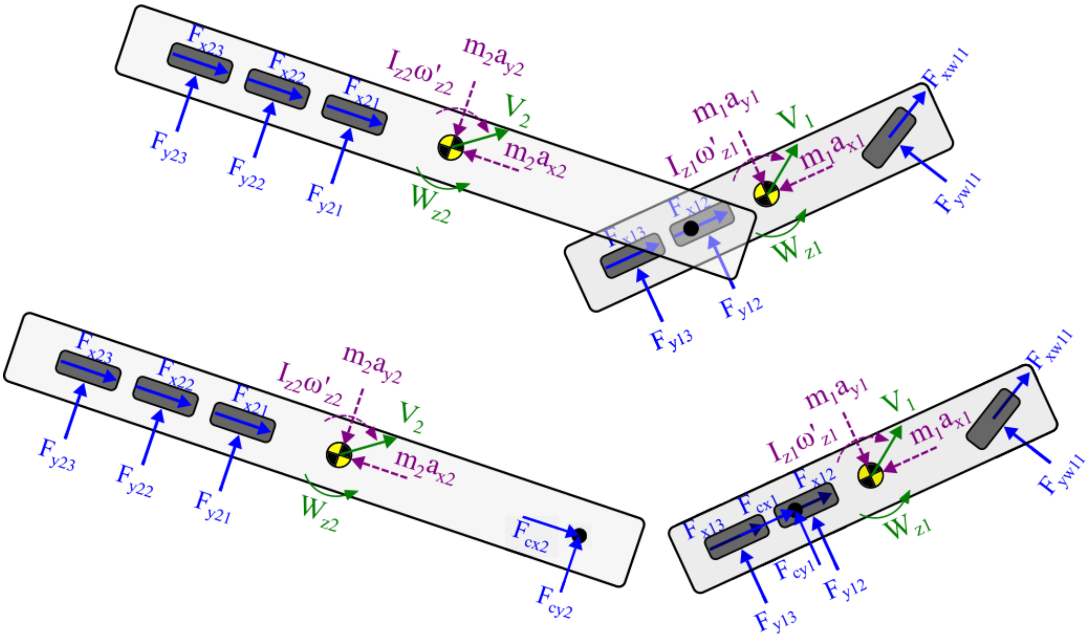

Figure 11. Single track model of truck-semitrailer combination vehicle.

The calculation of the equations in this chapter for tractor-semitrailer combination vehicle is presented in the appendix 2.

5.2 Vertical Dynamics

Vertical dynamics describes the vehicle's vertical movement. Vertical dynamics is usually used with tuning springs and dampers. The goal of vertical dynamics is minimizing the vertical acceleration of the vehicle's body. This will provide a better ride and increased comfort. In addition to those, it will also reduce load change at tyres and thus, improve safety. (Heißing & Ersoy, 2011, p. 35)

In this thesis some simple calculations are done to calculate the static tyre loads, which are then used to calculate axle cornering stiffnesses. The calculations are done as they are in the current version of OpenPBS. The free body diagram for the vertical dynamics calculations is presented in figure 12.

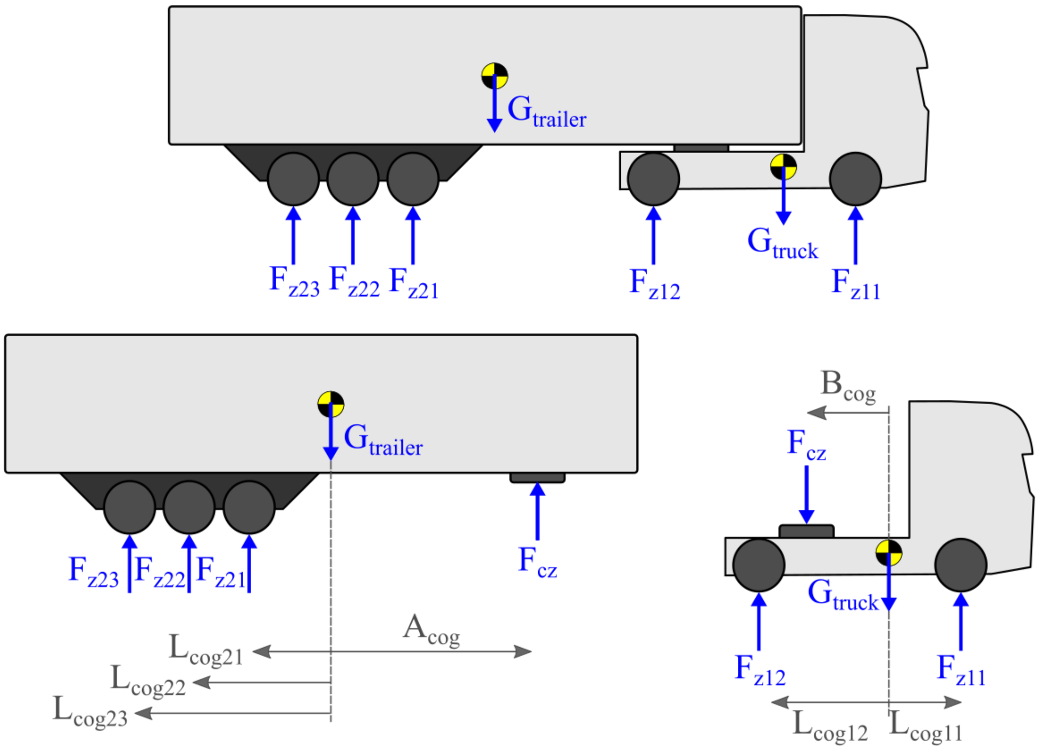

Figure 12. Free body diagram of tractor-semitrailer combination vehicle for static axle load calculations.

Calculation of static axle loads in vectorized model can be done as follows.

$$m \cdot g = F_z - F_{cz}\ (for\ the\ first\ unit) \tag{14}$$

$$0 = L_{cog} \cdot F_z - B_{cog} \cdot F_{cz} \tag{15}$$

$$C_{st} = Cc \cdot F_z \tag{16}$$

Where F_z is the axle's vertical force, F_{cz} is the vertical force on the coupling, Cc is the cornering coefficient

The vectorized presentation of the static load calculations for tractor-semitrailer combination vehicle is presented in the appendix 3.

5.3 Roll Dynamics

Roll dynamics effects to vehicle's vertical dynamics. Forces created through lateral acceleration in the bodies will affect the vehicle dynamics through the springs, dampers, sway bars and the non-linear force generation of the tyre. (Heißing & Ersoy, 2011, pp. 67,77-78)

Roll dynamics have a huge effect on vehicle stability in higher speeds, especially with vehicles with high centre of gravity. The model for roll dynamics in this thesis is based on the roll dynamics model in Jacobson's Vehicle dynamics compendium (Jacobson, 2016). Truck roll model is presented in figure 13.

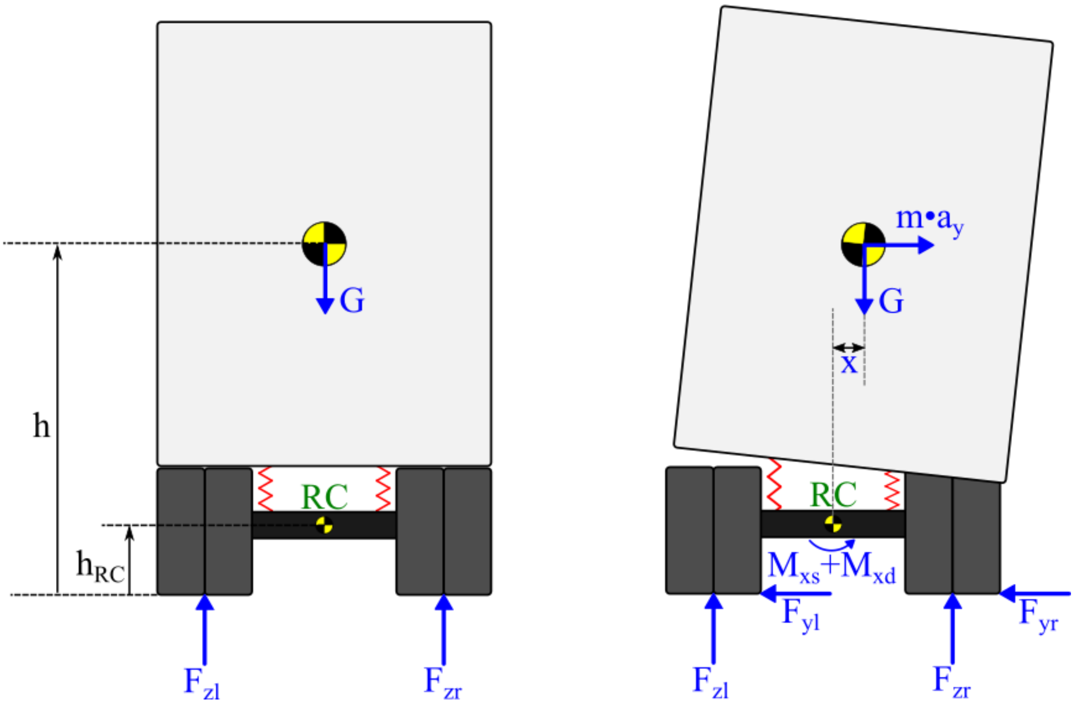

Figure 13. Description of truck roll model.

Adding roll dynamics to the simulation model adds some complexity to the model. Adding a roll plane will give more competent results in simulation. To add roll dynamics to the simulation model, you need to add equilibrium for unsprung mass, difference between lateral velocity in unsprung and sprung parts and constitution for suspension. Using Jacobson's paper as a base, the following equations can be added to the simulation model. (Jacobson, 2016)

Equilibrium of unsprung mass

$$M_x + F_y \cdot h_{RC} = M_{xs} + M_{xd} \tag{17}$$

Where M_x is the roll moment in roll centre, h_{RC} is the height of roll centre, M_{xs} is the torque created by springs and M_{xd} is the torque created by dampers.

Lateral velocity

$$v_{yu} = v_{ys} + (h - h_{RC}) \cdot \omega_x \tag{18}$$

Where v_{yu} is the lateral velocity of unsprung mass, v_{ys} is the lateral velocity of sprung mass, h is the height of units centre of gravity and ω_x is the roll rate of the vehicle.

Equilibrium for the whole vehicle in roll plane

$$I_{xs} \cdot \dot{\omega}_x = M_x + F_y \cdot h + m \cdot g \cdot (h - h_{RC}) \cdot p_x \tag{19}$$

Where I_{xs} is the inertia of unit in roll plane, m is the mass of unit, g is fall acceleration on earth, p_x is the roll angle of the unit.

Equations for torques in suspension

$$\dot{M}_{xs} = -c_{roll} \cdot \omega_x \tag{20}$$

$$M_{xd} = -d_{roll} \cdot \omega_x \tag{21}$$

Where c_{roll} is the springs roll stiffness and d_{roll} is the roll stiffness of the dampers.

Vehicle roll also must be accounted for in the coupling's lateral speed

$$v_{yc} = v_{ys} + (h - h_C) \cdot \omega_x \tag{22}$$

Where v_{yc} is the lateral speed of coupling and h_C is the height of the coupling.

The calculation for tractor-semitrailer combination vehicle presented in appendix 4.

5.4 Tyre

Vehicle handling and response to different inputs is greatly influenced by the mechanical force and the moment generating characteristics of tyres. This is because the tyres are the only contact between the vehicle and the road. The contact patch with road and the vehicle is relatively small and thus requires a lot from the tyre to produce all the needed forces. (Blundell & Harty, 2004, p. 248) The tyre is a force and a moment generating structure, that has a function to keep the vehicle on the road. Tyre has also to work as a dampener for vibrations. Primary requirement for the tyre is to produce force in all tyre directions (Longitudinal F_x, Lateral F_y and Vertical F_z). (Pacejka, 2012, pp. 59-60). Longitudinal forces are used to accelerate and decelerate the vehicle, Lateral forces are used to steer the vehicle and keep the vehicle on the road. Tyre's vertical forces are used to carry out the load of the vehicle. (Gillespie, 1992, p. 335). Tyre's coordinate system is presented in figure 14.

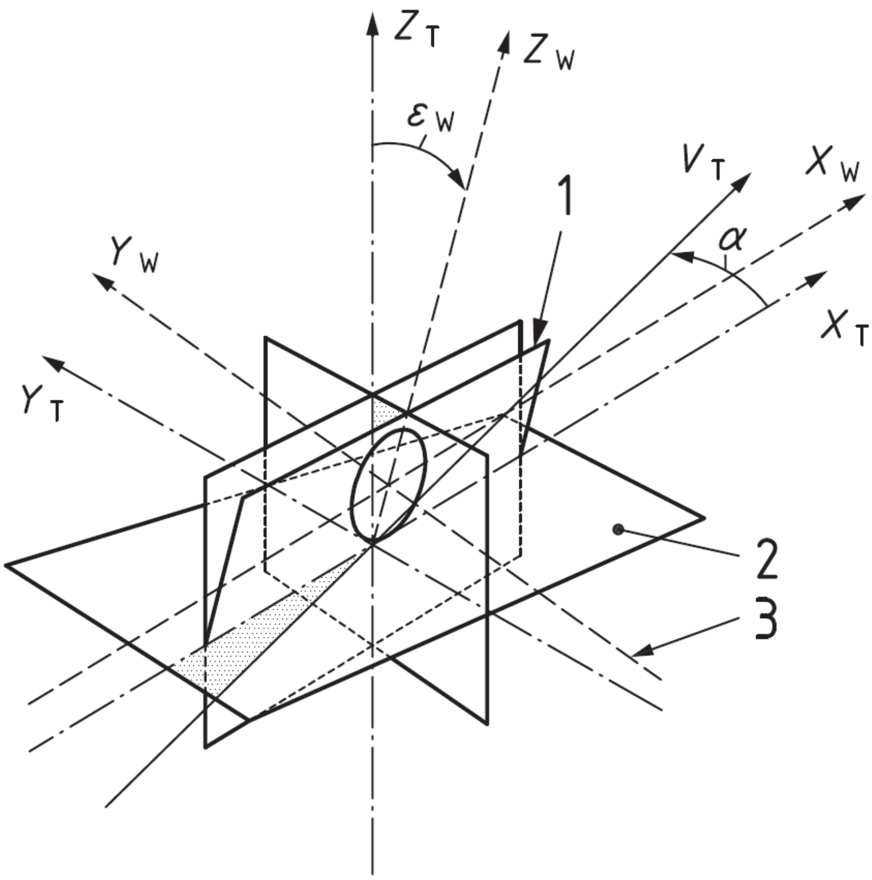

Figure 14. Tyre coordinate system (ISO 8855:2011(IDT)). Figure from standard published with permission from Finnish Standards Association SFS ry.

In figure 14 number 1 is the wheel plane, number 2 is the road plane and number 3 is the wheel-spin axis. X_W, Y_W and Z_W represent the wheel axis system. Axes X_W and Z_W are parallel to the wheel plane. Y_W axis is parallel to the wheel-spin axis, X_W, axis is parallel to road plane and the positive Z_W axis points upwards. Wheel axis system's origin is in the wheel centre. X_T, Y_T and Z_T represent the tyre axis system. In this system, the X_T and Y_T axes are parallel to the road plane and the Z_T axis is a normal to the road plane. The tyre axis system's origin is in the contact centre of the tyre. ε_W is the wheel camber angle and α represents the tyre slip angle. (ISO 8855:2011(IDT)).

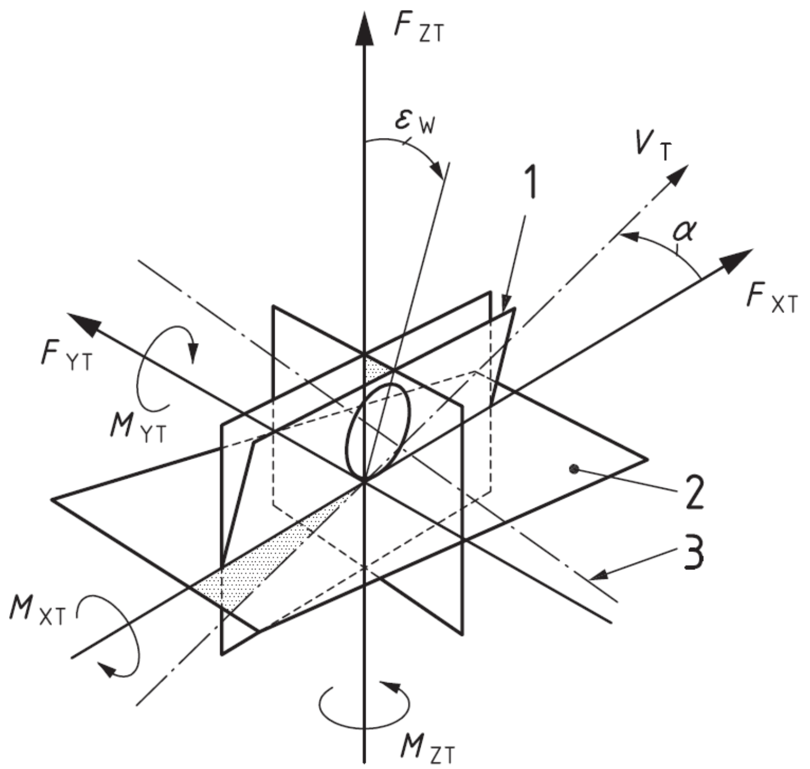

Figure 15. Forces from tyre (ISO 8855:2011(IDT)) Figure from standard published with permission from Finnish Standards Association SFS ry.

In figure 15 is presented the tyre forces in ISO coordinate system. In this figure F_{YT} is the tyre lateral force, F_{XT} is the tyre longitudinal force and F_{ZT} is the tyre vertical force. M_{YT} describes the tyres rolling resistance moment, M_{XT} is the tyre overturning moment and M_{ZT} is the tyres self-aligning moment. (ISO 8855:2011(IDT))

Tyres lateral and longitudinal force components are dependent of the tyre's vertical load. Each force component is created with multiple mechanisms working together. For example, longitudinal force reacts to driving, braking and rolling resistance. The lateral force is dependent on slip angle and camber angle. (Blundell & Harty, 2004, pp. 257-258).

5.4.1 Tyre Model

Tyre model is always a compromise between accuracy and the complexity. Tyre model needs to be developed to serve specific application of the model. For example, for ride and vibration studies the tyre model must be different compared to suspension loading and durability studies. In vehicle handling studies the function of the tyre model is to establish the forces and the moments in the road to tyre contact. The tyre model will create forces and apply them at each wheel centre and this way control the motion of the vehicle. (Blundell & Harty, 2004, pp. 291-294)

Tyre model can be modelled with desired degree of accuracy. The tyre lateral tyre force can be calculated to be completely linear. In this case the lateral tyre force will follow the following equation

$$F_y = -C_{st} \cdot \alpha \tag{23}$$

Where Fy is tyre's lateral force, C_{st} is the cornering stiffness and α is the tyre's slip angle.

Linear tyre model used previously in openPBS is only dependent upon slip angle of the tyre and static vertical load on the axle. Tyre force as a function of slip angle presented in figure 16.

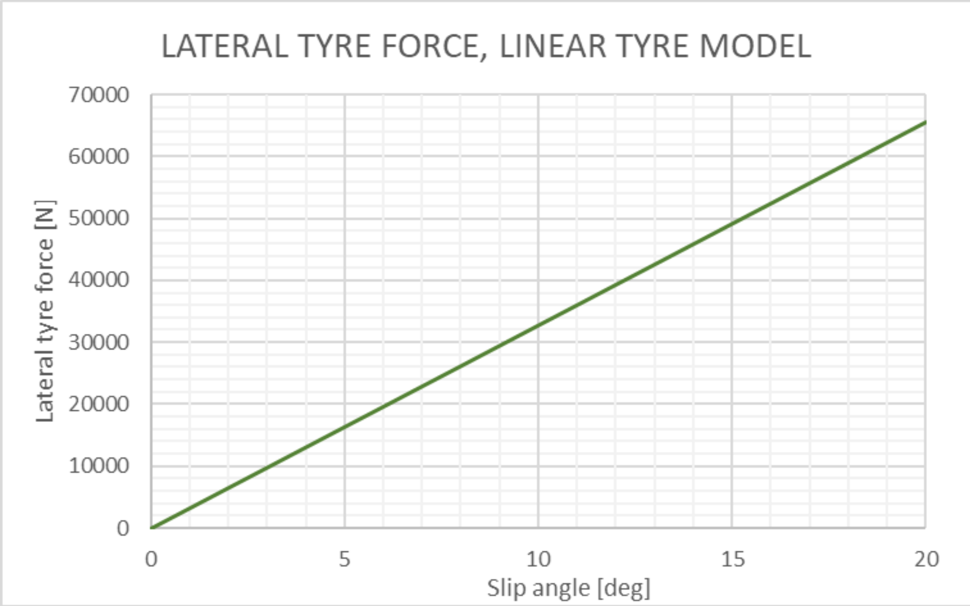

Figure 16. Lateral tyre force with axleload of 25 kN and cornering coefficient of 7.5 as a function of slip angle

In the simulation models created during this thesis a more complex tyre model will be used. This tyre model was provided by the project "Performance based standards II" (Chalmers University of technology, 2019). Tyre model is created for lateral stability estimation of heavy combination vehicles operated at dry paved road surface well below peak friction utilization. The tyre model is still in preliminary state and some more development must be done during the next steps of the project. The steady state tyre forces can be defined from

$$F_{YT,SS} = F_{ZT} \cdot u_y \cdot sin\left[C \cdot atan(\frac{CC_y}{C \cdot u_y} \cdot \alpha_y)\right] \tag{24}$$

Where $F_{YT,SS}$ is the steady state tyre lateral force, F_{ZT} is the tyre's normal force, u_y is the maximum lateral force coefficient, C is a shape factor, CC_y is a cornering coefficient and α_y is the slip angle.

The shape factor can be defined as

$$C = 2(1 + \frac{asin(u_2)}{\pi}) \tag{25}$$

Where u_2 is slide friction ratio, that has a standard value of 0.8.

The maximum lateral force coefficient can be defined as

$$u_y = u_{y,0} \cdot \frac{1}{1 - ug_y \cdot \frac{F_{ZT} - F_{ZT,0}}{F_{ZT,0}}} \tag{26}$$

Where $u_{y,0}$ is the maximum lateral force coefficient at nominal vertical tyre force with typical value of 0.8, ug_y is the maximum lateral force gradient that has a typically a value between -0.1 and -0.3. $F_{ZT,0}$ is the nominal tyre normal force that has a magnitude between 25-50 kN.

Cornering coefficient can be defined as

$$CC_y = CC_{y,0} \cdot \frac{1}{1 - ccg_y \cdot \frac{F_{ZT} - F_{ZT,0}}{F_{ZT,0}}} \tag{27}$$

Where $CC_{y,0}$ is the cornering coefficient at nominal tyre normal force and ccg_y is the maximum cornering coefficient gradient that typically has a value of -0.1.

The non-linear tyre model is a modified version of the widely used magic tyre model. Tyre model's idea is to have much smaller number of parameters compared to original magic tyre model, but with still having a connection to real, physical world with its parameters. Tyre model presented in figure 17.

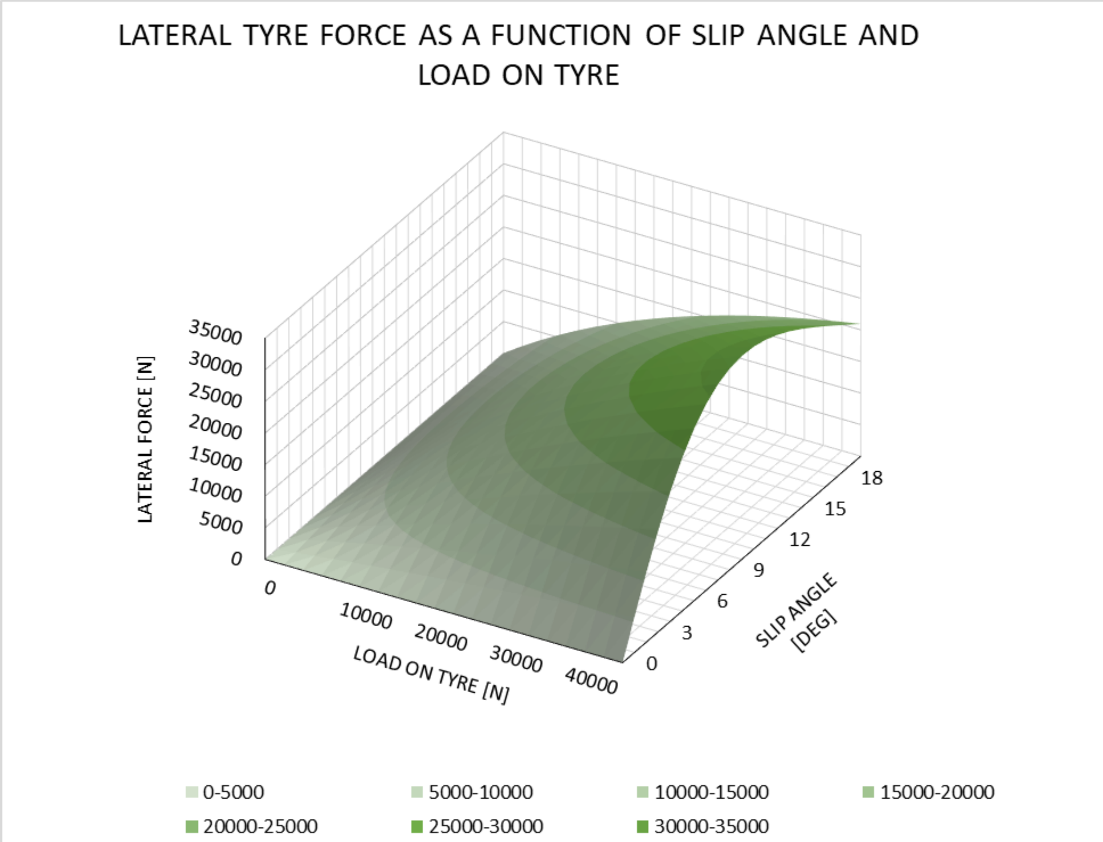

Figure 17. Non-linear tyre model used in this thesis in relation to slip angle and load on tyre, parameters used are described in equations 24 - 26.

5.4.2 Tyre relaxation

In tyre there is a delay on how fast it reaches steady state conditions. In long combinations, with multiple axles this delay will multiply trough the vehicle and thus, is an important parameter for simulation. (Jacobson, 2016).
Tyre relaxation can be modelled as a simple first order filter that will add delay to the tyre forces. That way the tyre does not react immediately to changes but with a delay.

$$\dot{F_y} = \frac{|v_x|}{L_r} \cdot \left(f\left(s_y, F_z, \mu, \dots\right) - F_y\right) \tag{28}$$

Where v_x is the unit's longitudinal velocity, $f(s_y,F_z,\mu,\dots)$ is the tyre force according to steady state conditions and L_r is the tyre relaxation length (normally 25 – 50 % of the tire circumference).

Tyre relaxation can also be implemented by adding the relaxation to tyre's slip angle instead of the tyre force. Adding a low pass filter to slip angle instead of the lateral force has the advantage that when load on tyre disappears (FZT = 0) lateral tyre force will directly become 0. The equation for slip angle will become

$$\frac{d}{dt}\alpha_y' = \frac{|v_x|}{L_r} \cdot \left(\alpha_y\left(v_x, v_y, \omega_z, \dots\right) - \alpha_y'\right) \tag{29}$$

Where α_y' is the slip angle with relaxation and $\alpha_y\left(v_x, v_y, w_z, \dots\right)$ is the slip angle according to steady state conditions.

In the simulation models both of the methods are in use. When using linear tyre model, tyre relaxation is applied to the force generation of the tyre. When the non-linear tyre model is in use, tyre relaxation is applied to the lateral slip angle of the tyre.

6 Vehicle Models

Different vehicle models developed during this thesis are described in this chapter. Four different simulation models were created and compared to each other to get an idea of the development in the simulation model and results. The parameters needed for the simulation models and the simulation results are described in the following chapters. All the simulations described in this chapter are a 3 meter wide single lane change, with a frequency of 0.3 Hz at a speed of 80 km/h. Vehicle parameters of the simulations are listed in appendix 5.

6.1 Model without roll (OpenPBS version 1)

Current version of OpenPBS vehicle model, that is used to evaluate vehicle’s lateral dynamics, is very simple model of articulated vehicle. The model is fully vectorized so that simulation of different combination vehicles or single vehicles is possible. Tyre model used in the simulation model is linear. The OpenPBS version 1 simulation model is pure single-track model with no real exemptions from the simple model.

This simulation model works with only 12 parameters. These parameters are listed in the following table 1.

Table 1. Needed parameters for simulating single lane change in OpenPBS version 1.

Parameter	Description
nu	Number of units. *e.g. "2"*
na	Maximum number of axles per unit. *e.g. "3"*
L	Axle positions from units first axle. *e.g. "[0,-3,-4.5;0,-1.3,-2.6]"*
w	Track width for each axle. *e.g. "[2.5,2.5,2.5;2.5,2.5,2.5]"*
X	Centre of gravity location from first axle of unit. *e.g. "[-1.4,2.1]"*
A	Front coupling position relative to first axle of unit. *e.g. "[0,6.4]"*
B	Rear coupling position relative to first axle of unit. *e.g. "[0,6.4]"*
driven	Defines what axles are driven. *e.g. "[0,1,1;0,0,0]"*
Cc	Cornering coefficient per axle. *e.g. "[5.5,5.5,5.5;5.5,5.5,5.5]"*
m	Units total mass. *e.g. "[9840,33100]"*
I	Units inertia around yaw axis. *e.g. "[0.299e5,5.735e5]"*
axlegroups	Used for calculating static loads on axles. *e.g. "[1,2,2;1,1,1]]"*

With low number of parameters, the results are not precise compared to high fidelity simulation models like VTM that require enormous amounts of parameters. On the other hand, simulation can be done very fast and simulation model can be used to quickly evaluate multiple different combination vehicles.

For comparison, simulations of single lane change with current openPBS vehicle model is used. Combination vehicle parameters and tyre parameters are taken from Volvo's VTM library, to make sure that realistic parameters are used.

Simulation results are presented in figure 18. Figure 18 presents vehicle trajectories for the first and last axle of the vehicle. These axles are in A-double case the first axle of the tractor and the last axle of the second semitrailer. Figure also has lateral acceleration and yaw rates for first and last unit of the combination vehicle. Rearward amplification of yaw rate is also presented in the figure.

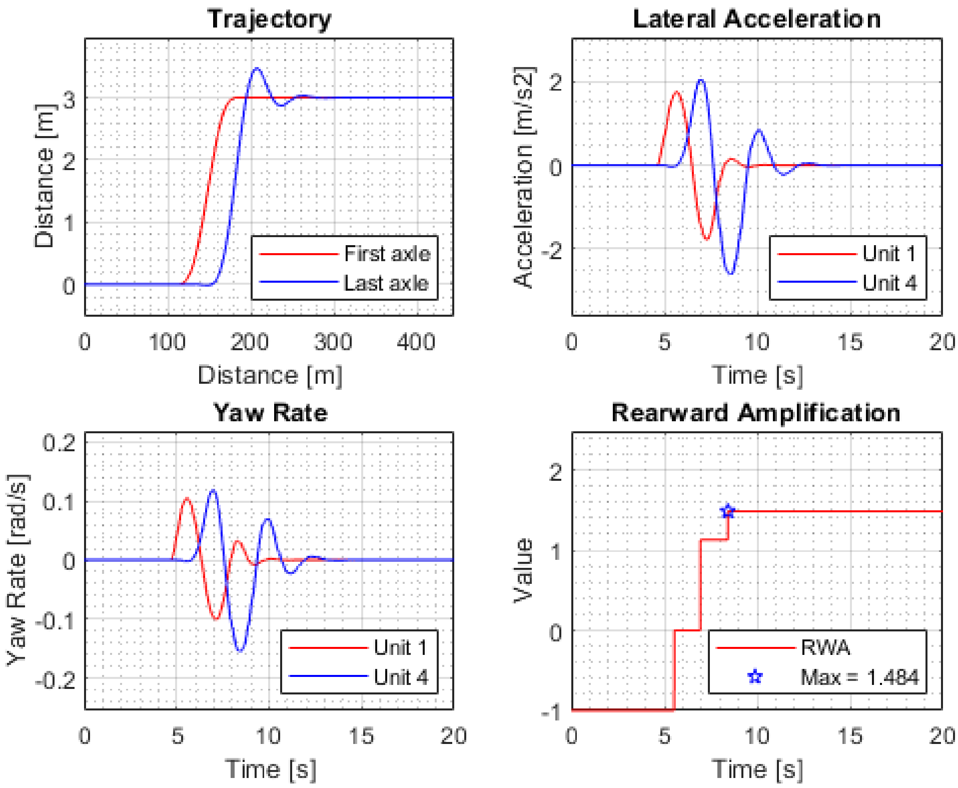

Figure 18. OpenPBS version 1 simulation results of a single lane change with A-double combination vehicle.

For A-double with OpenPBS simulation model one can get the following maximum values that are presented in table 2.

Table 2. Simulation results of an A-double combination vehicle in a single lane change.

Simulation results	**Max value of 1st unit**	**Max value of last unit**
Trajectory	3.00 m	3.47 m
Lateral Acceleration	-1.67 m/s^2	-2.60 m/s^2
Yaw Rate	-0.10 rad/s	-0.15 rad/s
Rearward Amplification		1.484

With OpenPBS simulation model the results are underestimated compared to real world situation. In his work Manjurul Islam (Islam, et al., 2019) found differences up to 50 % compared to high fidelity simulation models.

6.2 Model with roll

Roll model adds some complexity to the simulation model. More parameters are needed, but the simulation results become more accurate. Simulation model is constructed so that there are some additions made to the regular single-track model. These additions for example allow simulation of roll angle. Tyre model used in this simulation model is still linear.

Some parameters need to be added for the simulation model to be functional. Compared to single track model, the amount of vehicle parameters is increased by 6, bringing the total number of parameters to 18. Additional parameters to the basic single-track model are presented in table 3.

Table 3. Additional parameters to add roll dynamics into single track simulation model.

Parameter	Description
h	Centre of gravity height from ground. *e.g. "[1.0,1.9]"*
h_{RC}	Roll centre height. *e.g. "[0.5,0.5]"*
h_C	Coupling height. *e.g. "[0.5]"*
c_{roll}	Suspension roll stiffness. *e.g. "[1.4e6,4.5E6]"*
d_{roll}	Suspension roll dampening. *e.g. "[4.8e4,9.0e4]"*
I_x	Units inertia around roll axis. *e.g. "[4700,2.8e4]"*

With this simulation model the accuracy will increase, but the model will remain simple and only has a handful of parameters.

The simulation results for added roll dynamics with low CoG are presented in figure 19. The simulation is done with the same parameters as the previous simulation. Vehicle parameters are also taken from the same VTM vehicle model used earlier. In this chapter, two simulations will be conducted. There's a simulation with low centre of gravity height and another one with high centre of gravity height. This way it is possible to show the impact of the centre of gravity height and the roll dynamics. In this occasion low centre of gravity height refers to CoG height of 1 meter. On the contrary high centre of gravity refers to CoG height of 2.5 meters.

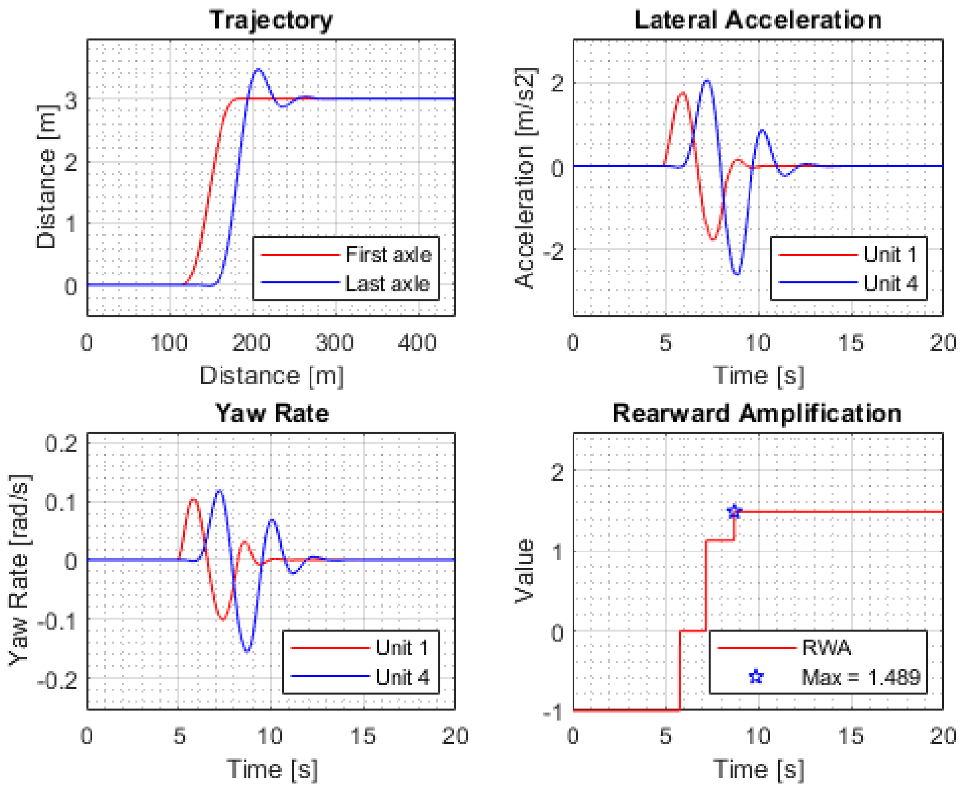

Figure 19. Simulation results of a single track model with added roll dynamics of an A-double combination vehicle with centre of gravity height of 1 meters in a single lane change.

The maximum values of single lane change simulation of an A-double with low centre of gravity are presented in table 4.

Table 4. Simulation results of A-double combination with roll dynamics added to the simulation model. Simulation is done with low CoG height.

Simulation results	**Max value of 1st unit**	**Max value of last unit**
Trajectory	3.00 m	3.47 m
Lateral Acceleration	-1.77 m/s^2	-2.61 m/s^2
Yaw Rate	-0.10 rad/s	-0.15 rad/s
Rearward Amplification		1.489

In simulations with low centre of gravity height the results are similar compared to the OpenPBS version 1 simulation model. There are small differences in the last unit's lateral acceleration and in the rearward amplification. But with centre of gravity height so low, only 1 meters, this can be expected. Simulation results for added roll dynamics and high CoG presented in figure 20.

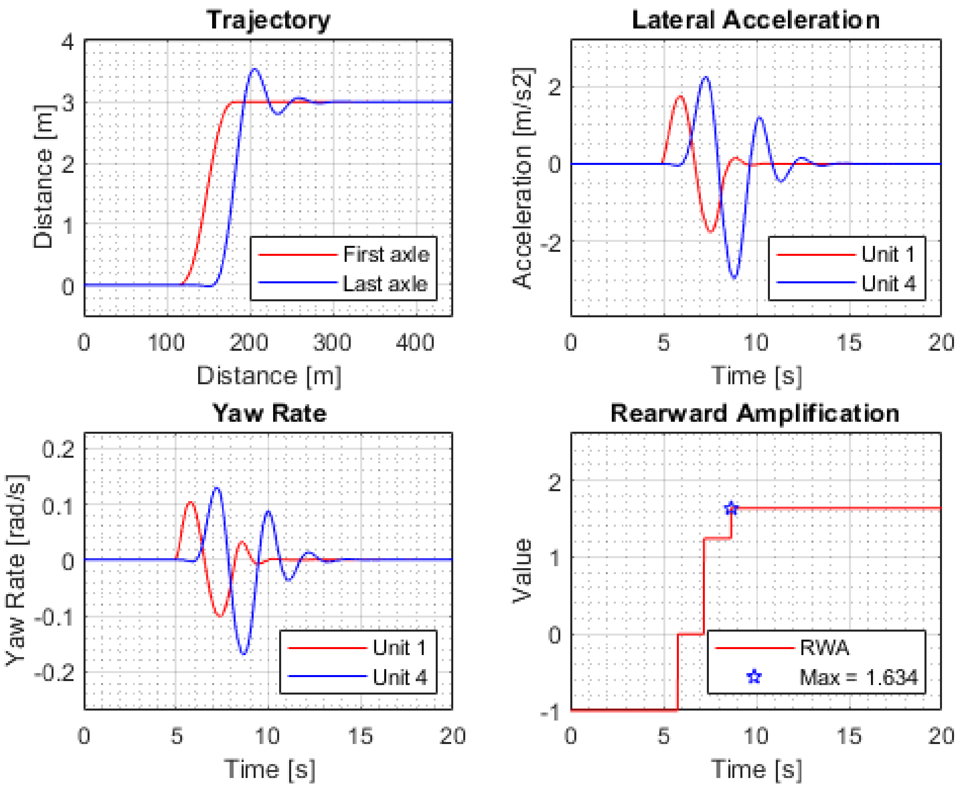

Figure 20. Simulation results of a single track model with added roll dynamics of an A-double combination vehicle with centre of gravity height of 2.5 meters in a single lane change.

The maximum values of single lane change simulation of an A-double with high centre of gravity are presented in table 5.

Table 5. Simulation results of A-double combination vehicle with roll dynamics added to the simulation model. Simulation is done with high CoG height.

Simulation results	Max value of 1st unit	Max value of last unit
Trajectory	3.00 m	3.54 m
Lateral Acceleration	-1.77 m/s^2	-2.96 m/s^2
Yaw Rate	-0.10 rad/s	-0.17 rad/s
Rearward Amplification		1.634

When comparing low and high centre of gravity results to one another it's easy to observe the effect of the higher centre of gravity. The last trailer overshoots by 0.07 meters, the lateral acceleration is 0.35 m/s^2 higher, the yaw rate increases by 0.02 rad/s and the rearward amplification increases by 0.145, compared to the low CoG simulation. The increase in rearward amplification is almost 10 % compared to the low CoG simulation

6.3 Model with roll and tyre relaxation

This simulation model adds tyre relaxation to the previous simulation model that added roll dynamics to single track model. Adding just the tyre relaxation to the vehicle model is very simple thing to do and it adds just one parameter to the model. Tyre relaxation is done by creating a first-degree filter to the lateral tyre force, thus adding a delay to the tyre force generation. Only added parameter to the vehicle model is the relaxation length. This addition will increase the number of parameters to 19. Added parameters are presented in table 6. The simulation still uses linear tyre model.

Table 6. Additional parameters to add tyre relaxation to vehicle model.

Parameter	Description
L_r	Relaxation length. *e.g. "[0.4,0.4,0.4;0.4,0.4,0.4]"*

Simulation results of vehicle model with roll dynamics and tyre relaxation is presented in figure 21. The centre of gravity height used is the same 2.5 meters as in previous simulations. The tyre cornering coefficient used is the same 7.5 as used in previous simulations

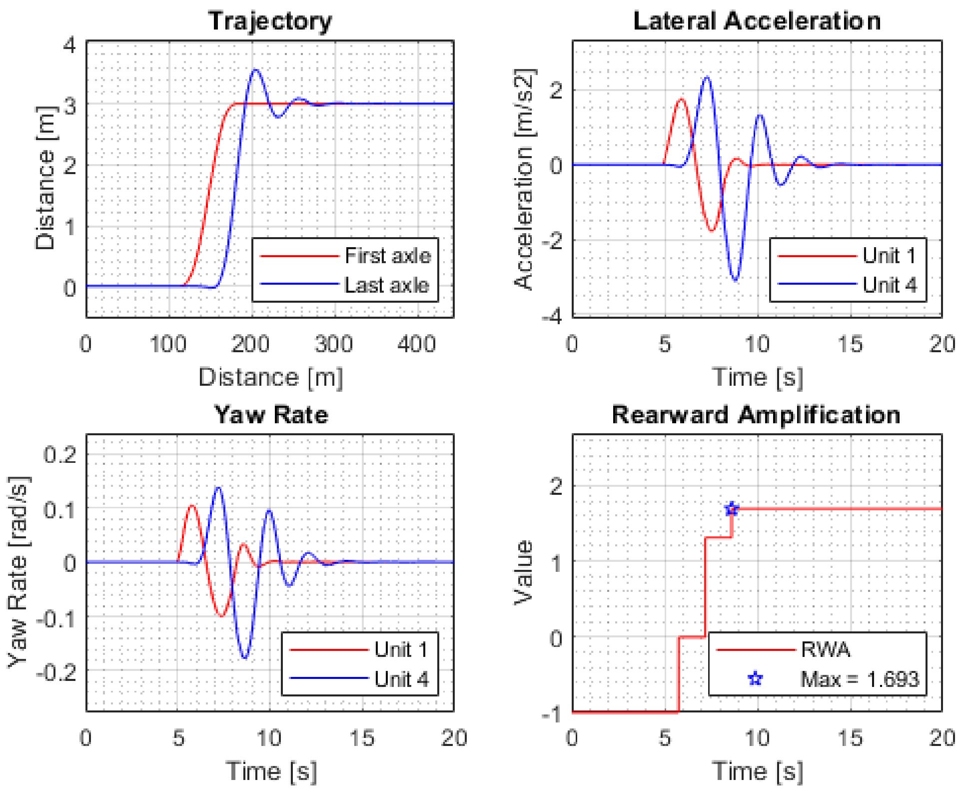

Figure 21. Simulation results of single lane change with a A-double combination vehicle with roll dynamics and tyre relaxation.

The maximum values of single lane change simulation of an A-double with high centre of gravity and tyre relaxation are presented in table 7.

Table 7. Simulation results of A-double combination with roll model with tyre relaxation.

Simulation results	**Max value of 1st unit**	**Max value of last unit**
Trajectory	3.00 m	3.56 m
Lateral Acceleration	-1.78 m/s^2	-3.10 m/s^2
Yaw Rate	-0.10 rad/s	-0.17 rad/s
Rearward Amplification		1.693

There is an increase especially in the values of the last unit. Most notable differences are the last unit's lateral acceleration increase of 0.14 m/s^2 and the difference of 0.059 in rearward amplification compared to the previous simulation with only the roll dynamics. The increase in the rearward amplification is a 3.6 %.

6.4 Model with roll, tyre relaxation and non-linear tyre model

This simulation model completes the simulation model development by adding non-linear tyre model to the simulation model with roll dynamics and tyre relaxation. Adding a non-linear tyre model adds 5 more parameters that are connected to the tyre model itself. These parameters are mostly parameters that have some specific values based on for example tyre properties. Same cornering coefficient of 7.5 is used. After tyre model, the number of parameters in vehicle model is 24. Additional parameters are presented in table 8.

Table 8. Additional parameters to introduce into the vehicle model to add tyre model to the simulation.

Parameter	Description
FZT	Nominal tyre force. *e.g. "25 000"*
$u_{y,0}$	Maximum lateral force coefficient at nominal vertical tyre force. *e.g. "0.8"*
ug_y	Maximum lateral force gradient. *e.g. "0.2"*
u_2	Slide friction ratio. *e.g. "0.8"*
ccg_y	Maximum cornering coefficient gradient. *e.g. "-0.1"*

Simulation results for A-double in a single lane change with non-linear tyre model, roll dynamics and tyre relaxation are presented in figure 22. The CoG height is 2.5 meters. Tyre parameters were provided for this thesis by the "performance based standards" -project.

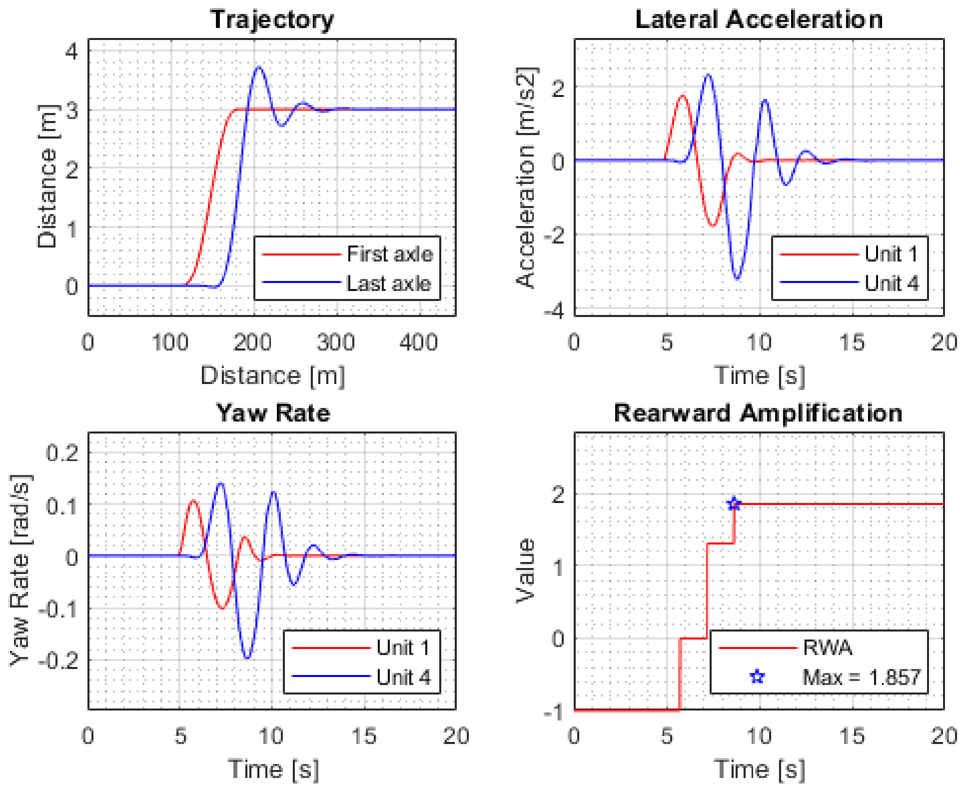

Figure 22. Simulation results of an A-double combination vehicle with roll dynamics and non-linear tyre model.

Maximum values of single lane change simulation of an A-double with high centre of gravity and non-linear tyre model are presented in table 9.

Table 9. Simulation results of A-double combination with non-linear tyre model added to vehicle model with roll dynamics and tyre relaxation.

Simulation results	**Max value of 1st unit**	**Max value of last unit**
Trajectory	3.00 m	3.71 m
Lateral Acceleration	-1.78 m/s^2	-3.23 m/s^2
Yaw Rate	-0.10 rad/s	-0.20 rad/s
Rearward Amplification		1.857

By adding the non-linear tyre model there is a significant increase in the values for the last vehicle unit. The overshoot of the last axle increases 0.15 meters, the lateral acceleration increases 0.13 m/s^2, the yaw rate increases by 0.03 rad/s and there is an increase of 0.164 in the rearward amplification. This increase in rearward amplification is about 9.6 % increase compared to the simulation with roll dynamics and tyre relaxation.

6.5 Vehicle parameters

To clarify the vehicle parametrization more, the vehicle parameters must be presented with figures. To present this a bit more clearly, this chapter is divided into two parts. In first part the vehicle measurements are described, and in the second part, all the other parameters are explained more carefully. In this chapter the focus is only on the vehicle parameters. This chapter was produced to help with work package 5 in “performance based standards II” -project.

6.5.1 Measurements

The vehicle measurements are presented in the figure 23. The figure contains basic description of how the vehicle parametrization is conducted. Because of its size, larger version of the figure is also available in the appendix 6.

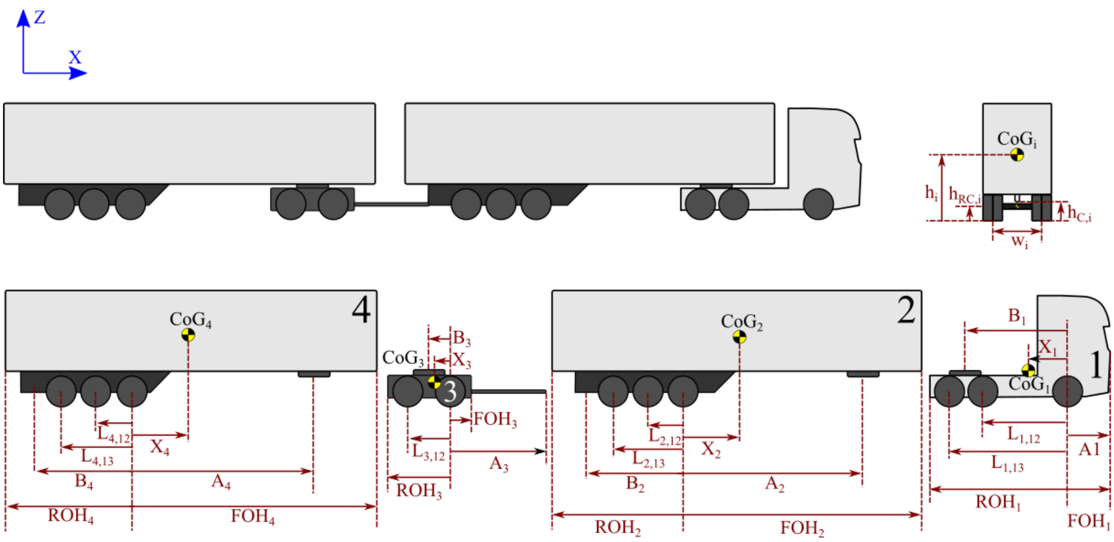

Figure 23. Vehicle parametrization in OpenPBS. Figure also available in appendix 6.

Size of the vehicle axle position matrix is defined by the parameters *nu* (Number of units) and *na* (Number of Axles). In A-double, *nu = 4* and *na = 3*, so *L* matrix has to have four rows and three columns. In this matrix 1st row represents 1st unit, 2nd row is 2nd unit etc. Similarly 1st colum is the 1st axle, 2nd column is the 2nd etc. So for example value at [2,2] is the 2nd unit's 2nd axle's distance from the first axle of the second unit.

$$L = \begin{bmatrix} 0 & L_{1,12} & L_{1,13} \\ 0 & L_{2,12} & L_{2,13} \\ 0 & L_{3,12} & 0 \\ 0 & L_{4,12} & L_{4,12} \end{bmatrix}$$

From the there the matrix can be defined in the Modelica format:

$L = [0, L_{1,12}, L_{1,13};\ 0, L_{2,12}, L_{2,13};\ 0, L_{3,12}, 0;\ 0, L_{4,12}, L_{4,13}]$

Which, for example in A-double combination vehicle translates to

$L = [0.0, -3.4, -4.77\ ;\ 0.0, -1.3, -2.6\ ;\ 0.0, -1.31, 0.0\ ;\ 0.0, -1.3, -2.6]$

Track width w is defined similarly to the axle position matrix. From the parameters *nu* and *na* we get the size of the track width matrix. In A-double's case the size of the matrix is 4 x 3.

$$W = \begin{bmatrix} W_{11} & W_{12} & W_{13} \\ W_{21} & W_{22} & W_{23} \\ W_{31} & W_{32} & 0 \\ W_{41} & W_{42} & W_{43} \end{bmatrix}$$

From there the matrix can be defined in the Modelica format:

$W = [W_{11}, W_{12}, W_{13}; W_{21}, W_{22}, W_{23}; W_{31}, W_{32}, 0; W_{41}, W_{42}, W_{43}]$

In an example A-double the values could be as follows:

$W = [2.09, 1.85, 1.85;\ 2.05, 2.05, 2.05;\ 2.05, 2.05, 0;\ 2.05, 2.05, 2.05]$

Parameters A, B, FOH, ROH, X, h, hRC, hC are defined in the same way between each other. Each of these parameters has one value per unit. Parameter A describes the unit's front coupling position from the first axle of the unit. For example for combination vehicle with 4 units this is presented in matrix format,

$A = [A_1 \quad A_2 \quad A_3 \quad A_4]$

and in Modelica format:

$A = \{A_1, A_2, A_3, A_4\}$

Example parameters for an A-double can be:

$A = \{0.0, 6.8, 3.9, 6.8\}$

Parameter B describes the unit's rear coupling position from the first axle of the unit. For example for combination vehicle with 4 units this is presented in matrix format,

$B = [B_1 \quad B_2 \quad B_3 \quad B_4]$

and in Modelica format:

$B = \{B_1, B_2, B_3, B_4\}$

Example parameters for an A-double can be:

$B \ = \ \{-3.775, -3.7, -0.65, -3.7\}$

FOH and ROH are the front and the rear overhang of the units, again from the first axle of the unit. For example for combination vehicle with 4 units, these are presented in matrix format,

$FOH = [FOH_1 \quad FOH_2 \quad FOH_3 \quad FOH_4]$

$ROH = [ROH_1 \quad ROH_2 \quad ROH_3 \quad ROH_4]$

and in Modelica format:

$FOH = \{FOH_1, FOH_2, FOH_3, FOH_4\}$

$ROH = \{ROH_1, ROH_2, ROH_3, ROH_4\}$

Example parameters of an A-double combination vehicle

$FOH = \{0.5,7.15,0.25,7.15\}$

$ROH = \{-4.8, -3, -1.55, -3.1\}$

Parameter X describes the CoG position of the unit from the first axle of the unit. For example for combination vehicle with 4 units this is presented in matrix format,

$X = [X_1 \quad X_2 \quad X_3 \quad X_4]$

and in Modelica format:

$X = \{X_1, X_2, X_3, X_4\}$

Example parameters for an A-double can be:

$X \ = \ \{-1.86, 1.54, -0.17, 1.54\}$

There are three height values, which are all distances from the ground level. For CoG height the parameter is h, for roll centre it's h_{RC} and for coupling it's h_C.

Parameter hC differs a bit from the others, as it only contains values for the different coupling heights between the units. So the amount of values is actually *nu – 1*. For example for combination vehicle with 4 units these presented in matrix format,

$h = [h_1 \quad h_2 \quad h_3 \quad h_4]$

$h_{RC} = [h_{RC_1} \quad h_{RC_2} \quad h_{RC_3} \quad h_{RC_4}]$

$h_C = [h_{C_1} \quad h_{C_2} \quad h_{C_3}]$

and in Modelica format:

$h = \{h_1, h_2, h_3, h_4\}$

$h_{RC} = \{h_{RC_1}, h_{RC_2}, h_{RC_3}, h_{RC_4}\}$

$h_C = \{h_{C_1}, h_{C_2}, h_{C_3}\}$

Example parameters for an A-double can be:

$h = \{0.97, 1.89, 0.74, 1.89\}$

$h_{RC} = \{0.681, 0.5450, 0.52, 0.5450\}$

$hC = \{1.0, 0.5, 1.0\}$

In A-double for example the first value of h_C is the height of the coupling between the tractor and the first semitrailer. Second value is the height of the coupling between the first semitrailer and the dolly. Third value is the coupling height between the dolly and the last semitrailer.

6.5.2 Other parameters

In the simulation model, the vehicle axlegroups must be defined to be able to calculate the static tyre loads. This can be done simply with a matrix, where the tractor's rear axles get a value of 2, every other axle get a value of 1 and where there is "no axle" the value is 0. For example, for an A-double combination vehicle:

$axlegroups = [1, 2, 2\ ;\ 1, 1, 1\ ;\ 1, 1, 0;\ 1, 1, 1]$

Next vehicle parameter that must be defined in the simulation model is the driven axles. This is done so that one can get accurate information of the driving force/friction needed for manoeuvres. This parameter is again a simple matrix, with Boolean values. For example, for an A-double combination vehicle:

$driven =$

$[false, true, true\ ;\ false, false, false\ ;\ false, false, false\ ;\ false, false, false]$

The total masses of combination vehicle's units must be defined in a matrix form also. The matrix has the same amount of values as the combination vehicle has units. For example, for A-double combination vehicle:

$m\ =\ \{9231.0, 31000.0, 2800.0, 31000.0\}$

In the simulation model, the inertia values for yaw and roll must be also defined. These are defined with two matrices that have the same amount of values as the combination vehicle has units. Parameter I_z is the inertia value in the yaw plane and the parameter I_x is the inertia value in roll plane. These values are for example for A-double combination vehicle:

$Iz\ =\ \{4.4483e4, 4.6524e5, 6.6674e3, 4.6524e5\}$

$Ix\ =\ \{4700.2, 2.8429e4, 1000, 2.8429e4\}$

Two last parameters that must be defined are the suspension parameters. These parameters are the roll stiffness and the roll damping of the suspension. These values are defined as matrices, and each axle has its own value. So, for A-double for example the matrices will be 4 x 3 matrices. The roll stiffness matrix can be defined as:

$$k_s = \begin{bmatrix} 4.6e5 & 4.8e5 & 4.8e5 \\ 1.5e6 & 1.5e6 & 1.5e6 \\ 1.5e6 & 1.5e6 & 0 \\ 1.5e6 & 1.5e6 & 1.5e6 \end{bmatrix}$$

And in Modelica format:

$k_s\ =$

$[4.6e5, 4.8e5, 4.8e5;\ 1.5e6, 1.5e6, 1.5e6;\ 1.5e6, 1.5e6, 0;\ 1.5e6, 1.5e6, 1.5e6]$

Similarly, the roll damping matrix can be defined as:

$$c_s = \begin{bmatrix} 14e3 & 16e3 & 16e3 \\ 30e3 & 30e3 & 30e3 \\ 30e3 & 30e3 & 0 \\ 30e3 & 30e3 & 30e3 \end{bmatrix}$$

And in Modelica format:

$c_s\ =\ [14e3, 16e3, 16e3;\ 30e3, 30e3, 30e3;\ 30e3, 30e3, 0;\ 30e3, 30e3, 30e3]$

This set of parameters are the parameters that have a direct connection to the vehicle itself. There are still some parameters that weren't disclosed here, like the parameters in the non-linear tyre model.

7 Results

In this chapter the results of the simulation models are presented and compared against one another. In the comparisons, different key values are used to present the results of the different simulation models as good as possible. These key values include rearward amplification, yaw damping, lateral load transfer and high-speed transient off tracking of different combination vehicles. Vehicle parameters of the simulations are listed in appendix 5.

7.1 Rearward amplification between simulation models

In addition to the simulation results presented in the chapter 6.1 - 6.4, more precise comparisons are needed. First the comparison in the simulation of three different combinations in a 3-meter-wide lane change with the 0.3 Hz frequency and at a speed of 80 km/h. The three combinations used were A-double, Nordic Combination and Double CAT combination. The rearward amplification results of these simulations are presented in figure 24. Used CoG heights are as follows, low CoG for all combination vehicles is 1 meter for all units. High CoG is 2.5 meters for A-double and Nordic combination and 2 meters for double CAT for all load bearing units. High centre of gravity is used in all other simulation models than Roll model with low CoG.

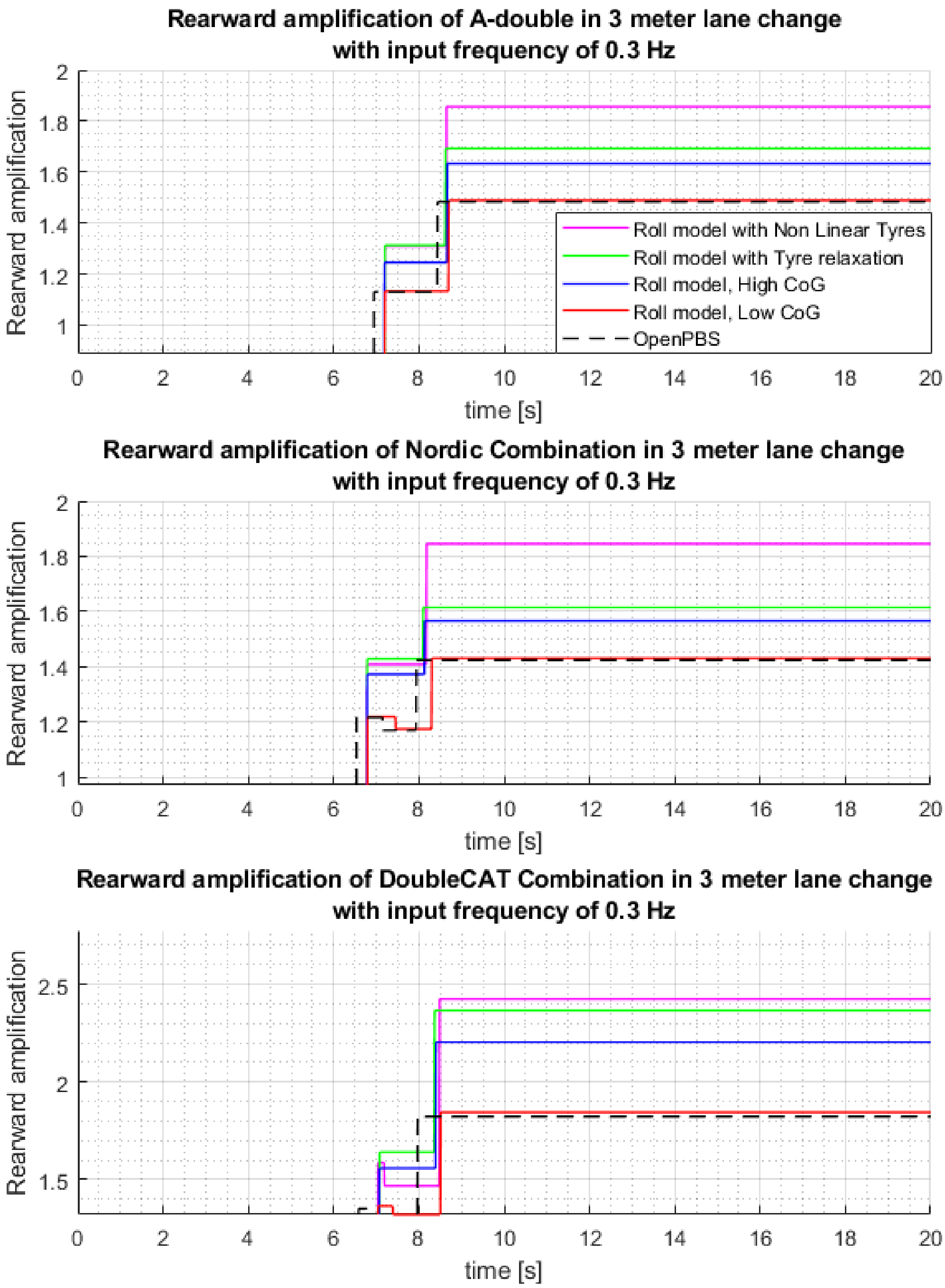

Figure 24. Combination vehicles rearward amplification in a 3 meter wide lane change with different simulation models.

From the figures the maximum values for the rearward amplification are presented in the table 10.

Table 10. Results from 3 meter lane change with different vehicles and simulation models.

	OpenPBS	**Roll model + Low CoG**	**Roll model + High CoG**	**Roll model + Tyre relaxation**	**Roll model + Non-linear tyres**
RWA (A-double)	1.484	1.489	1.634	1.693	1.857
Change from OpenPBS	± 0 %	+ 0.3 %	+ 10.1 %	+14.1 %	+ 25.1 %
RWA (Nordic Combination)	1.424	1.43	1.566	1.614	1.854
Change from OpenPBS	± 0 %	+ 0.4 %	+ 10.0 %	+ 13.3 %	+ 30.2 %
RWA (Double CAT)	1.823	1.845	2.204	2.366	2.425
Change from OpenPBS	± 0 %	+ 1.2 %	+ 20.9 %	+ 29.8 %	+ 33.0 %

With these simulation parameters we can see an increase of up to 33 % in combination vehicle's rearward amplification. The largest additions to the rearward amplification depend on the vehicle. For example, in double CAT the addition of the non-linear tyre model, doesn't make that huge of a difference, it might have something to do with the tyre parameters of the non-linear tyre model. In double CAT combination the effect of the CoG height is the most noticeable, because there are three units that are affected by the high CoG compared to just two in A-double and Nordic combinations.

For the rearward amplification, also a frequency study was made. In the frequency study, the vehicle parameters are the same, but instead of a lane change input, sine wave lateral acceleration on the first axle will be used instead. The lateral acceleration has maximum value of 1.5 m/s^2. The results are presented in figure 25.

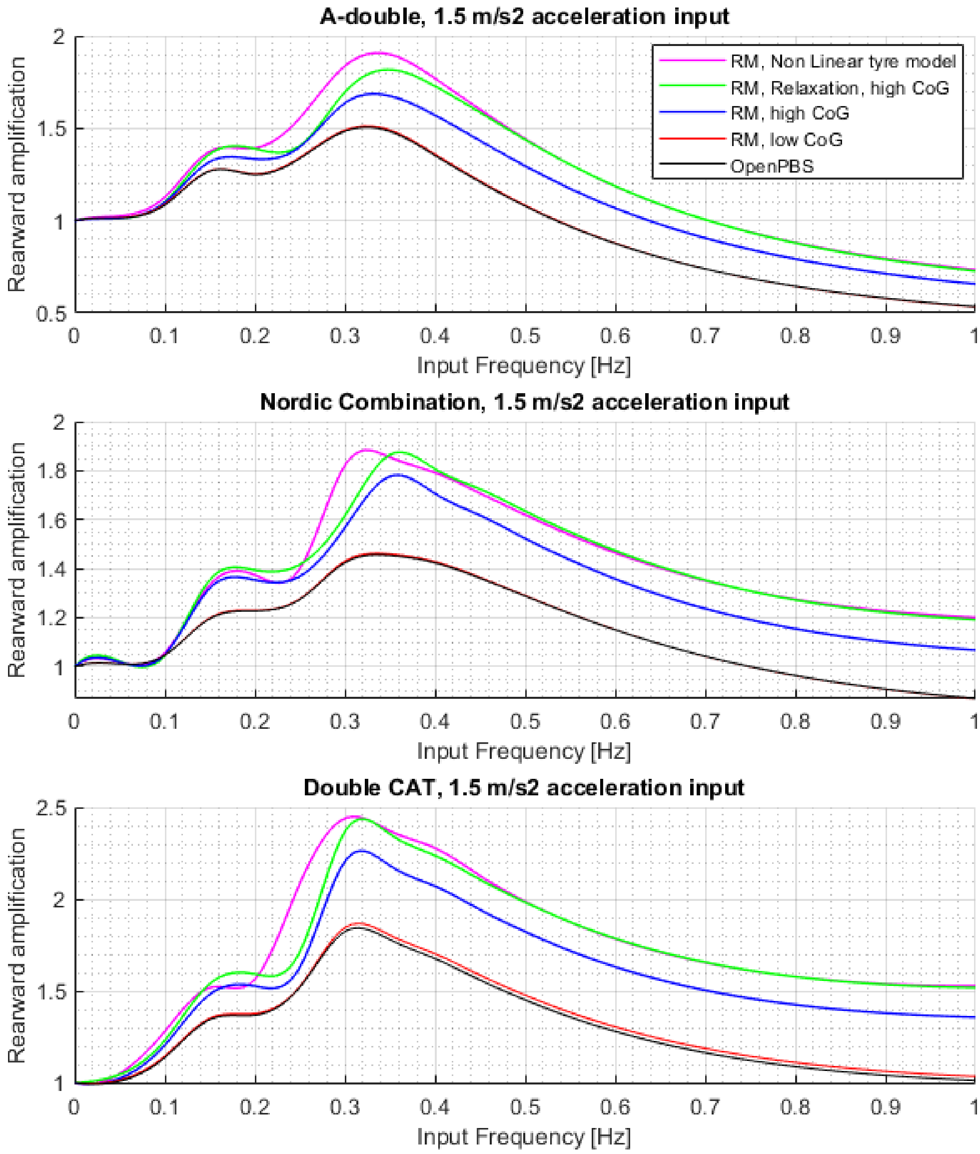

Figure 25. Frequency study on different combination vehicles and with different simulation models with lateral acceleration input.

From the frequency study, the maximum rearward amplification values are presented in table 11.

Table 11. Frequency study results of rearward amplification of different combination vehicles and simulation models.

	OpenPBS	Roll model + Low CoG	Roll model + High CoG	Roll model + Tyre relaxation	Roll model + Non-linear tyres
RWA (A-double)	1.506	1.512	1.686	1.818	1.908
Change from OpenPBS	± 0 %	+ 0.4 %	+ 12.0 %	+20.7 %	+ 26.7 %
RWA (Nordic combination)	1.456	1.462	1.782	1.875	1.884
Change from OpenPBS	± 0 %	+ 0.4 %	+ 22.4 %	+ 28.8 %	+ 29.4 %
RWA (Double CAT)	1.843	1.867	2.262	2.435	2.448
Change from OpenPBS	± 0 %	+ 1.3 %	+ 22.7 %	+ 32.1 %	+ 32.8 %

The increase in maximum values are the same as in lane change. From the results with these tyre parameters Nordic combination and Double CAT suffer most from the roll dynamics/high CoG, even though the final addition to the rearward amplification is close to be the same.

7.2 Yaw damping between simulation models

Yaw damping is also one of the key values used to describe the combination vehicle's stability. Using the same 3-meter lane change for the combination vehicles, we can get the values for the combination vehicle's yaw damping. The results for different simulation models are presented in figure 26.

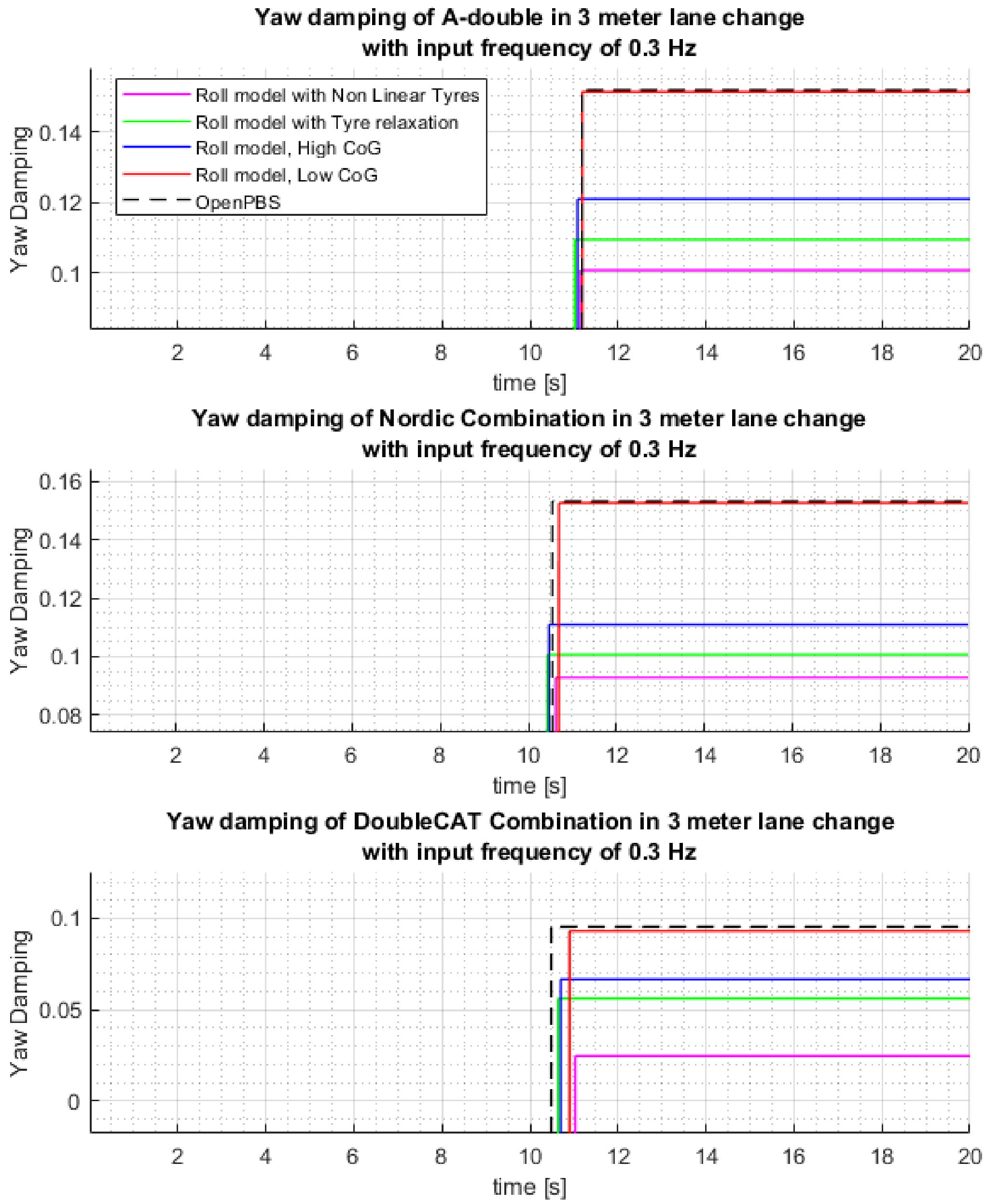

Figure 26. Yaw damping of combination vehicles in single lane change manoeuvre with different simulation models.

From the lane change simulation, the maximum values for the combination vehicle's yaw damping are presented in table 12.

Table 12. Yaw damping in different combination vehicles and simulation models during a single lane change.

	OpenPBS	Roll model + Low CoG	Roll model + High CoG	Roll model + Tyre relaxation	Roll model + Non-linear tyres
YD (A-double)	0.1519	0.1513	0.121	0.1096	0.1009
Change from OpenPBS	± 0 %	- 0.4 %	- 20.3 %	- 27.8 %	- 33.6 %
YD (Nordic Combination)	0.1533	0.1526	0.111	0.101	0.0929
Change from OpenPBS	± 0 %	- 0.5 %	- 27.6 %	- 34.2 %	- 39.4 %
YD (Double CAT)	0.095	0.093	0.067	0.056	0.025
Change from OpenPBS	± 0 %	- 2.2 %	- 30.1 %	- 41.1 %	- 74.1 %

Yaw damping has even bigger change than rearward amplification. In Double CAT there is over exaggeration, and during simulations, it was clear that the vehicle was not stable in the simulations. These results might be an indication that there is some problem with the tyre parameters, and they would need to be adjusted for the Double CAT simulation model. With only tyre relaxation, the simulation result is quite near the Nordic combination and the A-double, but when the non-linear tyre model is added the difference increases vastly.

This decrease in the yaw damping can be observed also easily from the coupling angles between the dolly and the last semitrailer of the A-double combination vehicle with the two different simulation models. Coupling angles are presented in figure 27.

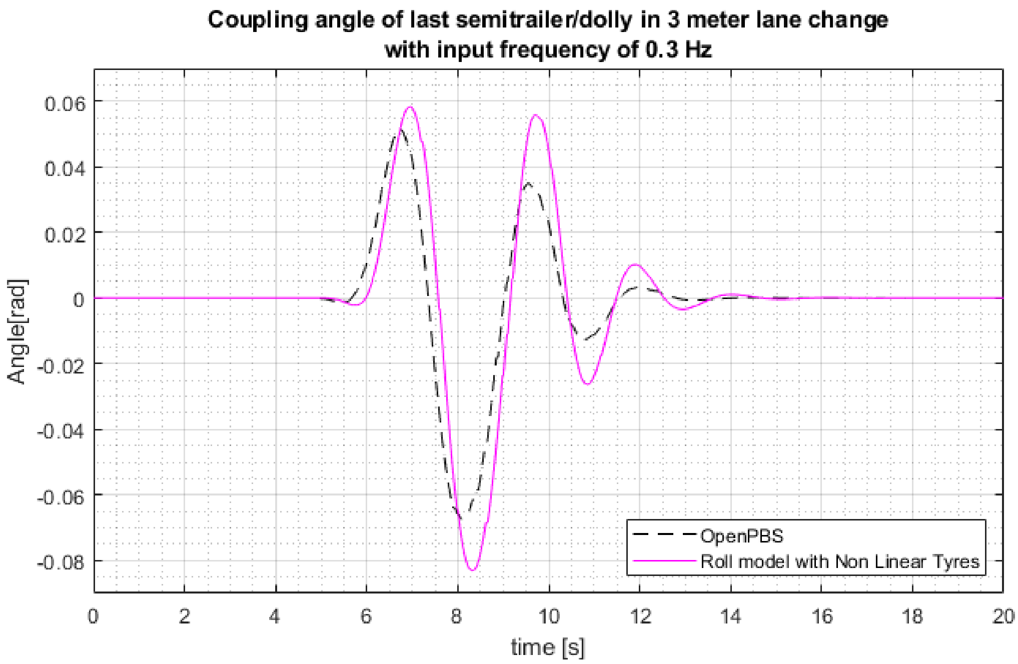

Figure 27. Coupling angle as a function of time in a simulation with A-double combination vehicle in single lane change manoeuvre.

From the figure, the yaw damping is much smaller in the roll model with non-linear tyres when compared to current version of openPBS.

7.3 Lateral load transfer between simulation models

Lateral load transfer describes the vehicles rollover stability relatively well. Lateral load transfers for different combination vehicles with different simulation models are presented in figure 28. For the simulation, same manoeuvre was used, a 3-meter lane change with 0.3 Hz frequency. The lateral load transfer in this figure is the combined lateral load transfer ratio of the last unit's axles of the combination vehicle.

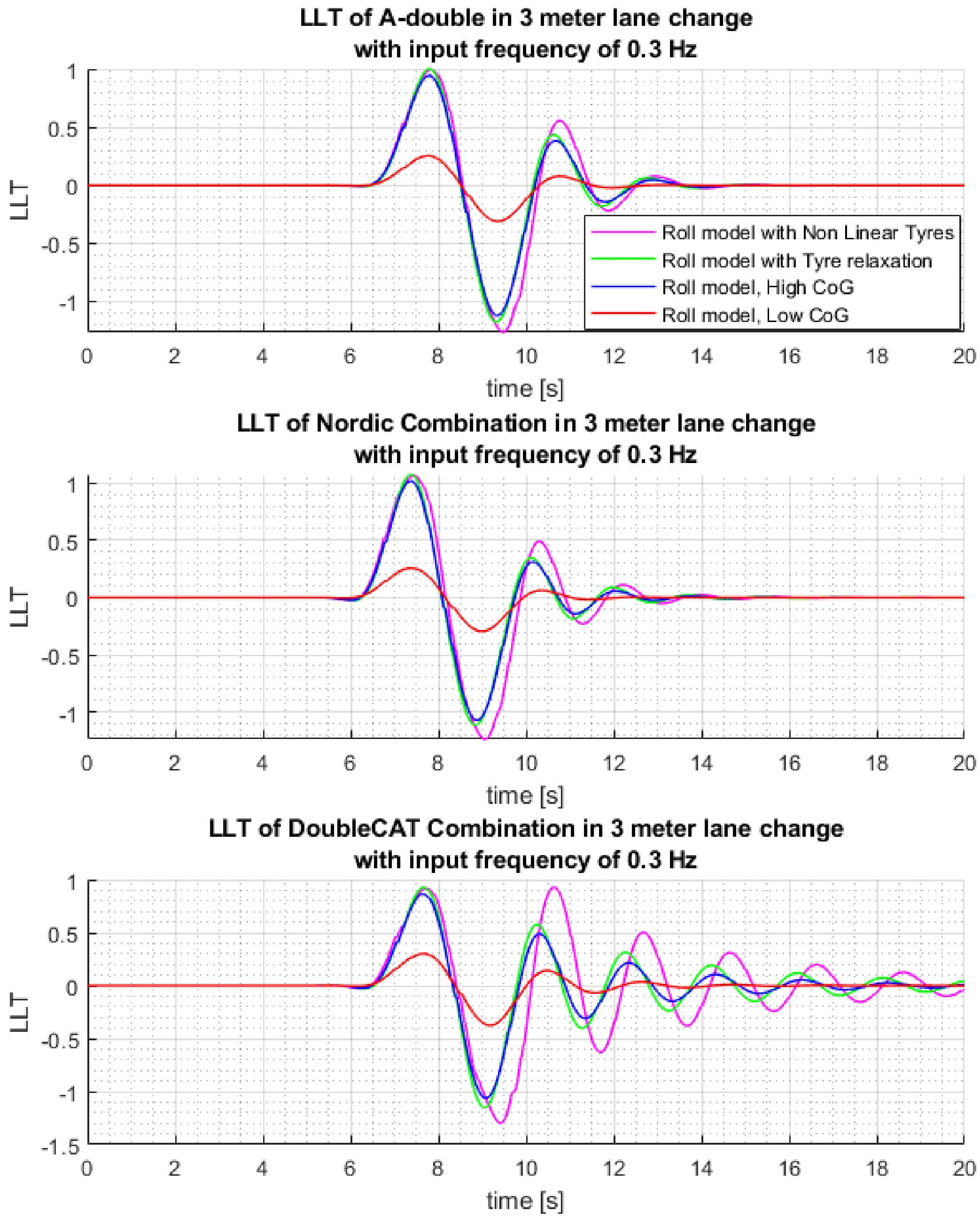

Figure 28. Lateral load transfer ratio of different combination vehicles with different simulation models in a single lane change manoeuvre.

There is a large difference in the lateral load transfer ratio between the different simulation models. Current version of OpenPBS does not have lateral load transfer built in. Value for lateral load transfer for OpenPBS was calculated with units' lateral acceleration, CoG height, mass and track width as follows:

$$\Delta F_{z_i} = \frac{m_i \cdot a_{y_i} \cdot h_{CoG_i}}{w_i} \tag{30}$$

Where ΔF_{z_i} is the change in vertical load in one side of the vehicle unit, a_{y_i} is the lateral acceleration of the unit, h_{CoG_i} is the CoG height of the unit and w_i is the track width of the unit.

The largest addition comes from the CoG height and small additions come also from the tyre relaxation and the non-linear tyre model in the roll model. In the lateral load transfer ratio, if the value exceeds ±1 the unit lifts the wheels on the other side and is at least very close to rollover. Maximum values of different simulation models are presented in table 13.

Table 13. Maximum lateral load transfer ratios between different simulation models and different combination vehicles.

	OpenPBS	**Roll model + Low CoG**	**Roll model + High CoG**	**Roll model + Tyre relaxation**	**Roll model + Non-linear tyres**
LLT (A-double)	0.501	0.309	1.121	1.174	1.258
Compared to OpenPBS	± 0 %	- 38 %	+ 123 %	+ 134 %	+ 151 %
LLT (Nordic Combination)	0.478	0.295	1.069	1.113	1.237
Compared to OpenPBS	± 0 %	- 38 %	+ 123 %	+ 132 %	+ 158 %
LLT (Double CAT)	0.449	0.372	1.065	1.152	1.297
Compared to OpenPBS	± 0 %	- 17 %	+ 137 %	+ 156 %	+ 188 %

From the results it can be observed, that the last unit of the combination vehicle is at least very close to rollover. One reason for this in A-Double and Nordic combination is the absence of roll stiffness in the fifth wheels of the simulation model. In this case, the dolly or the tractor do not "carry" any of the lateral load transfer and thus, make the combination vehicle more unstable. This lack of lateral load transfer in the dolly is shown in the figure 29.

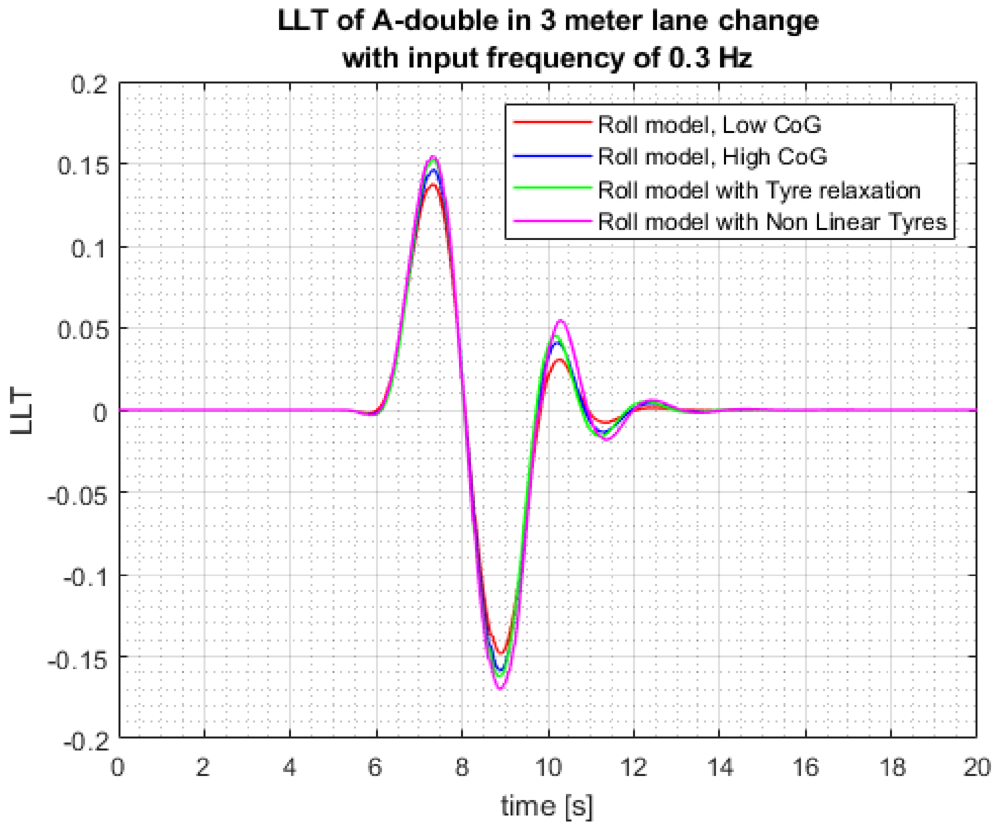

Figure 29. Lateral load transfer ratio in the dolly during lane change simulation with different simulation models.

The difference of lateral load transfer in the dolly between different simulation models is minimal, compared to the combination vehicle's last unit lateral load transfer ratio. The difference between the lowest and highest lateral load transfer ratio is only 0.02.

7.4 High speed transient off tracking between simulation models

The last key value, regarding the lateral movement in the PBS measures, is the high-speed transient off tracking. For these simulations the same 3-meter lane change and an A-double vehicle is used. Simulation results are presented in the figure 30.

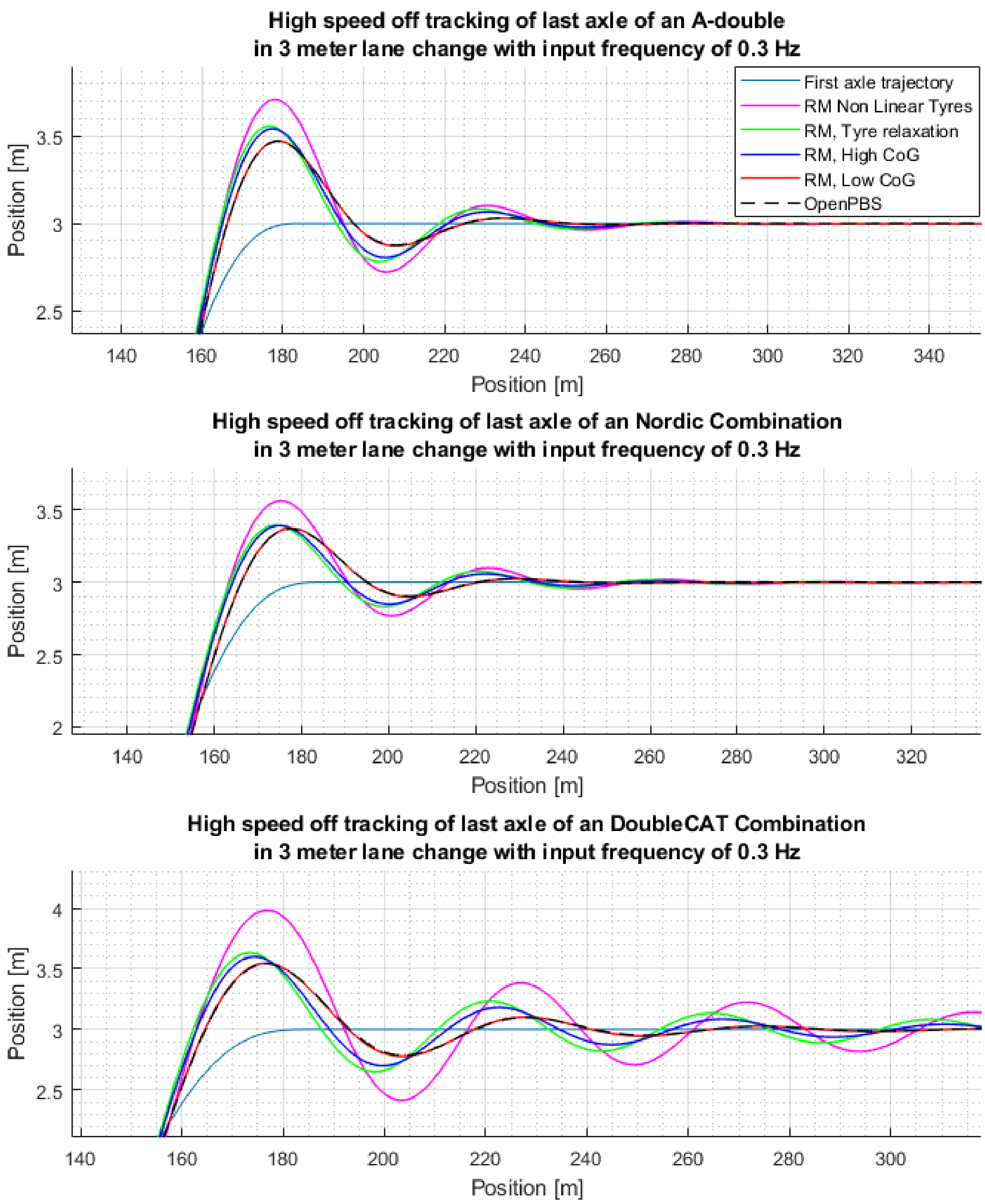

Figure 30. High speed transient off tracking simulation results of different combination vehicles doing a single lane change with different simulation models.

From the figure, again, the results are the worst for the roll model with non-linear tyre model. The results of the high-speed transient off tracking are presented in the table 14.

Table 14. High speed transient off tracking of last axle in combination vehicle in a single lane change with different simulation models.

	OpenPBS	Roll model + Low CoG	Roll model + High CoG	Roll model + Tyre relaxation	Roll model + Non-linear tyres
HSTO (A-double)	0.4707	0.4723	0.5420	0.5574	0.7101
Change from OpenPBS	± 0 %	+ 0.3 %	+ 15.1 %	+ 18.4 %	+ 50.9 %
HSTO (Nordic Combination)	0.3681	0.3709	0.3908	0.3936	0.5624
Change from OpenPBS	± 0 %	+ 0.8 %	+ 6.2 %	+ 6.9 %	+ 52.8 %
HSTO (Double CAT)	0.5425	0.5453	0.5996	0.6329	0.9818
Change from OpenPBS	± 0 %	+ 0.5 %	+ 10.5 %	+ 16.7 %	+ 81.0 %

From the results it can be observed that even though the roll model and tyre relaxation have some effect on the high-speed transient off tracking, the largest addition comes from adding the nonlinear tyre model to the simulation model.

7.5 Validation

For validation purposes, Volvo's high fidelity VTM models will be used. The results from VTM simulations will be compared against the roll model with tyre relaxation produced in this thesis. The simulations for the validation are conducted by using a single lane change input to the simulation model. This way the input for all models will be very similar to each other. The input will be a 3-meter lane change with 0.3 Hz input frequency at a speed of 80 km/h. The difference in the input is that VTM simulation models cannot handle a specific lateral acceleration input to the first axle. There is a driver input in the model, that will try to follow the acceleration input as well as possible.

7.5.1 Validation with linear tyres

So that the differences in the non-linear tyre models between the different simulation models are not considered, first validation simulations are done using linear tyre model in Modelica models and in VTM. First, it's necessary to compare the lateral accelerations between the different simulation models. Lateral accelerations are presented in figure 31.

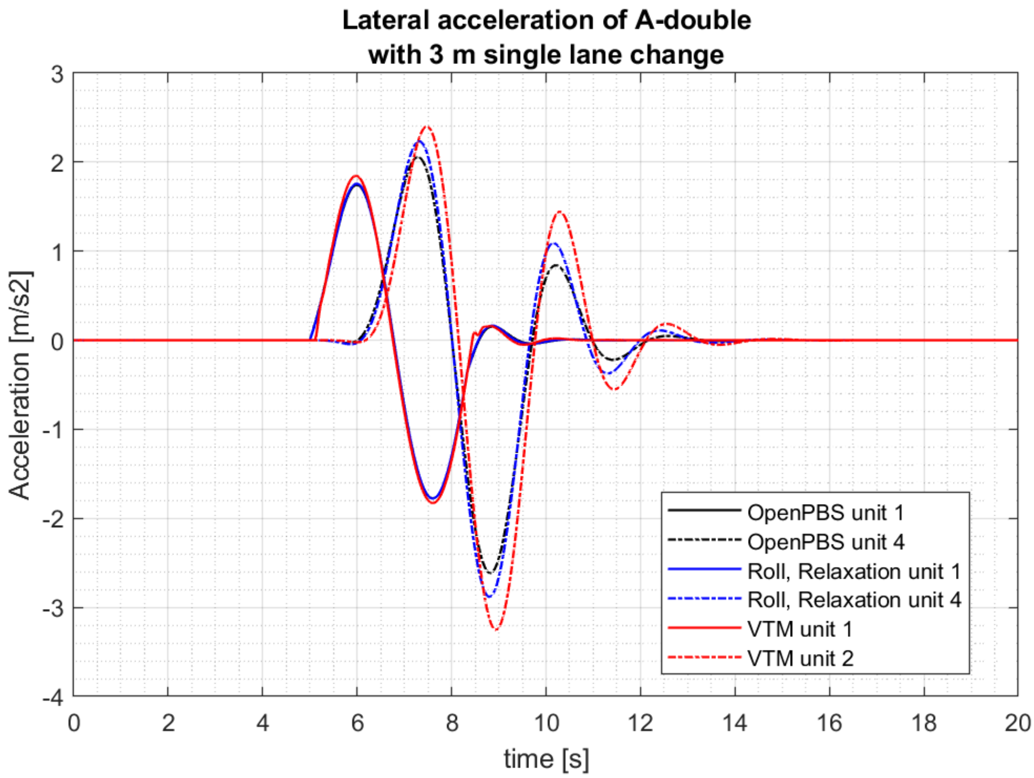

Figure 31. Comparison of lateral acceleration between OpenPBS, Roll model with tyre relaxation and VTM model with linear tyre model in a single lane change.

From the figure, it can be observed, that the previous version of OpenPBS is lacking in the lateral acceleration, the vehicle in simulation is too stable compared to real world and to VTM. The addition of roll dynamics and tyre relaxation helps the lateral acceleration but doesn't solve it completely. The differences in last units' maximum lateral acceleration is presented in table 15.

Table 15. Difference in lateral acceleration compared to VTM with linear tyre models.

	OpenPBS	**Roll model + Tyre relaxation**	**VTM**
Lat.Acc.	2.614	2.881	3.251
Compared to VTM	- 19.6 %	- 11.4 %	± 0 %

Similar results can be obtained from yaw rate results from lane change simulation. Yaw rate results are presented in figure 32.

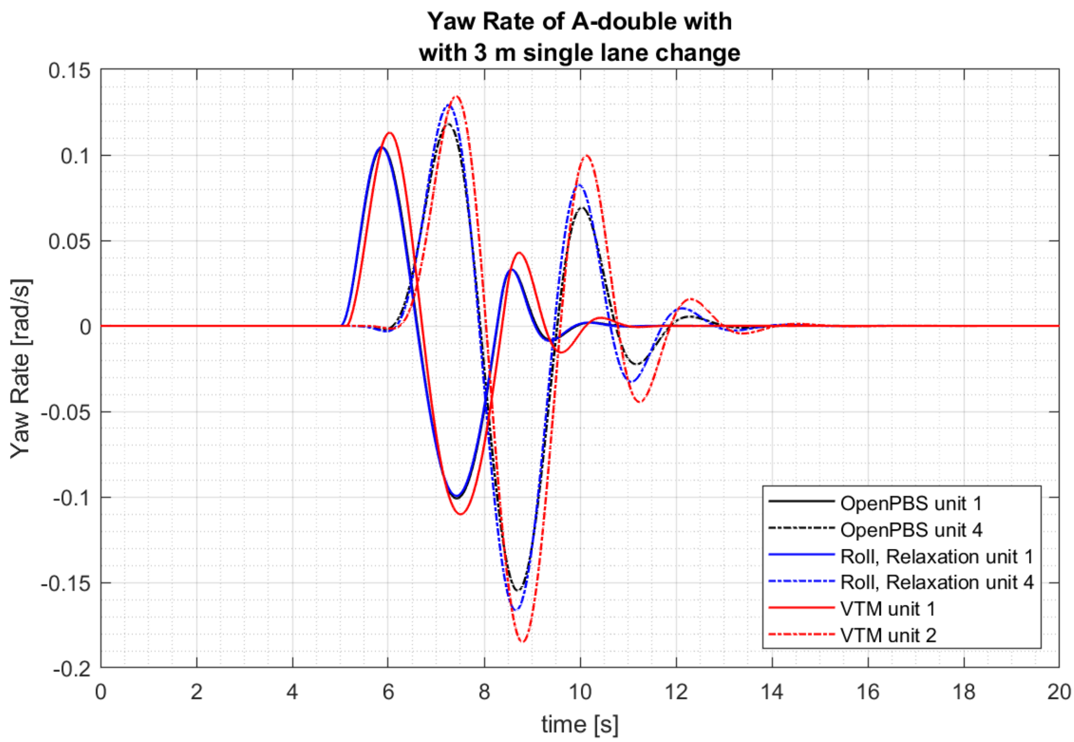

Figure 32. Comparison of yaw rate between openPBS, Roll model with tyre relaxation and VTM model with linear tyre model in a single lane change.

Similarly, to lateral acceleration there is improvement in the roll model with tyre relaxation compared to OpenPBS version 1, but the work is not finished. From yaw rate we can calculate also the rearward amplification of the combination vehicle. The maximum yaw rates of the last unit of the combination vehicle and the rearward amplification are presented in the table 16.

Table 16. Comparison between maximum yaw rate values and rearward amplification of A-double in a single lane change with linear tyre models.

	OpenPBS	Roll model + Tyre relaxation	VTM
Yaw Rate	0.1549	0.1664	0.1849
Compared to VTM	- 16.2 %	- 10.0 %	± 0 %
RWA	1.484	1.593	1.637
Compared to VTM	- 9.3 %	- 2.7 %	± 0 %

From the combination vehicle's trajectory, one can get the comparison values for the high-speed transient off tracking. The vehicle units' trajectories are presented in figure 33.

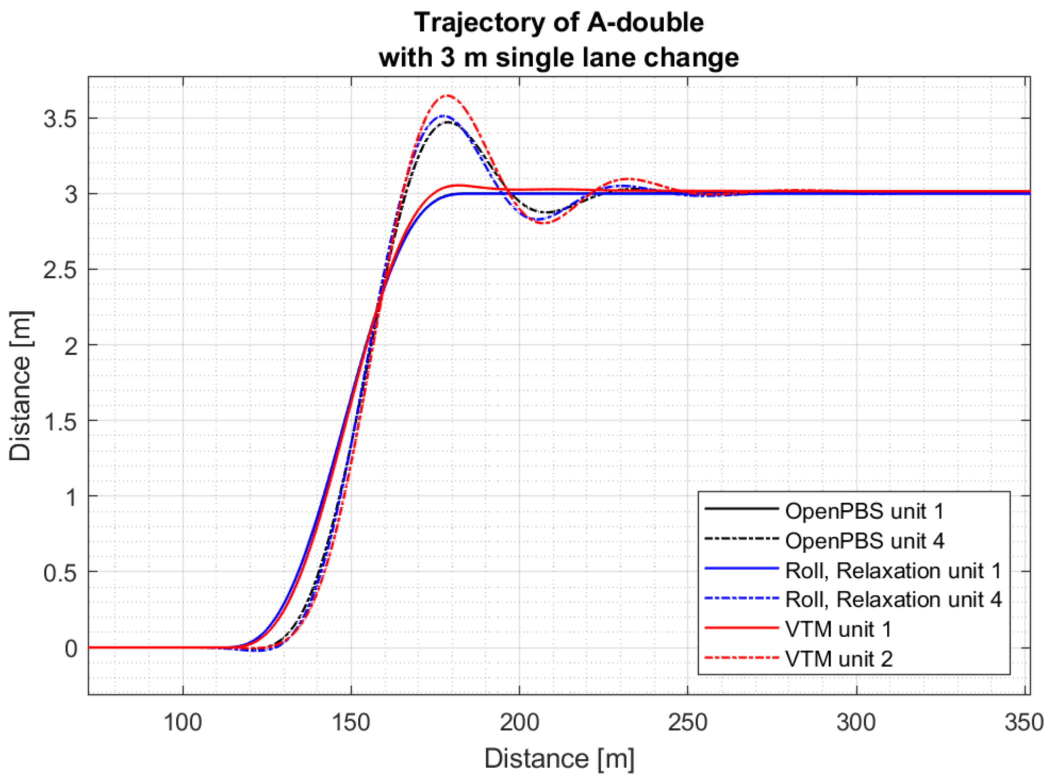

Figure 33. Trajectory of A-double combination vehicle with linear tyre model in a 3 meter wide lane change. Comparison between openPBS, roll model with tyre relaxation and VTM model.

The trajectory represents the trajectory of the first and last axles of combination vehicle. In an A-double that means the first axle of the tractor and the last axle of the second semitrailer. The maximum high-speed transient off tracking values are presented on the table 17.

Table 17. High speed transient off tracking in an A-double combination vehicle in a single lane change with linear tyre models.

	OpenPBS	**Roll model + Tyre relaxation**	**VTM**
HSTO	0.471	0.514	0.619
Compared to VTM	- 23.9 %	- 17.0 %	± 0 %

In high speed transient off tracking the differences between the OpenPBS and Roll model are relatively small compared to VTM. Still, there is an improvement compared to OpenPBS, where the value of the HSTO is almost one quarter of the high-fidelity model's HSTO value.

One of the largest differences between the VTM simulation models and the Modelica models is the stiffness in the fifth wheels. In VTM these couplings are roll stiff and in Modelica models, the couplings don't have any roll stiffness. This can be observed very easily from lateral load transfer, presented in figure 34.

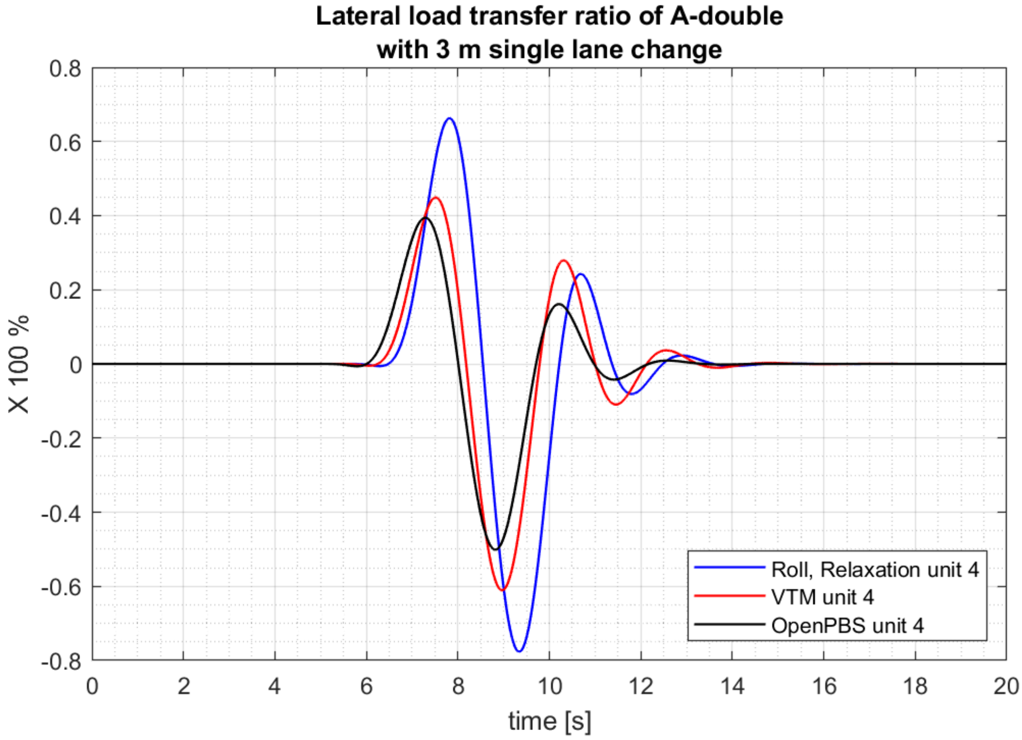

Figure 34. Lateral load transfer in last vehicle unit of an A-double combination vehicle in a single lane change. Comparison between OpenPBS, Roll model with tyre relaxation and VTM model.

The effect of fifth wheel stiffness can be observed from the figure 34 easily. In VTM the lateral load transfer ratio is much less compared to roll model with tyre relaxation. This phenomenon happens because the dolly in the roll model doesn't carry any of the load from the semitrailer in roll. Thus, the combination vehicle seems to be more unstable and near of rollover.

The last key value of vehicle stability was the yaw damping. For this simulation, the yaw damping results are presented in table 18.

Table 18. Yaw damping of A-double combination vehicle in single lane change input with linear tyre models.

	OpenPBS	Roll model + Tyre relaxation	VTM
YD	0.152	0.128	0.113
Compared to VTM	+ 34.5 %	+ 13.3 %	± 0 %

There is a clear development in the yaw damping of the combination vehicle. The yaw damping in OpenPBS is over 30 % higher than it is in the VTM simulation models. With Roll model with tyre relaxation this difference was brought down to 13 %.

7.5.2 Validation with non-linear tyre models

Similar comparison will be done for the simulation models with non-linear tyre models. VTM uses magic tyre model, whereas in simulation models produced in this thesis, the non-linear tyre model is a modified version of the magic tyre model. Similar tyre parameters are used between the two non-linear tyre models. In the comparisons, current version of OpenPBS is used, so the tyre model of the OpenPBS is linear. First comparison is with the lateral acceleration of the vehicle units, presented in figure 35.

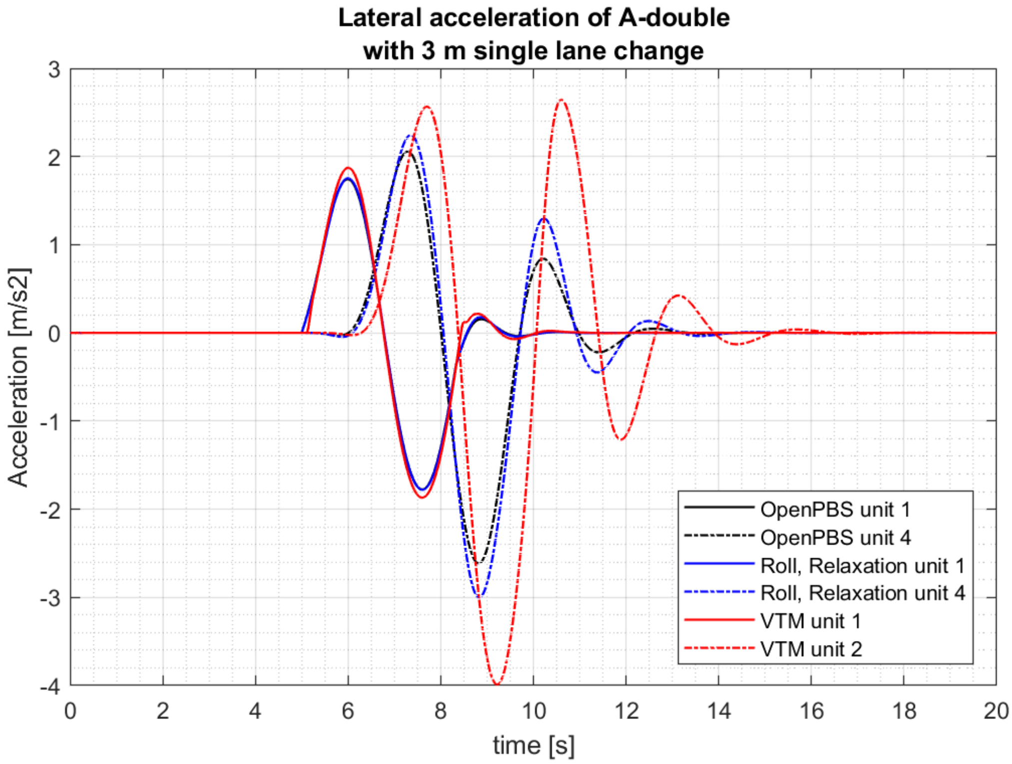

Figure 35. Comparison of lateral acceleration between OpenPBS with linear tyre model, Roll model with non-linear tyre model and VTM model with non-linear tyre model in a single lane change.

From the figure 35 it can be observed, that the magic tyre model has a much more significant effect on the lateral acceleration of the combination vehicle. The results are listed in the table 19.

Table 19. Difference in lateral acceleration compared to VTM with non-linear tyre models.

	OpenPBS	**Roll model + Tyre relaxation**	**VTM**
Lat.Acc.	2.614	3.001	3.988
Compared to VTM	- 34.5 %	- 24.7 %	± 0 %

Even though there is a change in the Modelica models, going from linear tyre model to non-linear tyre model, the change is not as great as it is with VTM, going from linear tyre model to magic tyre model. Similar results can be obtained from yaw rate results from lane change simulation. Yaw rate results are presented in figure 36.

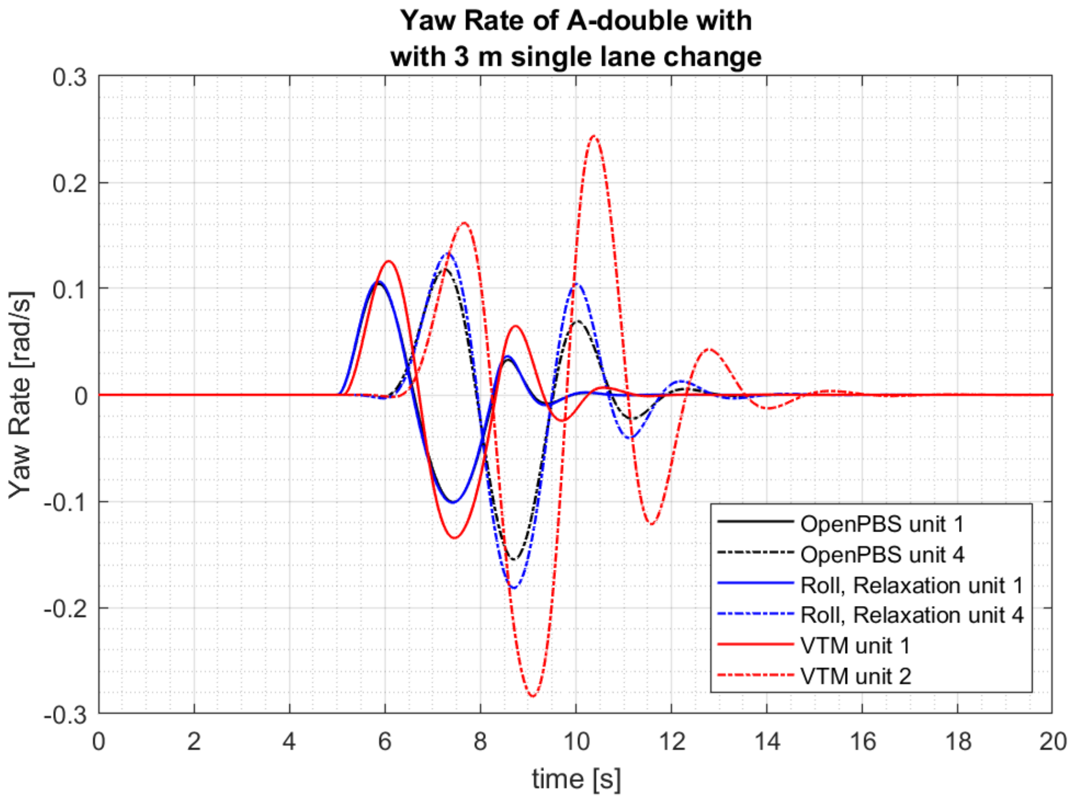

Figure 36. Comparison of yaw rate between openPBS with linear tyre model, roll model with non-linear tyre model and VTM model with non-linear tyre model in a single lane change.

The results are very similar to the lateral acceleration. The change in VTM is much more drastic, compared to Modelica models. The results are shown in the table 20.

Table 20. Comparison between maximum yaw rate values and rearward amplification of A-double in a single lane change with non-linear tyre models.

	OpenPBS	Roll model + Tyre relaxation	VTM
Yaw Rate	0.1549	0.1817	0.2836
Compared to VTM	- 45.4 %	- 35.9 %	± 0 %
RWA	1.484	1.702	2.105
Compared to VTM	- 29.5 %	- 19.1 %	± 0 %

In lateral acceleration, yaw rate and rearward amplification a 10 %-unit improvement can be observed, when comparing to VTM. From the combination vehicle's trajectory, we can get the comparison values for the high-speed transient off tracking. The results are presented in figure 37.

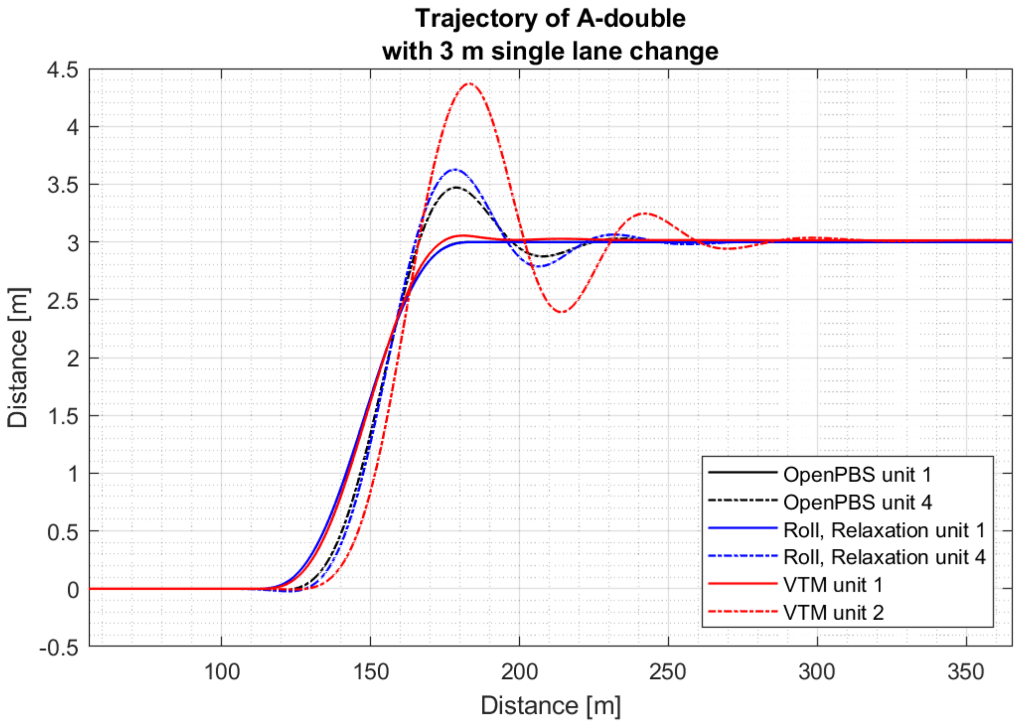

Figure 37. Trajectory of A-double combination vehicle with a non-linear tyre model in a single lane change. Comparison between openPBS with linear tyre model, roll model with non-linear tyre model and VTM model with non-linear tyre model.

With trajectory, similar results can be observed compared to previous key values. The effect of magic tyre model in VTM is much more significant than the effect of the non-linear tyre model in Modelica models. The results are collected to table 21.

Table 21. High speed transient off tracking in an A-double combination vehicle in a single lane change input with non-linear tyre models.

	OpenPBS	**Roll model + Tyre relaxation**	**VTM**
HSTO	0.471	0.628	1.343
Compared to VTM	- 64.9 %	- 53.2 %	± 0 %

In high speed transient off tracking the accuracy of the model is the worst. This also follows the results that Manjurul Islam got in his paper. (Islam, et al., 2019).

Combination vehicle's last units lateral load transfer is presented in figure 38.

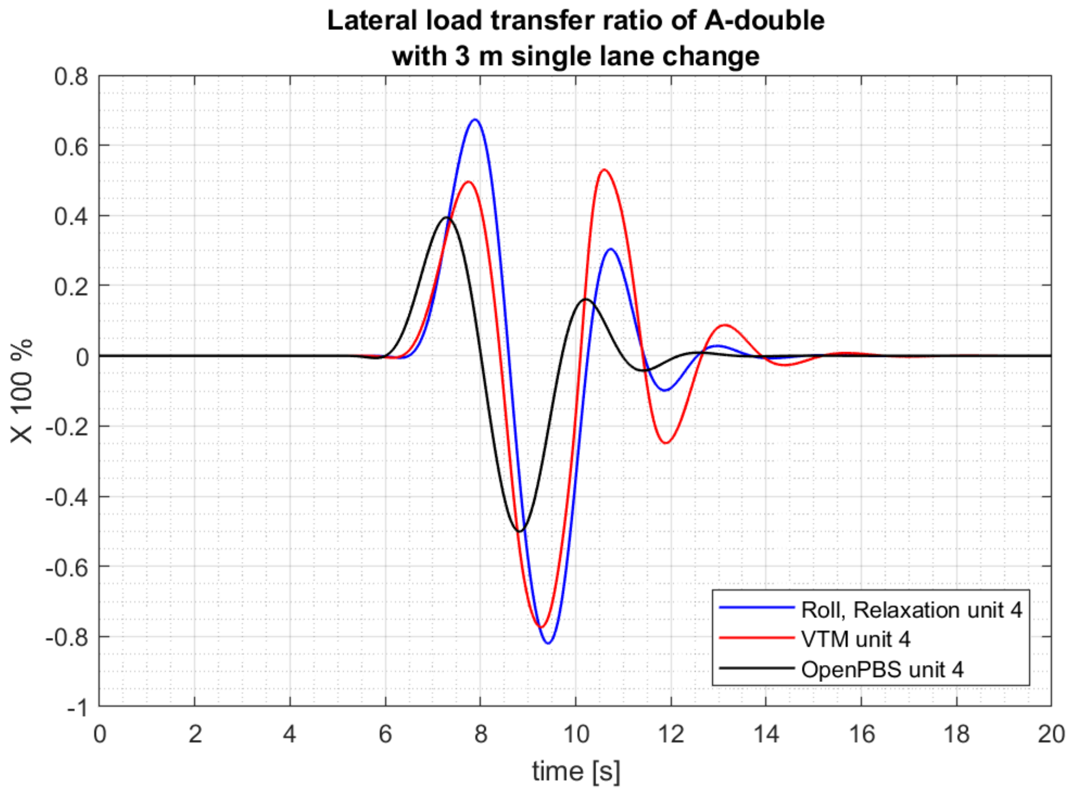

Figure 38. Lateral load transfer in last vehicle unit of A-double combination vehicle in a single lane change. Comparison between openPBS with linear tyre model, roll model with non-linear tyre model and VTM with non-linear tyre model.

Here, the same issue arises with the absence of the roll stiffness in the couplings in Modelica models.

The last key value was the yaw damping of the combination vehicle. The results of the yaw damping are presented in the table 22.

Table 22. Yaw damping of A-double combination vehicle in single lane change input with non-linear tyre models.

	OpenPBS	**Roll model + Tyre relaxation**	**VTM**
YD	0.152	0.118	0.067
Compared to VTM	+ 126.9 %	+ 76.1 %	± 0 %

In yaw damping the differences are very big. When comparing the yaw damping of VTM with non-linear tyre model to OpenPBS with linear tyre model, the damping in VTM is less than half of the damping in OpenPBS. With Roll model with non-linear tyres, there is already a huge improvement, but the model is still lacking.

7.6 Final model implemented in Open PBS

Last part of thesis work was implementing the developed simulation models to OpenPBS environment. This was done in couple of different stages, without touching the original OpenPBS simulation models and manoeuvres. The two different simulation models roll dynamics with tyre relaxation and roll dynamics with non-linear tyre model were implemented to OpenPBS as separate simulation models.

Implementing the new vehicle models to OpenPBS started by creating two new vehicle parameter models to support the vehicle models. These two vehicle parameter models were "VehicleModelRollDynamics" and "VehicleModelRollDynamicsNonLinTyre". There are some differences between the two vehicle parametrizations, and by creating two different models, it's possible to collect parameters and simulate combination vehicles with fewer parameters as the most complex model would need. The vehicle parameter models are shown in the figure 39.

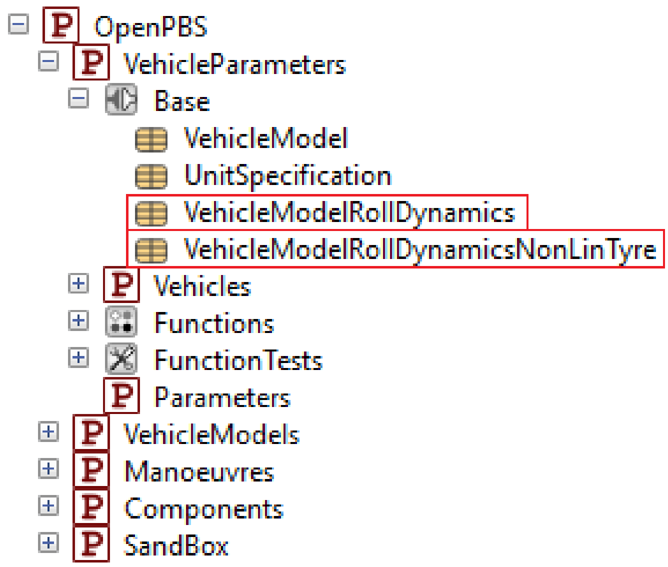

Figure 39. OpenPBS structure with added vehicle parameter models. Added vehicle parameter models are marked red.

Vehicle parameter models contain information of all the vehicle parameters needed in the simulation. The difference in these vehicle parameter models is the amount of the parameters. The original vehicle parameter model "VehicleModel" contains 20 parameters. These include the parameters described in the chapter 6.1 and in addition to those, information of the vehicle's max engine power, rolling resistance coefficient, frontal area and drag coefficient. Compared to the original vehicle parameter model, the new models have additional parameters, based on the additional dynamics in the simulation models. In "VehicleModelRollDynamics" the number of parameters is 26. Additional parameters are described in chapters 6.2 and 6.3. The additional parameters in "VehicleModelRollDynamicsNonLinTyre" bring the total number of parameters to 31. The additional parameters are shown in chapter 6.4.

For test purposes, two new vehicles were added to the OpenPBS environment. These vehicles were added to fullfill the parameter demands of the two added vehicle parameter models. These added vehicles were "Adouble6x4_RollDynamics" and "Adouble6x4_RollDynamics_NonLinTyre". These are shown in figure 40.

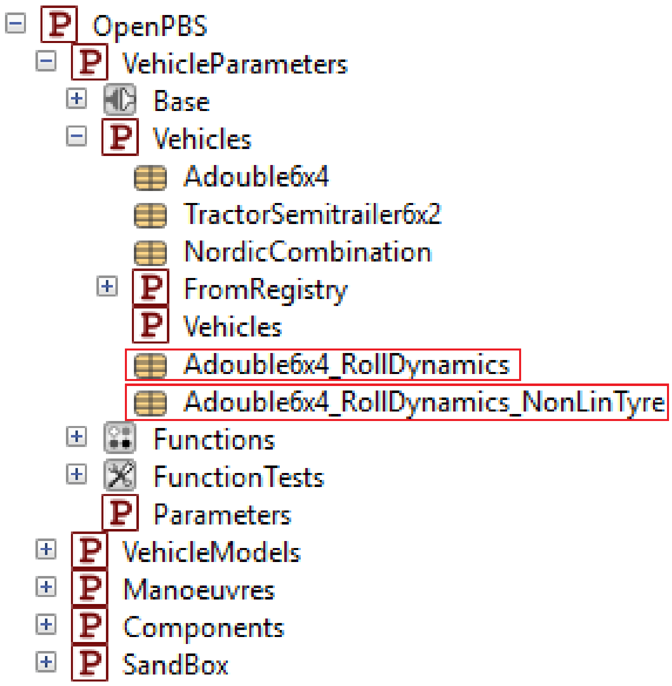

Figure 40. OpenPBS structure with added vehicles. Added vehicles are marked red.

As the names suggest "Adouble6x4_RollDynamics" is used to simulate with roll dynamics and tyre relaxation and "Adouble6x4_RollDynamics_NonLinTyre" is used to simulate with roll dynamics and non-linear tyre model. The parameters for these vehicles were obtained from Volvo's VTM simulation environment.

Next additions to OpenPBS environment were the vehicle models containing the equations to calculate the vehicle behaviour. Two vehicle models were added, called "SingleTrackRollDynamics" and "SingleTrackRollDynamicsNonLinTyre". The vehicle models differ from each other, so two different vehicle models were needed. Although in following updates to OpenPBS these vehicle models can be combined and with an additional parameter (e.g. "mode") select which vehicle model to use, it was not necessary for this stage of OpenPBS. Additional vehicle models are shown in figure 41.

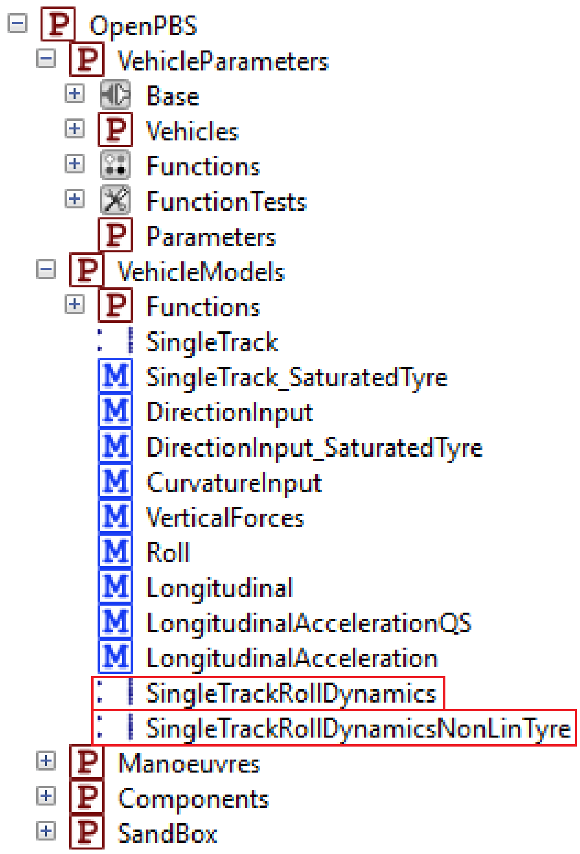

Figure 41. OpenPBS structure with added vehicle models. Added vehicle models marked red.

In comparison to the original “SingleTrack” model there are quite a lot of additions in the new vehicle models. “SingleTrackRollDynamics” vehicle model contains single track model that is reinforced with roll dynamics and tyre relaxation. “SingleTrackRollDynamicsNonLinTyre” vehicle model has roll dynamics, tyre relaxation and non-linear tyre model added to the basic single-track model.

Final addition to OpenPBS environment was the addition of two manoeuvres that use the single-track models described previously. The added manoeuvres are named so that it describes manoeuvres well. “SingleLaneChangeRollDynamics” and “SingleLaneChangeRollDynamicsNonLinTyre” were added to the OpenPBS environment. Compared to the original OpenPBS manoeuvre “SingleLaneChange” the only addition to the manoeuvre is the addition of lateral load transfer as a simulation output. They are also like each other, only difference being the use of different vehicle model. Again, in future updates of OpenPBS these manoeuvres can be combined to single manoeuvre, but it was not deemed necessary at this point. The added manoeuvres are shown in figure 42.

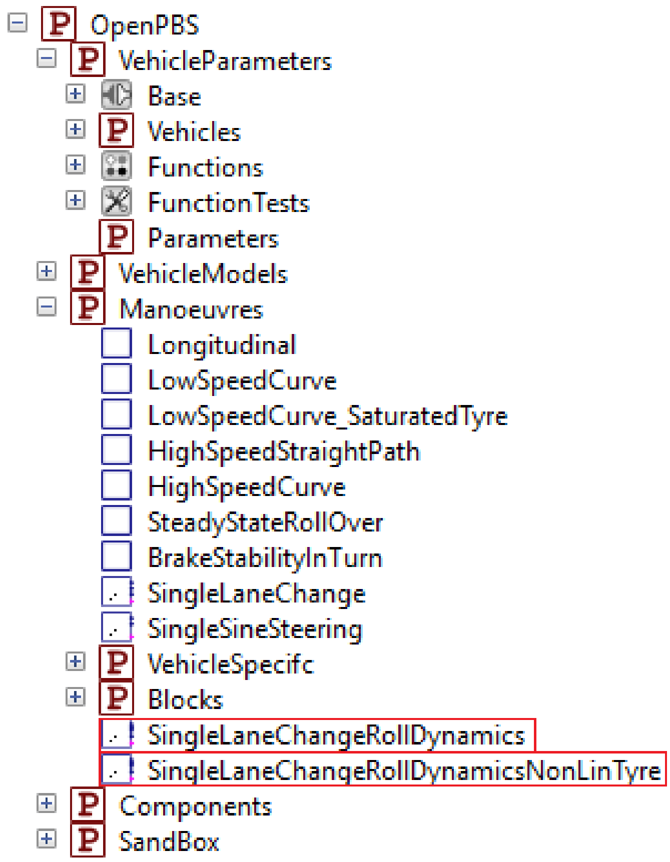

Figure 42. OpenPBS structure with added manoeuvres. Added manoeuvres are marked red.

With these additions to OpenPBS environment, simulation models with roll dynamics, tyre relaxation and non-linear tyre model can now be simulated in the OpenPBS environment.

8 Conclusions and Future work

This master's thesis was done during time period of February 2019 to July 2019 in Chalmers University of technology, with support also from Volvo Trucks and University of Oulu. In this chapter the conclusions of this thesis and the future work after the thesis are presented.

8.1 Conclusions

To better capture the influence of the centre of gravity height, roll dynamics were added to the simple single track model to increase the accuracy of the model. Tyre relaxation was added to the simulation model as a simple first order filter to add a delay to the force generation of the tyre. These improvements to the simulation model effected the selected key values in positive way, improving the accuracy of the simulation model. When comparing the simulation model produced in this thesis to the original OpenPBS simulation model the following results were obtained:

- RWA, increase from 1.48 to 1.59
- HSTO, increase from 0.47 to 0.51
- LLT, increase from 0.45 to 0.79
- YD, decrease from 0.15 to 0.13

There was improvement in the accuracy of all the key values used for evaluating the stability of combination vehicle. In some areas, like high speed transient off tracking, the difference is not that magnificent, but still noticeable. With lateral load transfer, the increase is big, but increase of that magnitude has to do with the simplifications made in the vehicle model (5th wheel roll stiffness).

A non-linear tyre model produced for this thesis by the "performance based standards II" -project was also implemented to the updated vehicle model. The implemented non-linear tyre model is a modified magic tyre model, designed for lateral stability estimation of heavy combination vehicles operated at dry paved road surface well below peak friction utilization. The addition of non-linear tyre model allowed the following results to be obtained, when comparing to original OpenPBS:

- RWA, increase from 1.48 to 1.70
- HSTO, increase from 0.47 to 0.63
- LLT, increase from 0.45 to 0.81
- YD, decrease from 0.15 to 0.12

With non-linear tyre model the accuracy of the vehicle models is increased again. Comparing these two results, one can notice that roll dynamics has the largest effect on lateral load transfer and the non-linear tyre model has the largest effect on high speed transient off tracking.

These phenomenas were modelled to the vehicle model using equations from Bengt Jacobson and Volvo Trucks. The number of parameters went from original OpenPBS with 20 to 26 with tyre relaxation and roll dynamics and to 31 with the non-linear tyre model. Added equations are described in chapter 5.

The simulation models were validated against Volvo's VTM library to see the state of the updated vehicle model in comparison to high fidelity simulation models. With linear tyre model, the updated vehicle model was able to cut the difference (OpenPBS vs. VTM) in half in most of the situations and key values. But with non-linear tyre model there is still a lot of difference when comparing to VTM.

In the last part of the thesis, the finalized vehicle models were integrated to OpenPBS environment. In current state, OpenPBS is now capable of simulating vehicle models with roll dynamics, tyre relaxation and non-linear tyre model.

8.2 Future work

There are still a lot of things to do regarding the development of the OpenPBS tool. On this thesis subject it seems that at least the non-linear tyre model or the tyre model parameters need work. The difference between high fidelity model (VTM) and simulation model with non-linear tyre model produced in this thesis was still quite large, even though the development in the key values was going to the right direction. Other thing regarding the development of this OpenPBS tool is the addition of additional equations, parameters etc. to get more physical phenomenas modelled to the simulation model. These might include for example frame flexibility, axle roll degree of freedom and coupling roll stiffness. With the findings in this thesis work at least the addition of coupling roll stiffnesses to 5th wheel couplings are an important addition to get realistic results especially in lateral load transfer. During the thesis work also some work was done outside the preliminary thesis work requirements. During these 6 months, there was an effort trying to add frame flexibility to the simulation model, but unfortunately because of time and resource limitations, fully working models were not completed in time for this thesis. The results can be found in the appendix 1.

Rearward amplification will change with lateral acceleration's peak value, especially if tyre models are non-linear in slip. One way to eliminate this variation is to define RA for small disturbances. It assumes that the model can be linearized, which is the case for all model versions in 6.1-6.4. The computation would then be a frequency analysis, as opposed to a time simulation. Many Modelica tools can linearize a model, so this is quite doable. Which measure to use should be decided from which traffic risk RA is motivated.

The current state of the OpenPBS is still a very simplified version of description of vehicle dynamics. There are a lot of assumptions made, regarding axle loads, coupling loads, forces from suspension and so on. These all assumptions should be checked and evaluated, what modifications could be made to better capture the vehicle dynamics in the simulation model.

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10 Appendix

1. A look into frame flexibility simulation model
2. Vectorized presentation of tractor-semitrailer combination vehicle's Single-track model
3. Vectorized presentation of tractor-semitrailer combination vehicle's Static axle loads
4. Vectorized presentation of tractor-semitrailer combination vehicle's Roll dynamics
5. Vehicle parameters for simulations
6. Vehicle parametrization description

Appendix 1, Frame Flexibility

Frame flexibility has a effect on the vehicle stability. In this thesis some time was used for creating a simulation model with frame flexibility included. Unfortunately, because of time restraints and some problems with Volvo's VTHAP simulation models, the simulation models with frame flexibility were not included in the thesis. Because of interest in the subject and time used for creating simulation models, this subject and the findings that were obtained during this thesis will be disclosed in this appendix.

To get an idea of frame flexibilitys effect on vehicle stability, Volvos VTM simulation models were used. Results from VTM models regarding frame stiffness effect on PBS values like rearward amplification and load transfer are described in the following tables. Simulations done within this project by Volvo GTT to help evaluate the necessity of introducing frame flexibility to OpenPBS simulation model. Original simulation results received from Volvo GTT in the end of this appendix. The results are presented in tables 1 and 2. Results are also presented in graphical form in figures 1 and 2.

Table 1. Frame stiffness effect on PBSs, A-double

A-double			
Lane change maneuvre	**Soft frame**	**Medium frame**	**Rigid frame**
Stiffness Tractor	50 kNm/rad	380 kNm/rad	20 000 kNm/rad
Stiffness Trailer	500 kNm/rad	4 500 kNm/rad	15 000 kNm/rad
RA_{ω}	2.27	2.18	2.12
RA_{a}	2.42	2.37	2.27
Off-tracking	0.62 m	0.58 m	0.55 m
Load transfer ratio	72 %	70 %	67 %
Yaw damping	0.24	0.26	0.26

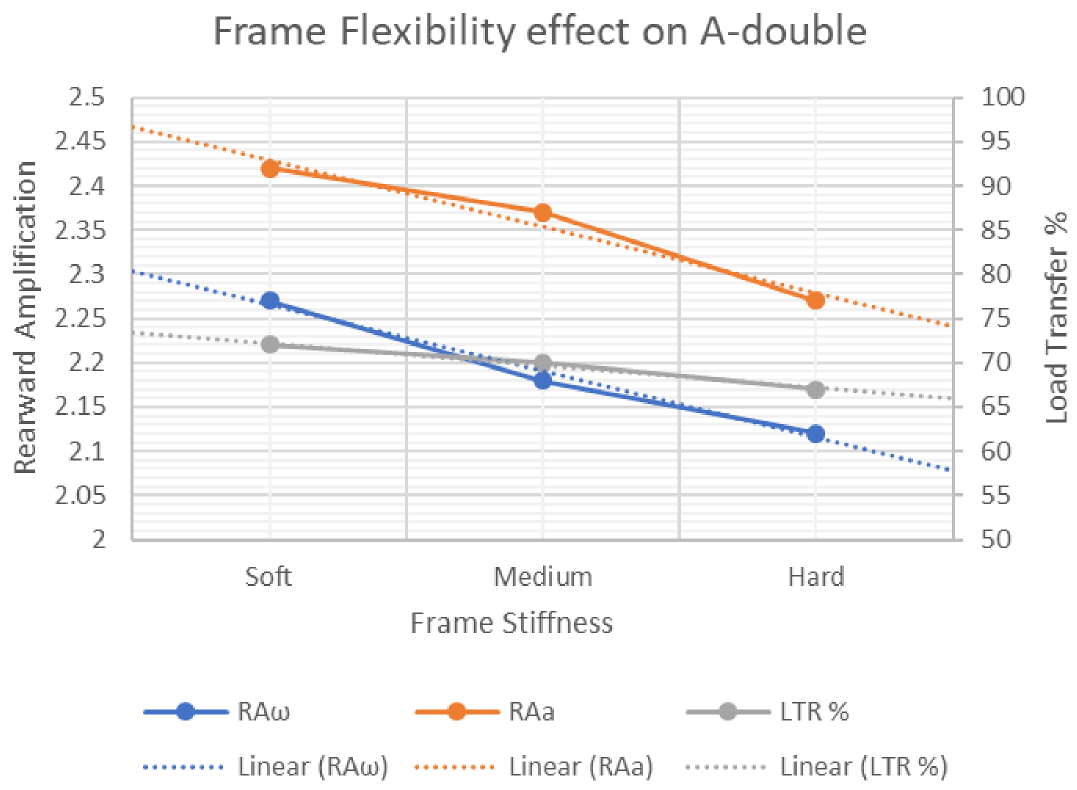

Figure 1 Frame Flexibility A-double

From the table 1 and figure 1 can be seen that increasing the frame stiffness will lower the rearward amplification and lateral load transfer values in a A-double vehicle combination.

Table 2. Frame stiffness effect on PBSs, AB-double

AB-double			
Lane change maneuvre	**Soft frame**	**Medium frame**	**Rigid frame**
Stiffness Tractor	757 kNm/rad	757 kNm/rad	5 000 kNm/rad
Stiffness Trailer	500 kNm/rad	4 500 kNm/rad	15 000 kNm/rad
RA_{ω}	1.75	1.75	1.64
RA_{a}	2.34	2.43	2.17
Off-tracking	0.71 m	0.69 m	0.62 m
Load transfer ratio	57 %	56 %	52 %
Yaw damping	0.29	0.33	0.34

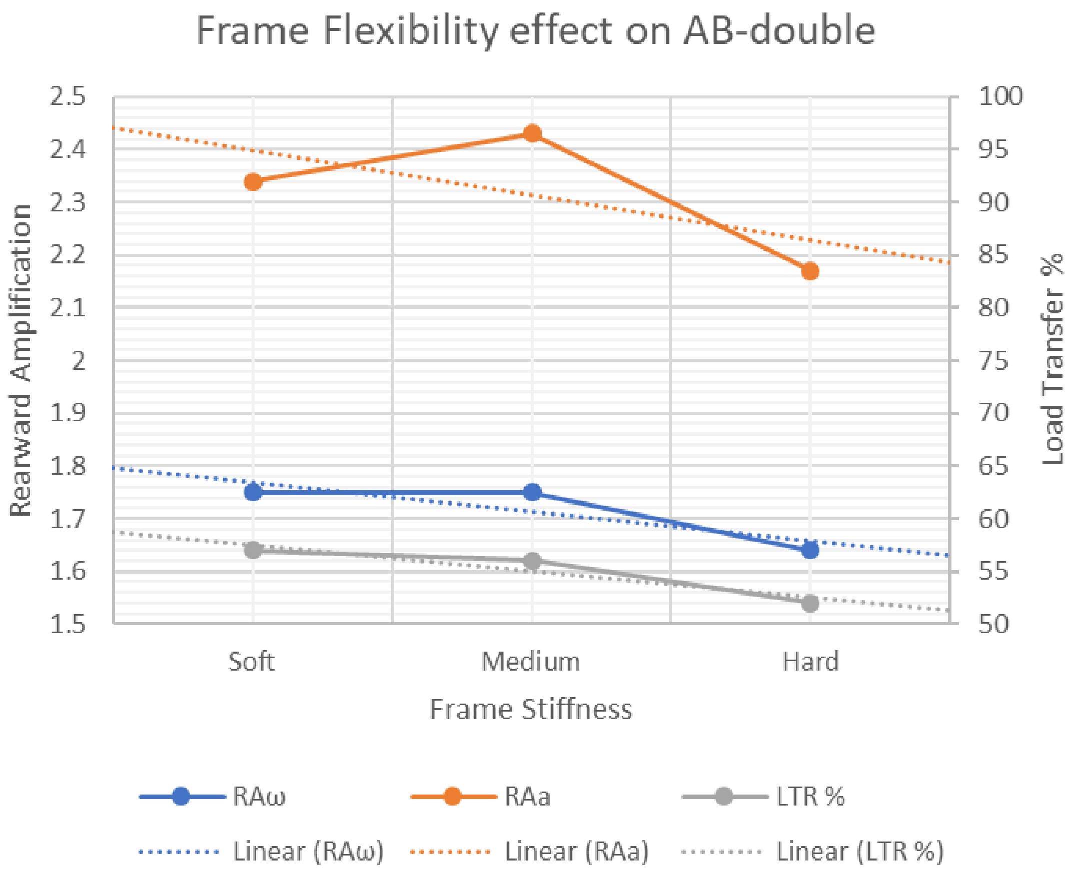

Figure 2. Frame flexibility AB-double

Similar results can be obtained from the AB-vehicle combination. Adding stiffness to the frames of vehicle units, will decrease the rearward amplification and lateral load transfer values.

Description of new vehicle model

Adding frame flexibility to the simulation model adds a new level of complexity to the vehicle model. There is some parameters that need to be added for the simulation model to be functional. Tyre model in this vehicle model is linear, but with added relaxation. Volvo's VTHAP simulation models were used as a base when designing the simulation models.

Compared to other vehicle models produced in this thesis, simulation model with roll dynamics and tyre relaxation, frame flexibility model to be functional you need to add 7 more parameters. Parameters needed are presented in table 3.

Table 3. Additional parameters to add Frame Flexibility to simulation model

Parameter	Description
b_r	Frame rail distance from each other. *e.g. "[0.85,1.0]"*
h_{frame}	Frame height from ground. *e.g. "[0.69,0.89]"*
k_f	Frame stiffness. *e.g. "[2.0e6,295e6]"*
c_f	Frame dampening. *e.g. "[2.0e5,26.0e6]"*
k_t	Tyre stiffness on axle. *e.g. "[2.45e6,2.86e6,2.86e6;2.36e6, 2.36e6, 2.36e6]"*
k_{cr}	5th wheel stiffness. *e.g. "[1e9]"*
c_{cr}	5th wheel dampening. *e.g. "[0]"*

Equations needed to simulate frame flexibility in the simulation models are defined in the paper produced by Volvo Trucks. (Volvo Group Trucks Technology, 2001. *Theory description of a generic vehicle combination model for stability studies*). This simulation model includes also 5th wheel stiffness and axle roll degree of freedom. Simulation model variables are described in figures 3 and 4.

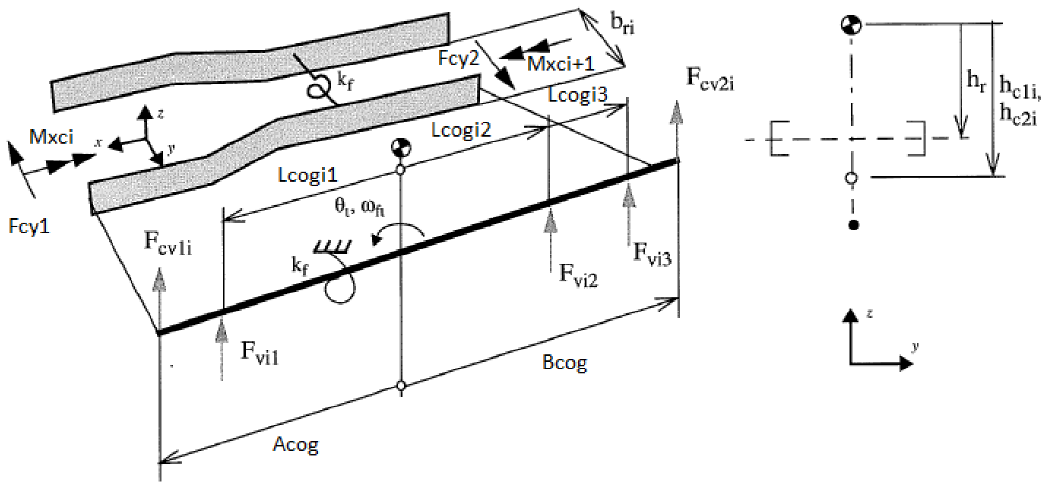

Figure 3. Frame model when calculating frame flexibility (Modified from Volvo GTT's report)

In figure 3 there are two frame rails which are connected to eachother by torsional spring. There are forces and torques from couplings and axles affecting the frame.

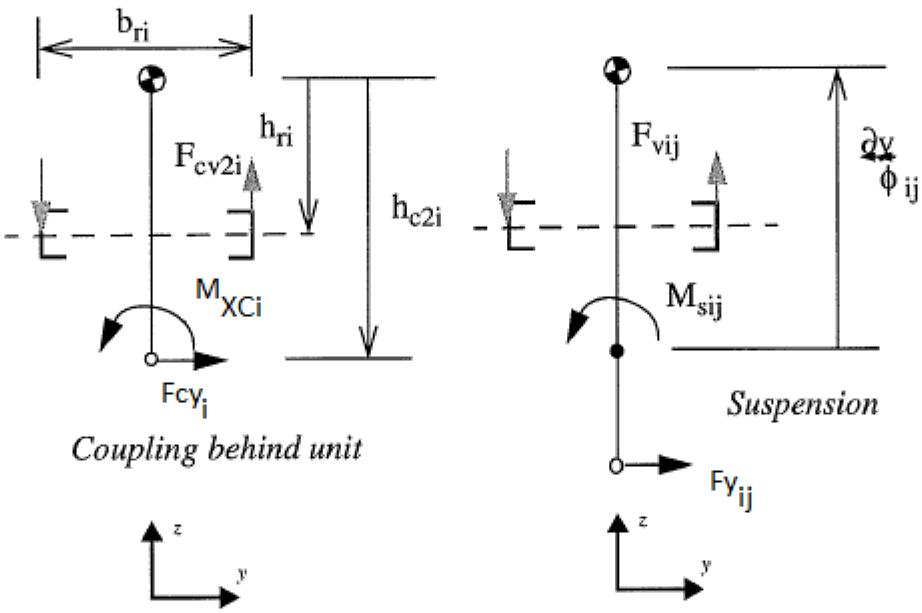

Figure 4. Forces affecting on the frame rails (Modified from Volvo GTT's report)

In figure 4 there is a description of the forces affecting the frame rails.

The following set of equations is used to describe roll dynamics and frame flexibility in simulation model.

Springs in series:

$$e_{ij} = \frac{1}{1 + \frac{k_{s_{ij}}}{k_{t_{ij}}}} \tag{1}$$

Where k_s is the suspension roll stiffness and k_t is the tyre stiffness

Kinematic relation between the roll at CoG and roll at suspension

$$p_{s_{ij}} = p_{x_i} - \frac{2 \cdot L_{cog_{ij}}}{b_{r_i}} \cdot \theta_{f_i} \tag{2}$$

Where p_s is the suspension roll angle, p_x is the is roll angle at CoG, L_{cog} is the axle position from CoG, b_r is the frame rail distance from each other and θ_f is the frame rail angle compared to each other

Torque created by suspension:

$$M_{s_{ij}} = k_{s_{ij}} \cdot e_{ij} \cdot p_{s_{ij}} \tag{3}$$

Roll at the front and rear couplings:

$$p_{c1_i} = p_{x_i} - \frac{2 \cdot A_{cog_i}}{b_{r_i}} \cdot \theta_{f_i}$$

And (4)

$$p_{c2_i} = p_{x_i} - \frac{2 \cdot B_{cog_i}}{b_{r_i}} \cdot \theta_{f_i}$$

Where A_{cog} is the front coupling position from CoG and B_{cog} is the rear coupling position from CoG.

Roll moment in couplings:

$$M_{xc_i} = -C_{cr_i} \cdot \left(\dot{p_{c2_i}} - \dot{p_{c1_{i+1}}}\right) - K_{cr_i} \cdot \left(p_{c2_i} - p_{c1_{i+1}}\right) \tag{5}$$

Where C_{cr} is the coupling roll damping and K_{cr} is the coupling roll stiffness.

Front and rear coupling loads on the frame:

$$F_{cv1_i} = \frac{-M_{xc_{i-1}} + F_{cy_{i-1}} \cdot \left(h_{c2_i} - h_{r_i}\right)}{b_{r_i}}$$

And (6)

$$F_{cv2_i} = \frac{M_{xc_i} - F_{cy_i} \cdot \left(h_{c2_i} - h_{r_i}\right)}{b_{r_i}}$$

Where F_{cy} is the lateral coupling force, h_{c2} is the rear coupling height and h_r is the frame height from ground

Suspension loads on the frame:

$$F_{v_{ij}} = \frac{M_{s_{ij}} + F_{y_{ij}} \cdot (h_i - h_{r_i})}{b_{r_i}} \tag{7}$$

Where F_y is the lateral tyre force and h is the CoG height.

Roll equation:

$$I_{x_i} \cdot \dot{\omega_x} = -\sum_j K_{s_{ij}} \cdot e_{ij} \cdot p_{s_{ij}} - \sum_j C_{s_{ij}} \cdot e_{ij} \cdot \dot{p_{s_{ij}}} + \sum_j F_{y_{ij}} \cdot h + \sum_j m_i \cdot g \cdot \left(h_i - h_{RC_{ij}}\right) \cdot p_{x_i} + h_{ci_2} \cdot F_{cy_i} - h_{cv1_i} \cdot F_{cy_{i-1}} + M_{xc_i} - M_{xc_{i-1}} \tag{8}$$

Where I_x is the inertia in roll plane, ω_x is the roll rate, C_s is the roll damping, m is the mass of the units, g is the fall acceleration and h_{RC} is the roll center height.

Frame roll equation:

$$I_{y_i} \cdot \dot{\omega_{f_i}} = k_{f_i} \cdot \theta_{f_i} - c_{f_i} \cdot \omega_{f_i} - F_{cv1_i} \cdot A_{cog_i} - F_{cv2_i} \cdot B_{cog_i} - \sum_j F_{v_{ij}} \cdot L_{cog_{ij}} \tag{9}$$

Where I_y is the frame roll inertia, ω_f is the frame roll rate, k_f is the frame stiffness and c_f is the frame damping.

Simulation

Simulation results for vehicle model with frame flexibility, roll dynamics and tyre relaxation are presented in figure 5.

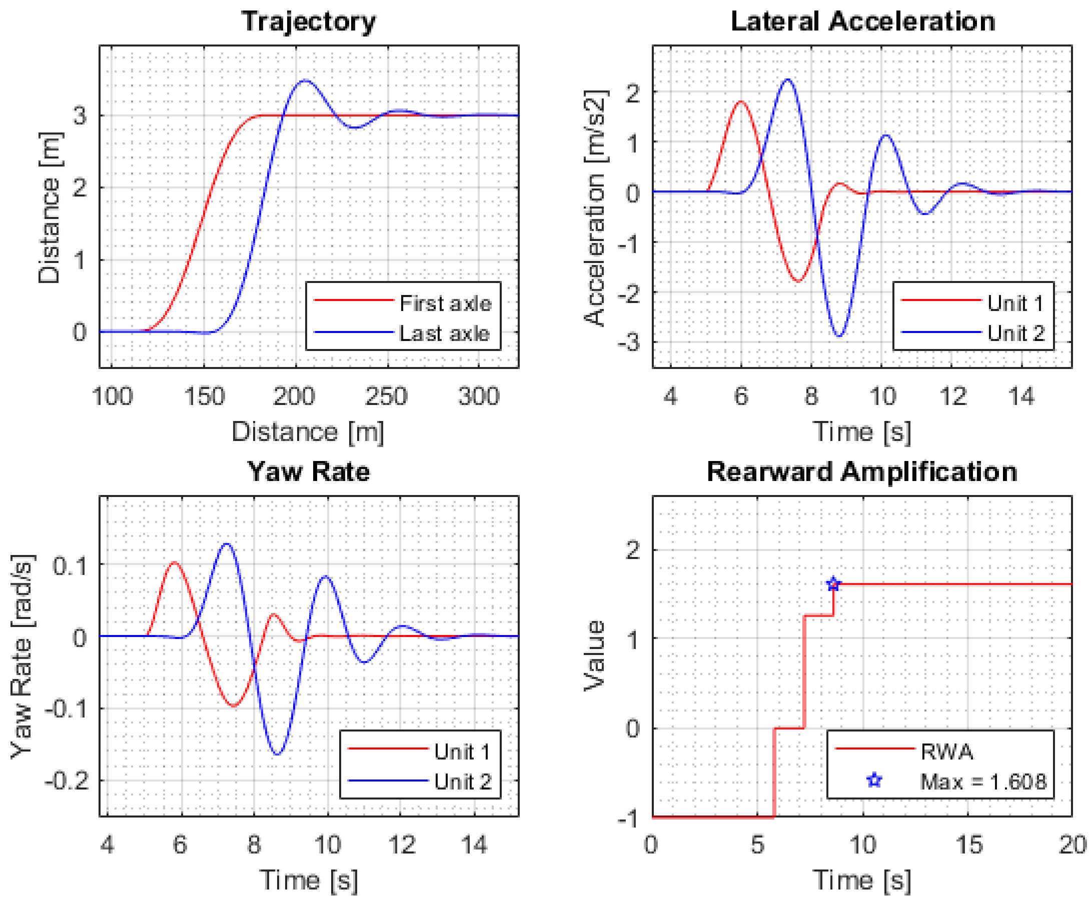

Figure 5. Simulation results for A-double in a single lane change, with Frame Flexibility, Roll dynamics and Tyre relaxation

Results from simulation are presented in table 4.

Table 4. Simulation results of A-double with frame flexibility

Simulation results	**Max value of 1st unit**	**Max value of last unit**
Trajectory	3 m	3.48 m
Lateral Acceleration	-1.77 m/s^2	-2.89 m/s^2
Yaw Rate	-0.1 rad/s	-0.16 rad/s
Rearward Amplification	1.61	

Simulation results with frame flexibility look relatively promising. For example when comparing to previous simulation model, with tyre relaxation and roll dynamics, the rearward amplification is increased. But this is not the the whole case as it was later found out.

Problems with frame flexibility model

During validation stage of the frame flexibility model, some problems were found with the model. The first problem with frame flexibility simulation model is the behaviour of key values relative to the frame stiffness. From Volvo's VTM simulations, it can be determined that when frame stiffness increases, for example rearward amplification has to go down in value. In closer look, the new simulation model with frame flexibility actually works the opposite. This error might be, for example, caused by a mistake in the simulation model. Behaviour shown in the figure 6.

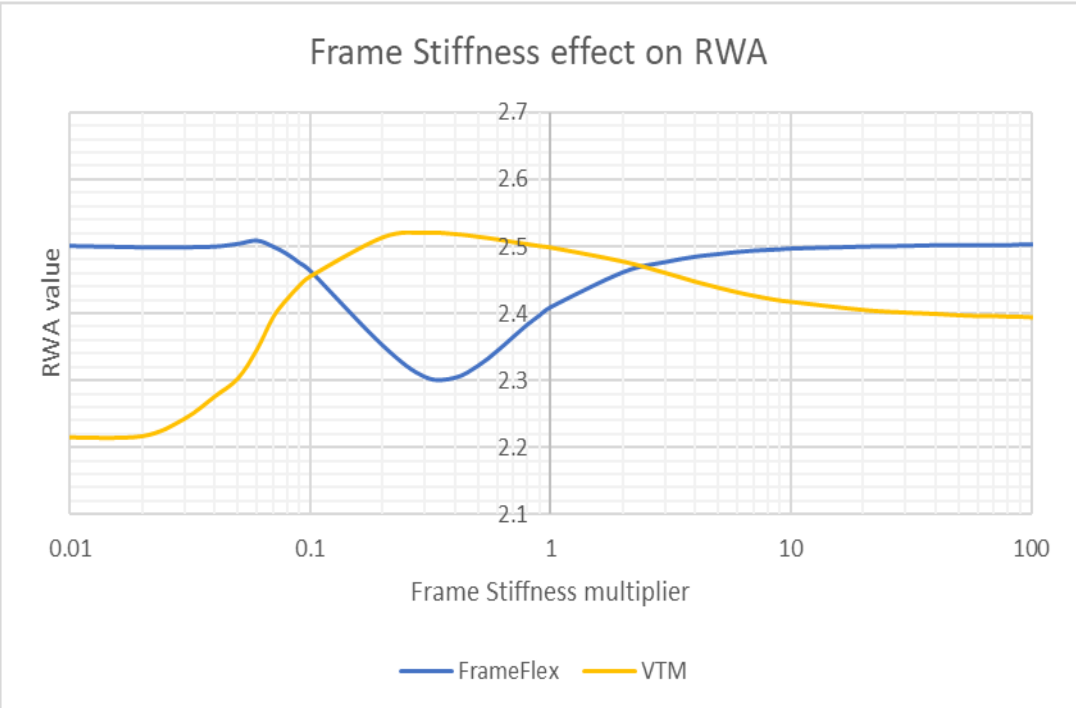

Figure 6. Frame Stiffness effect on rearward amplifications

From the figure 6, the opposite reaction to frame stiffness can be seen. Other problem with the simulation model was noticed when trying to debug the simulation model. During debug stage an attempt was made to compare the simulation model to Volvo's VTHAP simulation models, which are the base for this frame flexibility simulation model. It was discovered that VTHAP cannot actually handle four or more units.

Conclusions

There is obviously potential in this simulation model with frame flexibility. The added parameter amount is not that huge and it also adds the required coupling roll stiffnesses to the simulation model. The potential can be noticed with the added rearward amplification even with malfunctioning simulation model. Still, the simulation model with frame flexibility is all in all pretty complex and not that easily understood, which might fight against the basic idea with OpenPBS. But from these simulation models at least the coupling roll stiffness and axle roll degree of freedom could be easily adapted to current OpenPBS for added accuracy and complexity. Adding these phenomena to OpenPBS would require three new parameters to add to OpenPBS, tyre vertical stiffness, coupling roll stiffness and coupling roll damping. One way of approaching the frame flexibility might be to look for some simplification of the frame flexibility equations.

Original VTM simulation results obtained from Volvo GTT/Niklas Fröjd

Email from Niklas Fröjd:

Hej Ville etc

These are sample comparisons on a normal A-double done in VTM software at Volvo, to see how much influence frame torsional stiffness might have on PBS-measures.
Kf is normal rotX stiffness on trailing units. Corresponding stiffness for the tractor is found by taking bthe kfs value divided with the wheelbase which is in this case about 4 m.

This to be followed up by an AB-double combination (truck, dolly, link,semi)

Niklas

SOFT SETTING –A Double
Kfs Tractor (Soft setting): 50000.0Nm/rad
Kf trailer(Soft setting): 500000.0Nm/rad
Lane change yaw-rate amplification of last unit: 2.27
Lane change acceleration amplification of last unit: 2.42
Lane change off-tracking: 0.62 m
Lane change load transfer ratio: 72%
Yaw damping of last unit: 0.24

MEDIUM SETTING –A Double
Kfs Tractor (Medium setting): 380000.0Nm/rad
Kf trailer(Medium setting): 4500000.0Nm/rad
Lane change yaw-rate amplification of last unit: 2.18
Lane change acceleration amplification of last unit: 2.37
Lane change off-tracking: 0.58 m
Lane change load transfer ratio: 70%
Yaw damping of last unit: 0.26

HARD SETTING –A Double
Kfs Tractor (Hard setting): 20000000.0Nm/rad
Kf trailer(Hard setting): 15000000.0Nm/rad
Lane change yaw-rate amplification of last unit: 2.12
Lane change acceleration amplification of last unit: 2.27
Lane change off-tracking: 0.55 m
Lane change load transfer ratio: 67%
Yaw damping of last unit: 0.26

Email from Niklas Fröjd:

Hi Ville, Bengt, Miro-Tommi

Here is also the comparison between different frame compliance levels of an AB-double (TK+DY+LINK+SEMI).

BR Niklas

From: Kashampur Krishna (Consultant)
Sent: den 25 mars 2019 1:14
To: Fröjd Niklas
Subject: RE: Sensitivity study AB-double

Hej Niklas,

Updated Results for AB-double

SOFT SETTING –AB Double
new_34_05_TK3_DY2_LT2_ST3_GeneralCargo_EvenLoad
Application: general cargo
Kf Truck (Soft setting): 757000.0Nm/rad
Kf trailer(Soft setting): 500000.0Nm/rad
Trailer tyres: new Mixed
Max load height any unit: 4.4 m
Gross combination mass: 74.0 ton
Total wheel base: 29.3 m
Total length: 33.4 m
Inner radius in 90 deg R12.5m turn: 4.08 m
Max outboard off-tracking in 90 deg R12.5m turn: 0.35 m
Lane change yaw-rate amplification of last unit: 1.75
Lane change acceleration amplification of last unit: 2.34
Lane change off-tracking: 0.71 m
Lane change load transfer ratio: 57%
Yaw damping of last unit: 0.29
Max straight line off-tracking at 0.19 m
Load on assumed drive axles: 20.7%
Pressed load on assumed drive axles: 22.8%

MEDIUM SETTING –AB Double
new_34_05_TK3_DY2_LT2_ST3_GeneralCargo_EvenLoad
Application: general cargo
Kf Truck (Medium setting): 757000.0Nm/rad
Kf trailer(Medium setting): 4500000.0Nm/rad
Trailer tyres: new Mixed
Max load height any unit: 4.4 m
Gross combination mass: 74.0 ton
Total wheel base: 29.3 m
Total length: 33.4 m
Inner radius in 90 deg R12.5m turn: 4.08 m
Max outboard off-tracking in 90 deg R12.5m turn: 0.35 m
Lane change yaw-rate amplification of last unit: 1.75
Lane change acceleration amplification of last unit: 2.43
Lane change off-tracking: 0.69 m
Lane change load transfer ratio: 56%
Yaw damping of last unit: 0.33
Max straight line off-tracking at 0.19 m
Load on assumed drive axles: 20.7%
Pressed load on assumed drive axles: 22.8%

HARD SETTING –AB Double
new_34_05_TK3_DY2_LT2_ST3_GeneralCargo_EvenLoad
Application: general cargo
Kf Truck (Hard setting): 5000000.0Nm/rad
Kf trailer(Hard setting): 15000000.0Nm/rad
Trailer tyres: new Mixed
Max load height any unit: 4.4 m
Gross combination mass: 74.0 ton
Total wheel base: 29.3 m
Total length: 33.4 m
Inner radius in 90 deg R12.5m turn: 4.08 m
Max outboard off-tracking in 90 deg R12.5m turn: 0.34 m
Lane change yaw-rate amplification of last unit: 1.64
Lane change acceleration amplification of last unit: 2.17
Lane change off-tracking: 0.62 m
Lane change load transfer ratio: 52%
Yaw damping of last unit: 0.34
Max straight line off-tracking at 0.19 m
Load on assumed drive axles: 20.7%
Pressed load on assumed drive axles: 22.8%

Appendix 2, Vectorized presentation of semitrailer -combination's single track model

Number of units: 2
Number of axles per unit: 3

Slip angle:

$$\alpha = \frac{v_y + L_{cog} \cdot \omega_z}{v_x} - \delta$$

Modelica code

```
alpha + delta = (matrix(vyu) * ones(1,na) + Lcog .* (matrix(wz) *
ones(1,na))) ./ (matrix(vx) * ones(1,na));
```

$$\begin{bmatrix} \alpha_{11} & \alpha_{12} & \alpha_{13} \\ \alpha_{21} & \alpha_{22} & \alpha_{23} \end{bmatrix} = \frac{\begin{bmatrix} v_{y1} \\ v_{y2} \end{bmatrix} \cdot \begin{bmatrix} 1 & 1 & 1 \end{bmatrix} + \begin{bmatrix} L_{cog11} & L_{cog12} & L_{cog13} \\ L_{cog21} & L_{cog22} & L_{cog23} \end{bmatrix} .\cdot \begin{bmatrix} \omega_{z1} & \omega_{z1} & \omega_{z1} \\ \omega_{z2} & \omega_{z2} & \omega_{z2} \end{bmatrix}}{\begin{bmatrix} v_{x1} \\ v_{x2} \end{bmatrix} \cdot \begin{bmatrix} 1 & 1 & 1 \end{bmatrix}} - \begin{bmatrix} \delta_{11} & \delta_{12} & \delta_{13} \\ \delta_{21} & \delta_{22} & \delta_{23} \end{bmatrix}$$

$$\rightarrow \alpha_{11} = \frac{v_{y1} + L_{cog11} \cdot \omega_{z1}}{v_{x1}} - \delta_{11}$$

Lateral tire force:

$$F_{yw} = -C \cdot \alpha$$

Modelica code

```
Fyw = -C .* alpha;
```

$$\begin{bmatrix} F_{yw11} & F_{yw12} & F_{yw13} \\ F_{yw21} & F_{yw22} & F_{yw23} \end{bmatrix} = -\begin{bmatrix} C_{11} & C_{12} & C_{13} \\ C_{21} & C_{22} & C_{23} \end{bmatrix} .\cdot \begin{bmatrix} \alpha_{11} & \alpha_{12} & \alpha_{13} \\ \alpha_{21} & \alpha_{22} & \alpha_{23} \end{bmatrix}$$

$$\rightarrow F_{yw11} = -C_{11} \cdot \alpha_{11}$$

Conversion to unit's coordinate system

$$F_x = F_{xw} \cdot \cos(\delta) - F_{yw} \cdot \sin(\delta)$$

$$F_y = F_{xw} \cdot \sin(\delta) + F_{yw} \cdot \cos(\delta)$$

Modelica code

```
Fx = Fxw .* cos(delta) - Fyw .* sin(delta);
Fy = Fxw .* sin(delta) + Fyw .* cos(delta);
```

$$\begin{bmatrix} F_{x11} & F_{x12} & F_{x13} \\ F_{x21} & F_{x22} & F_{x23} \end{bmatrix} = \begin{bmatrix} F_{xw11} & F_{xw12} & F_{xw13} \\ F_{xw21} & F_{xw22} & F_{xw23} \end{bmatrix} .\cdot cos\left(\begin{bmatrix} \delta_{11} & \delta_{12} & \delta_{13} \\ \delta_{21} & \delta_{22} & \delta_{23} \end{bmatrix}\right) - \begin{bmatrix} F_{yw11} & F_{yw12} & F_{yw13} \\ F_{yw21} & F_{yw22} & F_{yw23} \end{bmatrix} .\cdot sin\left(\begin{bmatrix} \delta_{11} & \delta_{12} & \delta_{13} \\ \delta_{21} & \delta_{22} & \delta_{23} \end{bmatrix}\right)$$

$$\begin{bmatrix} F_{y11} & F_{y12} & F_{y13} \\ F_{y21} & F_{y22} & F_{y23} \end{bmatrix} = \begin{bmatrix} F_{xw11} & F_{xw12} & F_{xw13} \\ F_{xw21} & F_{xw22} & F_{xw23} \end{bmatrix} .\cdot sin\left(\begin{bmatrix} \delta_{11} & \delta_{12} & \delta_{13} \\ \delta_{21} & \delta_{22} & \delta_{23} \end{bmatrix}\right) + \begin{bmatrix} F_{yw11} & F_{yw12} & F_{yw13} \\ F_{yw21} & F_{yw22} & F_{yw23} \end{bmatrix} .\cdot cos\left(\begin{bmatrix} \delta_{11} & \delta_{12} & \delta_{13} \\ \delta_{21} & \delta_{22} & \delta_{23} \end{bmatrix}\right)$$

$\rightarrow F_{x11} = F_{xw11} \cdot \cos(\delta_{11}) - F_{yw11} \cdot \sin(\delta_{11})$

$\rightarrow F_{y11} = F_{xw11} \cdot \sin(\delta_{11}) + F_{yw11} \cdot \cos(\delta_{11})$

Coupling

$$v_x[i+1] = v_x[i] \cdot \cos(\theta[i]) - \left(v_y[i] + B_{cog}[i] \cdot \omega_z[i]\right) \cdot \sin(\theta[i])$$

$$v_y[i+1] + A_{cog}[i+1] \cdot \omega_z[i+1] = \left(v_y[i] + B_{cog}[i] \cdot \omega_z[i]\right) \cdot \cos(\theta[i]) + v_x[i] \cdot \sin(\theta[i])$$

Modelica code

```
vx[i+1] = vx[i] * cos(theta[i]) - (vys[i] + Bcog[i] * wz[i] + (hC[1] -
hRC[i]) * wx[i]) * sin(theta[i]);
vys[i+1] + Acog[i+1] * wz[i+1] + (hC[1] - hRC[i+1]) * wx[i+1] = (vys[i] +
Bcog[i] * wz[i] + (hC[1] - hRC[i]) * wx[i]) * cos(theta[i]) + vx[i] *
sin(theta[i]);
```

$$v_x[2] = v_x[1] \cdot \cos(\theta[1]) - \left(v_y[1] + B_{cog}[1] \cdot \omega_z[1]\right) \cdot \sin(\theta[1])$$

$$v_y[2] + A_{cog}[2] \cdot \omega_z[2] = \left(v_y[1] + B_{cog}[1] \cdot \omega_z[1]\right) \cdot \cos(\theta[1]) + v_x[1] \cdot \sin(\theta[1])$$

Equilibrium of forces

$$m \cdot \left(\dot{v_x} - v_y * \omega_z\right) = F_x + F_{cx}$$

Modelica code

```
m .* (d_vx - vys .* wz) = vector(Fx * ones(na,1) - [matrix(Fcx);0] +
[0;matrix(Fcx)] .* cos([0;matrix(theta)]) - [0;matrix(Fcy)] .*
sin([0;matrix(theta)]));
```

$$\begin{bmatrix} m_1 \\ m_2 \end{bmatrix} .\cdot \left(\begin{bmatrix} \dot{v_{x1}} \\ \dot{v_{x2}} \end{bmatrix} - \begin{bmatrix} v_{y1} \\ v_{y2} \end{bmatrix} \cdot \begin{bmatrix} \omega_{z1} \\ \omega_{z2} \end{bmatrix} \right) = \begin{bmatrix} F_{x11} & F_{x12} & F_{x13} \\ F_{x21} & F_{x22} & F_{x23} \end{bmatrix} \cdot \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} - \begin{bmatrix} F_{cx} \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ F_{cx} \end{bmatrix} .\cdot \cos\left(\begin{bmatrix} 0 \\ \theta \end{bmatrix} \right) - \begin{bmatrix} 0 \\ F_{cy} \end{bmatrix} .\cdot \sin\left(\begin{bmatrix} 0 \\ \theta \end{bmatrix} \right)$$

$\rightarrow m_1 \cdot \left(\dot{v_{x1}} - v_{y1} \cdot \omega_{z1}\right) = F_{x11} + F_{x12} + F_{x13} - F_{cx}$

$\rightarrow m_2 \cdot \left(\dot{v_{x2}} - v_{y2} \cdot \omega_{z2}\right) = F_{x21} + F_{x22} + F_{x23} + F_{cx} \cdot \cos(\theta) - F_{cy} \cdot \sin(\theta)$

$$m \cdot \left(\dot{v_y} + v_x \cdot \omega_z\right) = F_y + F_{cy}$$

Modelica code

```
m .* (d_vys + vx .* wz) = vector(Fy * ones(na,1) - [matrix(Fcy);0] +
[0;matrix(Fcx)] .* sin([0;matrix(theta)]) + [0;matrix(Fcy)] .*
cos([0;matrix(theta)]));
```

$$\begin{bmatrix} m_1 \\ m_2 \end{bmatrix} .\cdot \left(\begin{bmatrix} \dot{v_{y1}} \\ \dot{v_{y2}} \end{bmatrix} + \begin{bmatrix} v_{x1} \\ v_{x2} \end{bmatrix} .\cdot \begin{bmatrix} \omega_{z1} \\ \omega_{z2} \end{bmatrix} \right) = \begin{bmatrix} F_{y11} & F_{y12} & F_{y13} \\ F_{y21} & F_{y22} & F_{y23} \end{bmatrix} \cdot \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} - \begin{bmatrix} F_{cy} \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ F_{cx} \end{bmatrix} .\cdot \sin\left(\begin{bmatrix} 0 \\ \theta \end{bmatrix} \right) - \begin{bmatrix} 0 \\ F_{cy} \end{bmatrix} .\cdot \cos\left(\begin{bmatrix} 0 \\ \theta \end{bmatrix} \right)$$

$\rightarrow m_1 \cdot \left(\dot{v_{y1}} + v_{x1} \cdot \omega_{z1}\right) = F_{y11} + F_{y12} + F_{y13} - F_{cy}$

$\rightarrow m_2 \cdot \left(\dot{v_{y2}} + v_{x2} \cdot \omega_{z2}\right) = F_{y21} + F_{y22} + F_{y23} + F_{cx} \cdot \sin(\theta) - F_{cy} \cdot \cos(\theta)$

$$I_z \cdot \dot{\omega}_z = L_{cog} \cdot F_y \pm L_{C-cog} \cdot F_{cy}$$

Modelica code

```
Iz .* d_wz = vector(Lcog .* Fy * ones(na,1) - matrix(Bcog) .* [matrix(Fcy);0]
+ matrix(Acog) .* ([0;matrix(Fcx)] .* sin([0;matrix(theta)]) +
[0;matrix(Fcy)] .* cos([0;matrix(theta)])));
```

$$\begin{bmatrix} I_{z1} \\ I_{z2} \end{bmatrix} .\cdot \begin{bmatrix} \dot{\omega_{z1}} \\ \dot{\omega_{z2}} \end{bmatrix} = \begin{bmatrix} L_{cog11} & L_{cog12} & L_{cog13} \\ L_{cog21} & L_{cog22} & L_{cog23} \end{bmatrix} .\cdot \begin{bmatrix} F_{y11} & F_{y12} & F_{y13} \\ F_{y21} & F_{y22} & F_{y23} \end{bmatrix} \cdot \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} - \begin{bmatrix} B_{cog1} \\ B_{cog2} \end{bmatrix} .\cdot \begin{bmatrix} F_{cy} \\ 0 \end{bmatrix} + \begin{bmatrix} A_{cog1} \\ A_{cog2} \end{bmatrix} .\cdot$$
$$\left(\begin{bmatrix} 0 \\ F_{cx} \end{bmatrix} .\cdot \sin\left(\begin{bmatrix} 0 \\ \theta \end{bmatrix} \right) + \begin{bmatrix} 0 \\ F_{cy} \end{bmatrix} .\cdot \cos\left(\begin{bmatrix} 0 \\ \theta \end{bmatrix} \right) \right)$$

$$\rightarrow I_{z1} \cdot \dot{\omega_{z1}} = L_{cog11} \cdot F_{y11} + L_{cog12} \cdot F_{y12} + L_{cog13} \cdot F_{y13} - B_{cog1} \cdot F_{cy}$$

$$\rightarrow I_{z2} \cdot \dot{\omega_{z2}} = L_{cog21} \cdot F_{y21} + L_{cog22} \cdot F_{y22} + L_{cog23} \cdot F_{y23} + A_{cog2} \cdot \left(F_{cx} \cdot \sin(\theta) + F_{cy} \cdot \cos(\theta)\right)$$

Appendix 3, Vectorized presentation of semitrailer -combination's Static axle loads

Number of units: 2

Number of axles per unit: 3

$$m \cdot g = F_z - F_{cz}$$

Modelica code

```
m * g = vector(Fz * ones(na, 1) - [matrix(Fcz); 0] + [0; matrix(Fcz)]);
```

$$\begin{bmatrix} m_1 \\ m_2 \end{bmatrix} .\cdot \begin{bmatrix} g \\ g \end{bmatrix} = \begin{bmatrix} F_{z11} & F_{z12} & F_{z13} \\ F_{z21} & F_{z22} & F_{z23} \end{bmatrix} \cdot \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} - \begin{bmatrix} F_{cz} \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ F_{cz} \end{bmatrix}$$

$$\rightarrow m_1 \cdot g = F_{z11} + F_{z12} + F_{z13} - F_{cz}$$

$$\rightarrow m_2 \cdot g = F_{z21} + F_{z22} + F_{z23} + F_{cz}$$

$$0 = L_{cog} \cdot F_z - B_{cog} \cdot F_{cz}$$

Modelica code

```
zeros(nu) = vector(Lcog .* Fz * ones(na, 1) - matrix(Bcog) .* [matrix(Fcz);
0] + matrix(Acog) .* [0; matrix(Fcz)]);
```

$$\begin{bmatrix} 0 \\ 0 \end{bmatrix} = \begin{bmatrix} L_{cog11} & L_{cog12} & L_{cog13} \\ L_{cog21} & L_{cog22} & L_{cog23} \end{bmatrix} .\cdot \begin{bmatrix} F_{z11} & F_{z12} & F_{z13} \\ F_{z21} & F_{z22} & F_{z23} \end{bmatrix} \cdot \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} - \begin{bmatrix} B_{cog1} \\ B_{cog2} \end{bmatrix} .\cdot \begin{bmatrix} F_{cz} \\ 0 \end{bmatrix} + \begin{bmatrix} A_{cog1} \\ A_{cog2} \end{bmatrix} .\cdot \begin{bmatrix} 0 \\ F_{cz} \end{bmatrix}$$

$$\rightarrow \ 0 = L_{cog11} F_{z11} + L_{cog12} F_{z12} + L_{cog13} F_{z13} - B_{cog1} F_{cz}$$

$$\rightarrow \ 0 = L_{cog21} F_{z21} + L_{cog22} F_{z22} + L_{cog23} F_{z23} + A_{cog2} F_{cz}$$

$$C = Cc \cdot F_z$$

Modelica code

```
C = Cc .* Fz
```

$$\begin{bmatrix} C_{11} & C_{12} & C_{13} \\ C_{21} & C_{22} & C_{23} \end{bmatrix} = \begin{bmatrix} Cc_{11} & Cc_{12} & Cc_{13} \\ Cc_{21} & Cc_{22} & Cc_{23} \end{bmatrix} .\cdot \begin{bmatrix} F_{z11} & F_{z12} & F_{z13} \\ F_{z21} & F_{z22} & F_{z23} \end{bmatrix}$$

$$\rightarrow C_{11} = Cc_{11} \cdot F_{z11}$$

Appendix 4, Vectorized presentation of semitrailer -combination's Roll dynamics

Number of units: 2
Number of axles per unit: 3

Equilibrium of unsprung mass:

$$M_x + F_y \cdot h_{RC} = M_{xs} \cdot M_{xd}$$

Modelica code

```
Mxs + Mxd = vector(matrix(Mx) + Fy * ones(na,1) .* matrix(hRC));
```

$$\begin{bmatrix} M_{x1} \\ M_{x2} \end{bmatrix} + \begin{bmatrix} F_{y11} & F_{y12} & F_{y13} \\ F_{y21} & F_{y22} & F_{y23} \end{bmatrix} \cdot \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} .\cdot \begin{bmatrix} h_{RC1} \\ h_{RC2} \end{bmatrix} = \begin{bmatrix} M_{xs1} \\ M_{xs2} \end{bmatrix} + \begin{bmatrix} M_{xd1} \\ M_{xd2} \end{bmatrix}$$

$$\rightarrow M_{x1} + (F_{y11} + F_{y12} + F_{y13}) \cdot h_{RC1} = M_{xs1} + M_{xd1}$$

$$\rightarrow M_{x2} + (F_{y21} + F_{y22} + F_{y23}) \cdot h_{RC2} = M_{xs2} + M_{xd2}$$

Compatibility of Lateral velocities:

$$v_{yu} = v_{ys} + (h - h_{RC}) \cdot \omega_x$$

Modelica code

```
vyu = vector(matrix(vys) + matrix(wx) .* (matrix(h) - matrix(hRC)));
```

$$\begin{bmatrix} v_{yu1} \\ v_{yu2} \end{bmatrix} = \begin{bmatrix} v_{ys1} \\ v_{ys2} \end{bmatrix} + \left(\begin{bmatrix} h_1 \\ h_2 \end{bmatrix} - \begin{bmatrix} h_{RC1} \\ h_{RC2} \end{bmatrix} \right) .\cdot \begin{bmatrix} \omega_{x1} \\ \omega_{x2} \end{bmatrix}$$

$$\rightarrow v_{yu1} = v_{ys1} + (h_1 - h_{RC1}) \cdot \omega_{x1}$$

$$\rightarrow v_{yu2} = v_{ys2} + (h_2 - h_{RC2}) \cdot \omega_{x2}$$

Equilibrium for the whole vehicle in roll plane:

$$I_{xs} \cdot \dot{\omega}_x = M_x + F_y \cdot h + m \cdot g \cdot (h - h_{RC}) \cdot p_x$$

Modelica code

```
Ix .* d_wx = vector(matrix(Mx) + Fy * ones(na,1) .* matrix(h) + matrix(m) .*
matrix(fill(g, nu, 1)) .* matrix(h - hRC) .* matrix(px) + matrix([Fcy;0]) .*
(matrix(h) - matrix(fill(hC, nu, 1))) + matrix([0 ; sin(theta) .* Fcy +
cos(theta) .* Fcx]) .* (matrix(h) - matrix(fill(hC, nu, 1))));
```

$$\begin{bmatrix} I_{xs1} \\ I_{xs2} \end{bmatrix} .\cdot \begin{bmatrix} \dot{\omega}_{x1} \\ \dot{\omega}_{x2} \end{bmatrix} = \begin{bmatrix} M_{x1} \\ M_{x2} \end{bmatrix} + \begin{bmatrix} F_{y11} & F_{y12} & F_{y13} \\ F_{y21} & F_{y22} & F_{y23} \end{bmatrix} \cdot \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} .\cdot \begin{bmatrix} h_1 \\ h_2 \end{bmatrix} + \begin{bmatrix} m_1 \\ m_2 \end{bmatrix} .\cdot \begin{bmatrix} g \\ g \end{bmatrix} .\cdot \left(\begin{bmatrix} h_1 \\ h_2 \end{bmatrix} - \begin{bmatrix} h_{RC1} \\ h_{RC2} \end{bmatrix} \right) .\cdot \begin{bmatrix} p_{x1} \\ p_{x2} \end{bmatrix} + \begin{bmatrix} F_{cy} \\ 0 \end{bmatrix} .\cdot \left(\begin{bmatrix} h_1 \\ h_2 \end{bmatrix} - \begin{bmatrix} h_{C1} \\ h_{C2} \end{bmatrix} \right) + \begin{bmatrix} 0 \\ \sin(\theta) \cdot F_{cy} + \cos(\theta) \cdot F_{cx} \end{bmatrix} .\cdot \left(\begin{bmatrix} h_1 \\ h_2 \end{bmatrix} - \begin{bmatrix} h_{C1} \\ h_{C2} \end{bmatrix} \right)$$

$$\rightarrow I_{xs1} \cdot \dot{\omega}_{x1} = M_{x1} + \left(F_{y11} + F_{y12} + F_{y13}\right) \cdot h_1 + m_1 \cdot g \cdot (h_1 - h_{RC1}) \cdot p_{x1} + F_{cy} \cdot (h_1 - h_{C1})$$

$\rightarrow\ I_{xs2} \cdot \dot{\omega_{x2}} = M_{x2} + \left(F_{y21} + F_{y22} + F_{y23}\right) \cdot h_2 + m_2 \cdot g \cdot (h_2 - h_{RC2}) \cdot p_{x2} + (\sin(\theta) \cdot F_{cy} + \cos(\theta) \cdot F_{cx}) \cdot (h_2 - h_{C2})$

Coupling

$\dot{M_{xs}} = -c_{roll} \cdot \omega_x$
$M_{xd} = -d_{roll} \cdot \omega_x$

Modelica code

```
d_Mxs = -croll .* wx;
Mxd = -droll .* wx;
```

$$\begin{bmatrix} \dot{M_{xs1}} \\ \dot{M_{xs2}} \end{bmatrix} = -\begin{bmatrix} c_{roll1} \\ c_{roll2} \end{bmatrix} .\cdot \begin{bmatrix} \omega_{x1} \\ \omega_{x2} \end{bmatrix}$$

$$\begin{bmatrix} M_{xd1} \\ M_{xd2} \end{bmatrix} = -\begin{bmatrix} d_{roll1} \\ d_{roll2} \end{bmatrix} .\cdot \begin{bmatrix} \omega_{x1} \\ \omega_{x2} \end{bmatrix}$$

Appendix 5, Vehicle parameters for simulations

Double CAT VTM

nu = 3
na = 4
axlegroups = [1,2,2,2; 1,1,1,0; 1,1,1,0]
driven=[false,true,true,false; false,false,false,false; false,false,false,false]
m = {31726,23300,23300}
Iz = {1.8694e5,1.2283e5,1.2283e5}
L = [0,-3.7,-5.07, -6.45; 0, -1.30, -2.6, 0; 0, -1.3, -2.6, 0]
A = {0,5.4,5.4}
B = {-7.5150,-3.1,-3.1}
X = {-3.7922, -1.1249, -1.1249}
h = {1.9550,1.9159,1.9159}
hRC = {0.82, 0.52, 0.545}
hC = {0.71,0.5}
Cc = 7.5
Lr = 0.4
ks = [0.3414e6, 0.8001e6, 0.8001e6, 0.7942e6; 1.5e6, 1.5e6, 1.5e6, 0; 1.5e6, 1.5e6, 1.5e6, 0]
cs = [16902, 32865, 32865, 32865; 30000, 30000, 30000, 0; 30000, 30000, 30000, 0]
Ix = {2.7073e4,2.0847e4, 2.0847e4}
W=[2.09, 1.85, 1.85, 2.09; 2.05, 2.05, 2.05, 0; 2.05, 2.05, 2.05, 0]

Nordic Combination VTM

nu = 3
na = 4
axlegroups = [1,2,2,2; 1,1,0,0; 1,1,1,0]
driven=[false,true,true,false; false,false,false,false; false,false,false,false]
m = {31726,2800,31000}
Iz = {1.8694e5,6.6675e3,4.6524e5}
L = [0,-3.7,-5.07, -6.45; 0, -1.31, 0, 0; 0, -1.3, -2.6, 0]
A = {0,3.9,6.8}
B = {-7.5150,-0.65,-3.7}
X = {-3.7922, -0.1760, 1.5461}
h = {1.9550,0.74,1.8987}
h = {1,0.74,1}
h = {2.5,0.74,2.5}
hRC = {0.82, 0.52, 0.545}
hC = {0.71,1.1}
Cc = 7.5
Lr = 0.4
ks = [0.3414e6, 0.8001e6, 0.8001e6, 0.7942e6; 1.5e6, 1.5e6, 0, 0; 1.5e6, 1.5e6, 1.5e6, 0]
cs = [16902, 32865, 32865, 32865; 30000, 30000, 0, 0; 30000, 30000, 30000, 0]
Ix = {2.7073e4,1000, 2.8429e4}
W=[2.09, 1.85, 1.85, 1.85; 2.05, 2.05, 0, 0; 2.05, 2.05, 2.05, 0]

A-double VTM

nu = 4
na = 3
axlegroups = [1, 2, 2 ; 1, 1, 1 ; 1, 1, 0; 1, 1, 1]
driven=[false, true, true ; false, false, false ; false, false, false ; false, false, false]
m = {9231.0, 31000.0, 2800.0, 31000.0}
Iz = {4.4483e4,4.6524e5,6.6674e3,4.6524e5}
L = [0.0, -3.4, -4.77 ; 0.0, -1.3, -2.6 ; 0.0, -1.31, 0.0 ; 0.0, -1.3, -2.6]
A = {0.0, 6.8, 3.9, 6.8}
B = {-3.775, -3.7, -0.65, -3.7}
X = {-1.8641, 1.5461, -0.1670, 1.5461}
h = {0.9704, 1.8987, 0.74, 1.8987}
h = {0.9704, 1.0, 0.74, 1.0}
h = {0.9704, 2.5, 0.74, 2.5}
hRC = {0.681, 0.5450, 0.52, 0.5450}
hC = {1.0, 0.5, 1.0}
Cc = 7.5
Lr = 0.4
ks = [4.6388e5, 4.8284e5, 4.8284e5; 1.5e6, 1.5e6, 1.5e6; 1.5e6, 1.5e6, 0; 1.5e6, 1.5e6, 1.5e6]
cs = [14119, 16981, 16891; 30000, 30000, 30000; 30000, 30000, 0; 30000, 30000, 30000]
Ix = {4700.2, 2.8429e4, 1000, 2.8429e4}
W=[2.09, 1.85, 1.85; 2.05, 2.05, 2.05; 2.05, 2.05, 0; 2.05, 2.05, 2.05]

Appendix 6, Vehicle parametrization for simulations