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Modelling and assessment of performance based standards for high capacity vehicles

Master's thesis in Vehicle Dynamics

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Cover:

Autofreight research project, A-double combination vehicle in real transport operation testing (photo: Borås Stad)

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Abstract

Interest in High capacity transports (HCT) has increased due to better transport efficiency and saving potential in CO₂ emissions. HCTs are already allowed on the specified parts of the roads in Sweden (74 tons, 25.25 metres) and longer combination vehicles (30-35 metres) are being considered. The assessment of HCTs could be based on performance based standards (PBS). PBSs are performance requirements on the vehicle to ensure their safe operation on the roads.

In Sweden, there are two existing tools for evaluating performance of HCT vehicles with respect to PBS. The goal of this thesis is to help the research project "Performance Based Standards II" to further develop the assessment tool OpenPBS and compare it with the other existing PBS tool.

For the comparison, OpenPBS was modified to use the same input parameters as the other tool. Also, the manoeuvres and fixed simulation parameters were aligned and the same tyre model was used. According to this thesis, OpenPBS would be the more suitable tool for evaluating HCT vehicles than the other tool. Motivation for OpenPBS is the stronger physical base for the modelling, thanks to Modelica format. For instance, future vehicle combinations should be easier to support. However, it should be mentioned that OpenPBS is a more advanced tool due to object orientation and therefore requires more competence in physical modelling.

Key words: Vehicle Dynamics, Modelica, Performance Based Standards, High Capacity Transports, Long Heavy Combination Vehicles

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Preface

This thesis is part of master's degree in mechanical engineering at University of Oulu. The thesis work was done in collaboration with Chalmers University of Technology and Volvo GTT in Göteborg during the time period from February 2020 to June 2020. The final version of the report was written in Ylivieska, Finland during June and July 2020. This thesis is also published and approved as a master's thesis at University of Oulu corresponding to the master's degree requirements.

I would like to thank my examiners and supervisors for their time and support during the thesis work; Bengt Jacobson, Sogol Kharrazi, Niklas Fröjd, Ville Santahuhta, Miro-Tommi Tuutijärvi and special thanks for professor emeritus Mauri Haataja, who encouraged me to seize this opportunity.

Thanks Tuuli & Sisu for letting me go and even bigger thanks for letting me come back.

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Mikko Karisaari

Notations

Abbreviations

<i>CAT</i>	Centre Axle Trailer
<i>CoG</i>	Centre of Gravity
<i>GA</i>	Gradeability
<i>HCT</i>	High Capacity Transport
<i>LCV</i>	Long Combination Vehicle
<i>PBS</i>	Performance Based Standards
<i>RWA</i>	Rearward Amplification
<i>SRT</i>	Steady state Rollover Threshold
<i>TASP</i>	Tracking Ability on Straight Path
<i>VTR</i>	Vägtrafikregistret
<i>YD</i>	Yaw Damping

Roman upper-case letters

<i>A</i>	Distance to unit's front coupling from first axle of unit
<i>A_w</i>	Axle width
<i>B</i>	Distance to unit's rear coupling from first axle of unit
<i>C</i>	Shape factor of non-linear tyre model
<i>CC_y</i>	Cornering coefficient at actual load
<i>CC_{y0}</i>	Cornering coefficient at nominal tyre load
<i>C_d</i>	Air resistance coefficient
<i>C_{st}</i>	Cornering stiffness (slip)
<i>CYT</i>	Tyre lateral (deformation) stiffness
<i>CZT</i>	Tyre vertical (deformation) stiffness
<i>D_K</i>	Distribution of the vehicle roll stiffness
<i>E</i>	Gap between the double tyres
<i>F_{cz}</i>	Vertical coupling force
<i>F_d</i>	Aerodynamic drag
<i>F_m</i>	Force on driven wheels from powertain
<i>FOH</i>	Front overhang
<i>F_r</i>	Rolling resistance
<i>VI</i>	

F_u	Vehicle unsprung load
F_{YT}	Tyre lateral force
F_z	Vertical axle force
F_{ZT}	Vertical tyre force
F_{ZT0}	Tyre nominal load value
I_x	Inertia on roll plane
I_z	Inertia on yaw plane
KC	Cornering coefficient (Reference tool)
L	Axle positions relative to first axle of unit
L_{CoG}	Axle distance to CoG
L_r	Relaxation length
ROH	Rear overhang
S	Tyre width
W	Track width
W_e	Effective track width
W_{ev}	Vehicle effective track width
W_{fw}	Fifth wheel representative track width
X	CoG longitudinal position from the first axle of unit

Roman lower-case letters

$axfs$	Distance between the first and last axle of combination
ayl	Lateral acceleration when wheel lift-off happens
aym	Theoretical maximum lateral acceleration
ccg_y	Maximum cornering coefficient gradient
ccr	Roll damping of coupling
cs	Suspension roll damping
fr	Rolling resistance coefficient
g	Fall acceleration
h	Centre of gravity height
hs	Sprung mass centre of gravity height
hC	Height of coupling
hRC	Height of roll centre of unit
i	Unit
j	Axle

k_{cr}	Roll stiffness of coupling
k_{es}	Equivalent suspension roll stiffness
k_s	Suspension roll stiffness
k_t	Roll stiffness from tyre vertical stiffness
k	Roll stiffness of an axle
k_v	Roll stiffness of the vehicle
m	Mass of unit
nu	Number of units
na	Maximum number of axles
pz	Yaw angle
r_{fc}	Defines roll-free couplings
r_y	Axle lateral position
t	Time
u_{g_y}	Maximum lateral force gradient
u_y	Maximum lateral force coefficient at actual load
u_{y0}	Maximum lateral force coefficient at tyre nominal load
u_2	Peak friction
v_x	Longitudinal velocity
v_y	Lateral velocity
v_{ys}	Lateral velocity at vehicle CoG
w_z	Yaw rate

Greek lower-case letters

α	Slip angle
φ	Roll-angle
φ_g	Roll gradient
φ_{gH}	Normalized rollgradient
ρ	Air density
η	Total efficiency of powertrain

SI units and radians used where not else stated

1 Introduction

This thesis is a part of a research project called “Performance Based Standards II” coordinated by the Swedish National Road and Transport Research Institute, VTI. Thesis is done in collaboration with Chalmers University of Technology and Volvo GTT. The thesis will help the project to develop the assessment tool OpenPBS.

1.1 Background

The growing freight volumes, congestion problems and emission reduction targets have increased the attractiveness of High Capacity Transport (HCT) in road transports. HCTs have a better transport efficiency and great saving potential in CO₂ emissions. In Sweden, the HCT vehicles were approved for the roads in April 2018 with the maximum gross weight of 74 tons. The maximum length of the combination is expected to rise above the current 25.25 metres as it rose to 34.5 metres in Finland at the beginning of 2019 (Finlex, 2020). Long Combination Vehicles (LCV) about 30-35 metres have even more savings potential. The existing legal requirements are mostly regulations for the single vehicle units. There can be many variations between LCVs and allowing the suitable ones could be based on Performance Based Standards (PBS). (Kharrazi, S. et al., 2017, p. 13) PBS are requirements on vehicle’s performance to ensure that they can operate safely on the roads, instead of strict dimensions and weight regulations.

The project “Performance Based Standards for High Capacity Transport in Sweden” led to a proposal for using PBS scheme in Sweden, considering the winter conditions. The project “Performance Based Standards II” aims to further develop the proposed PBS scheme and support its implementation. One of the main goals of the project is to support development of “OpenPBS tool”, which should enable to simulate and assess HCT vehicles with respect to PBS in terms of legislation. Specifying the HCT’s required safe performance with the PBS could be therefore used in legislation instead of limiting the vehicle’s weight and length. Swedish national road and transport research institute VTI is the coordinator of the project,

involving Volvo Group Trucks Technology, Scania, Nokian Heavy Tyres Ltd, Transportstyrelsen, Trafikverket, Chalmers University of Technology, University of Oulu and Parator Industri AB. (Chalmers University of Technology, 2020)

1.2 Problem motivating the project

To evaluate the HCT vehicles accurately, the OpenPBS tool should have models with adequate complexity regarding the respective performance measures.

Adequate complexity is mainly a balance between computing accurate performance measures using the limited parameter set for each individual vehicle unit available from register data.

In this scope, some models have been developed in the first PBS project, but there is a need for further development of models and implementation of them in the OpenPBS tool in Modelica. Further the OpenPBS tool should be compared with the other existing PBS tool in Sweden and the differences should be analysed. The other existing PBS tool is called “the Reference tool” in this thesis.

1.3 Envisioned solution

Some modifications to the OpenPBS tool need to be made, so that the comparison between OpenPBS and the Reference tool can be made. The idea is to use the same parameter input in OpenPBS as in the Reference tool. Then vehicle unit parameters from the vehicle register should be converted to OpenPBS vehicle parameters. Conversion of the vehicle parameters can be made in Python script, which will be used for running the comparison. The fixed simulation parameters should be aligned inside the models. Also, some improvements and fixes will be made in the Modelica models in OpenPBS.

The vision is that the driver should be able to assess the loaded vehicle via an automatic assessment tool by inserting registration numbers of combination. Also, some additional values should be defined by user, such as vertical forces on each axle and height of the load. All other vehicle parameters must be standardized or calculated or easily available. Therefore, the models has to be as simple as possible, minimizing the number of needed vehicle parameters. Thus, OpenPBS does not try to compete with high-fidelity modelling tools such as Adams or TruckMaker.

1.4 Objective

The thesis will help the “PBS II” project to develop the assessment tool OpenPBS. Also, OpenPBS will be compared with the other existing PBS tool in Sweden.

1.5 Deliverables

The deliverables are listed below:

- Improvements in OpenPBS to cover more PBS measures and handle relevant combination vehicles. Relevance to be determined by project.
- A method/tool that compares OpenPBS and the Reference tool
- Consecutive runs of the comparison, with intermediate improvements
- Analysis of differences in terms of:
 - parameter conversion from vehicle register’s vehicle data and user-given parameters to calculated model parameters
 - the definitions of manoeuvres, measures, and models

1.6 Limitations

Comparison between OpenPBS and the Reference tool can be done only for Nordic combinations and B-double combinations because the Reference tool includes only those two combinations. The Reference tool also does not compute all PBSs. Hence, only PBSs considered in the Reference tool can be compared.

1.7 Method

The thesis consists of the following steps:

- Information study / literature search. Including learning Modelica and getting familiar with OpenPBS.
- Compare OpenPBS to the Reference tool
 - Complete the parameter input to OpenPBS to match the Reference tool (e.g. vertical loads, the number of tyres on each axle, ...)
 - Align manoeuvres between OpenPBS and the Reference tool
 - Select 100 combination vehicles, defined as combinations of 10-100 real vehicle units. The units should come from Swedish

Vehicle Registry, received from Transportstyrelsen (Swedish Transport Agency).

- Propose and develop an automated process for comparison of OpenPBS and the Reference tool. Run comparison of around 100 combinations and analyze differences.
- Implement new models in OpenPBS and further develop the vehicle input process
 - Implement the SRT estimation method
 - Implement the traction (longitudinal) measures

2 Comparison method

Python script was created to compare OpenPBS with the Reference tool, as illustrated in Figure 1. The comparison script retrieves parameters for each unit from an Excel table, using the registration numbers of the units in the vehicle combination. The parameters in the Excel file are taken from the Swedish vehicle registry (Vägtrafikregistret=VTR). VTR parameters are used to define the vehicle parameters of OpenPBS. Conversion of the parameters is presented in Chapter 4. After the parameter conversion, vehicle parameters are transferred to OpenPBS via OMPython. Then the comparison code runs the selected OpenPBS manoeuvres and gives the results of simulation. Compared PBS measures and manoeuvres are presented in Chapter 5. The Python script also runs the Reference tool, where the same VTR unit parameters are used. The code is automated, so it can run both tools for several vehicles at once.

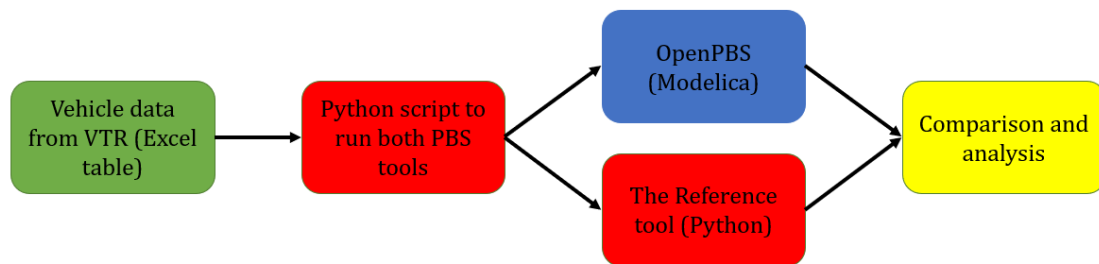


Figure 1. Flow chart of the comparison code structure.

2.1 Compared vehicles

The Reference tool can calculate PBS measures only for two types of vehicle combinations, Nordic combination and B-double. These combinations are presented in Figure 2. It should be noted that there are two types of Nordic combinations, rigid truck with a fulltrailer or dolly and semitrailer. The main difference between these two types of Nordic combination is in the couplings. Drawbar of a fulltrailer has a pitch-moment-free rear end while drawbar of a dolly has a pitch-moment-rigid rear end. Respectively, fulltrailer's turntable is pitch-rigid while dolly's fifth wheel is pitch-free. B-double consists of tractor, link-trailer, and semitrailer, so there are two fifth wheel couplings. (Jacobson 2019, p. 67)

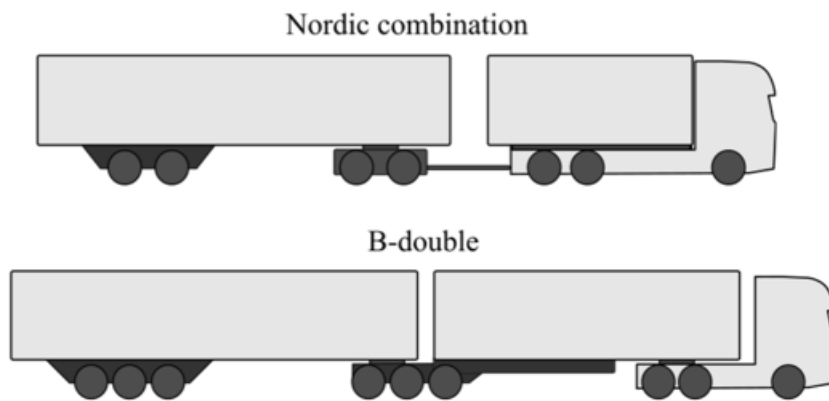


Figure 2. Compared HCT combination vehicles, based on (ISO 18868:2013).

A list of one hundred combination vehicles using 75 real units from VTR was prepared for the comparison as following:

- 35 Nordic combinations with a converted dolly
- 35 Nordic combinations with a fulltrailer
- 30 B-doubles.

Number of B-double combinations is smaller, because those combinations are quite similar to each other, and variations of vehicle dimensions are smaller than in Nordic combinations.

A vehicle is defined in the Python script as a vector of unit registration numbers in coupling order starting with the first unit. In Python format, a vehicle combination could be:

```
Openpbs(["XXX123", "YYY456", "ZZZ789"])
```

A vehicle type is assigned for each unit in VTR. Then the combination type is known from the vector of vehicle types:

```
if karscode==['BA', 'DC', 'DA'] or karscode==['BA', 'DB']:
    combType="NordicCombination"
elif(karscode==['BC', 'DA_Semitrailer_Link', 'DA']):
    combType="BDouble"
```

In the code above, BA is a rigid truck, BC is a tractor, DC is a dolly, DA is a semitrailer, DB is a fulltrailer and DA_Semitrailer_Link is a link-trailer.

2.2 OpenPBS

OpenPBS is an open assessment tool for evaluating and simulating different PBSs for the combination vehicles. It is based on standard formats for dynamic models, Modelica (Modelica, 2020). The simplest possible vehicle models are used to keep the required number of parameters as low as possible but giving the results out in relevant accuracy. Open means it is available for everyone and it can be downloaded from GitLab for free. In the future, OpenPBS could be released on a web application for non-expert users, such as drivers. (Jacobson, et al., 2017, pp. 7-9) Since the source code is open, it can also be used by more advanced users such as truck and trailer manufacturers.

OpenPBS is based on Modelica language. Improvements to OpenPBS in this thesis are made using OpenModelica simulation software. OpenModelica is an open-source modelling and simulation environment based on Modelica language (OpenModelica, 2020). Using open-source software is consistent to develop the simulation tool, which should be available for everyone.

The OpenPBS consists of Modelica packages. The package VehicleParameters specifies the vehicles in terms of vehicle parameters (see Table 3). Equations are included in the package VehicleModels that uses the vehicle parameters defined in VehicleParameters packages. PBS manoeuvres are described in the package Manoeuvres, where user can choose the vehicle or redefine the manoeuvre or simulation model. This package structure means that all simulation model parts can be modified and improved without affecting each other. (Santahuhta, 2019, pp. 15-16)

2.3 Reference tool

The other existing PBS tool in Sweden is called "Reference tool" in this thesis. The Reference tool is based on "home-made" time integration algorithms. It has been implemented in Python. It can handle only Nordic combinations and B-doubles. For each combination type, there is a Python code, which defines the additional model parameters, i.e. additional on top of VTR parameters. For calculating the PBS measures there is an individual Python script, which is used

via “call-function” after parameters are defined. Not all PBS measures are defined in the Reference tool.

The available version of those scripts were not parametrized, so one task was to parametrize the Python scripts of the Reference tool. After that the scripts were verified comparing results to online version of this tool.

3 Updates to OpenPBS

To use the same input parameters as the Reference tool, some modifications to OpenPBS Vertical forces model need to be made as explained in the following sections. Also, the differences in tyre models are studied and the tyre parameters in OpenPBS are modified to reach the sufficient match to the tyre model of the Reference tool. A Steady state Rollover Threshold model developed in the first PBS project (Kharrazi, S. et al., 2017) is implemented in the OpenPBS tool as presented in Chapter 5.1.1. Some fixes to OpenPBS Longitudinal model are made as well and Yaw damping model is modified to match the Reference tool's model.

3.1 Vertical forces

The Reference tool uses user defined axle loads, which is appropriate especially when the authorities carry out roadside checks. Therefore, the vertical axle forces F_z should be input parameter also in OpenPBS. In the earlier version of OpenPBS, each unit's total mass, m , and centre of gravity longitudinal position, X , which are difficult to access, were used as input parameter. To make this conversion possible, also vehicles kerb masses m_k are needed as an input parameter, to calculate total masses, CoG longitudinal positions and vertical coupling forces. Kerb masses are obtained from VTR by registration numbers.

OpenPBS parameters are defined in a matrix format, where the number of rows is the number of vehicle units and number of columns matches the maximum axle amount. Parameter definitions are presented in Ville Santahuhta's "Roll dynamics and tyre relaxation in heavy combination vehicle models for transient lateral manoeuvres" (2019, p. 49-54). For instance for a Nordic Combination, the axle loads are defined as:

$$F_z = \begin{bmatrix} 8 & 9 & 9 & 6 \\ 9 & 9 & 0 & 0 \\ 8 & 8 & 8 & 0 \end{bmatrix} \cdot 1000 \text{ kg} \cdot g,$$

where the first row is the axle loads of 4-axle rigid truck starting from the front axle. Respectively, second row is dolly's axle loads and third row semitrailer's axle loads. Zero means non-existing or lifted axle. Kerb masses are defined for

each vehicle unit of combination, so for the Nordic Combination the kerb masses could be presented as:

$$m_k = \{14240 \quad 2360 \quad 7810\},$$

where the first value is truck's, second dolly's and the last one is semitrailer's kerb mass.

For each vehicle unit there are 2 equilibrium equations, vertical forces, and pitch momentum. Hence, the maximum number of equilibrium equations is two times the number of units. Also, the mass of tractor and dolly is known because those do not carry any load by themselves. In Table 1 is presented that if number of vehicle units is higher than 2, there are not enough equations to solve all unknowns.

Table 1. Vertical force equations and unknowns.

Combination	Number of units	Number of equations	Number of unknowns	Definition
Tractor with semitrailer	2	4	4 (m_2, X_1, X_2, F_{CZ1})	well determined
Nordic combination	3	6	7 ($m_1, m_3, X_1, X_2, X_3, F_{CZ1}, F_{CZ2}$)	under determined
B-double	3	6	7 ($m_2, m_3, X_1, X_2, X_3, F_{CZ1}, F_{CZ2}$)	under determined
A-double	4	8	9 ($m_2, m_4, X_1, X_2, X_3, X_4, F_{CZ1}, F_{CZ2}, F_{CZ3}$)	under determined
DoubleCAT	3	6	8 ($m_1, m_2, m_3, X_1, X_2, X_3, F_{CZ1}, F_{CZ2}$)	under determined
Tractor with semitrailer and CAT	3	6	7 ($m_2, m_3, X_1, X_2, X_3, F_{CZ1}, F_{CZ2}$)	under determined

The centre of gravity position of an unladen vehicle is not available from VTR due to EU legislation. Therefore, an assumption is needed. In the Reference tool, the drawbar's vertical coupling force is assumed to be zero. This is true when the drawbar is pitch moment free. For center-axle trailers and dollies, it is stipulated

that the vertical load of the drawbar shall not exceed 1000 kg or 10 % of the total mass of the trailer, whichever is less (UN/ECE 2018 2.13). Normally, the drawbar's vertical load is very low, so it can be assumed to be zero also in OpenPBS. Then there are as many variables as equations in most cases.

To solve total masses m , CoG positions X and vertical coupling forces in OpenPBS using vertical axle loads F_z and kerb masses m_k from VTR, first step is to calculate load masses m_l for every vehicle unit. Following equations are presented with the matrix indexes i (the unit number) and j (the axle number). So, for tractor and dolly load masses are zero. For rigid truck and centre-axle trailer:

$$m_l[i] = \frac{\sum_{j=1}^{j=na} F_z[i,j]}{g} - m_k[i], \quad (1)$$

and for semitrailer connected to a dolly or tractor, the load mass is:

$$m_l[i] = \frac{\sum_{j=1}^{j=na} F_z[i,j] + \sum_{j=1}^{j=na} F_z[i-1,j]}{g} - (m_k[i] + m_k[i-1]) \quad (2)$$

In Equation (1) from the sum of axle forces divided gravity has been deducted the vehicle's own mass. In (2) there is also tractor's or dolly's vertical axle forces and kerb masses, because they carry part of semitrailer's mass. This is explained in Figure 3.

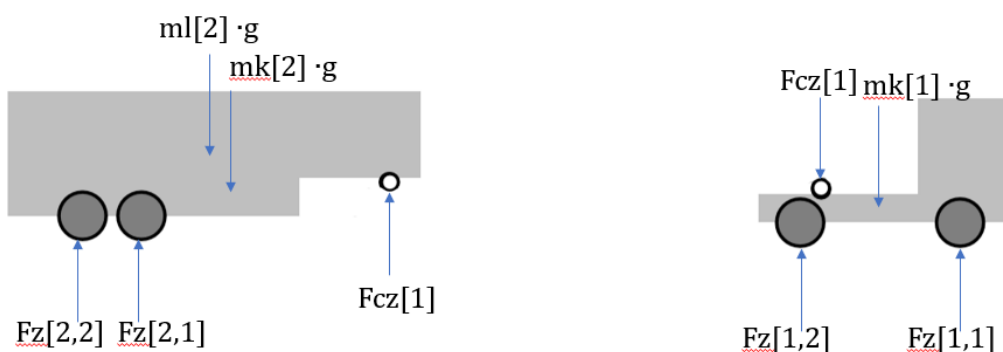


Figure 3. Free-body diagram of tractor and semitrailer.

The equations above use the assumption, that the drawbar's vertical force is zero. For the B-double combination, different assumption is needed, because there is not drawbar coupling at all. The Reference tool assumes that vertical force of link-trailer's rear coupling is $15600 \text{ kg} \cdot g$, which could be possible if the rear semitrailer is fully loaded. Therefore, this assumption leads to significantly

incorrect results when driving with partial loads or unladen. Therefore, a different assumption was formed that considers the load of the combination. Assuming that if rear semitrailer is unloaded, the vertical coupling force on link-trailer's rear coupling is $1500 \text{ kg} \cdot g$ and when the semitrailer is loaded to its maximum allowed payload, the coupling force is $15600 \text{ kg} \cdot g$, a linear line can be formed as in Figure 4. The value of $15600 \text{ kg} \cdot g$ is used to be consistent to the Reference tool. For unloaded B-double, a nominal coupling force of $1500 \text{ kg} \cdot g$ is considered for the last semitrailer.

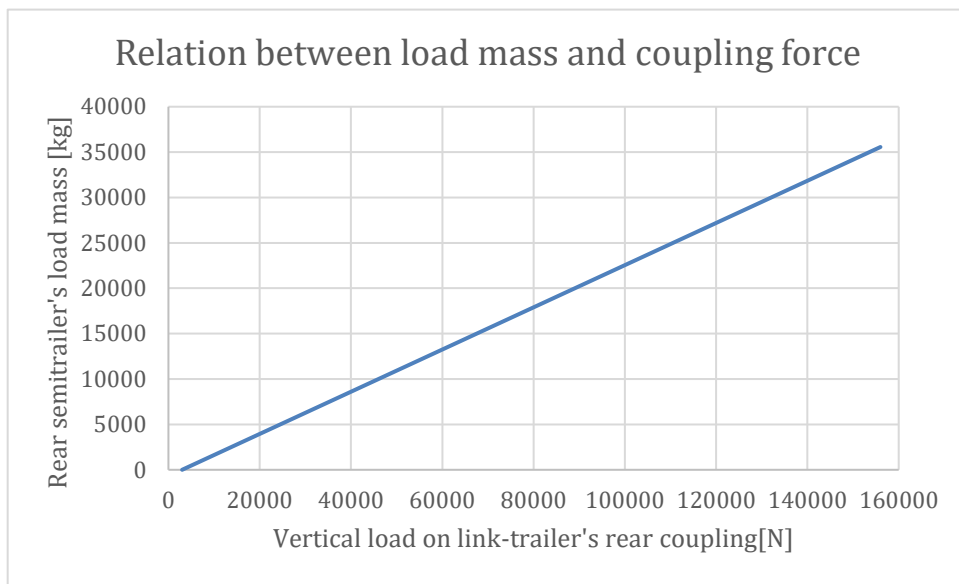


Figure 4. Linearized relation between semitrailer's load mass and link's vertical force of the rear coupling. Semitrailer's maximum load mass is assumed to 35000 kg .

According to Figure 4, the load mass of the B-double's rear semitrailer can be defined as:

$$m_l[i] = \frac{m_{l_{max}} \cdot (F_{cz}[i-1]/g - 1500 \text{ kg})}{(15600 \text{ kg} - 1500 \text{ kg})}, \quad (3)$$

where $m_{l_{max}}$ is the maximum load mass. It is the maximum allowed mass of the semitrailer minus the kerb mass.

Because tractor's load mass is zero, the load mass of the link-trailer can be calculated summing whole combinations' axle loads minus sum of kerb masses and the load mass of rear semitrailer as following:

$$m_l[i] = \frac{\sum F_z[i,j]}{g} - \sum m_k - m_l[i + 1]. \quad (4)$$

Total masses for every unit can be calculated as following:

$$m[i] = m_k[i] + m_l[i]. \quad (5)$$

Now, when all total masses are known, the vertical coupling forces can be solved from vertical equilibrium as:

$$F_{cz}[i] = \sum_{j=1}^{j=na} F_z[i,j] + F_{cz}[i-1] - (m_l[i] + m_k[i]) \cdot g. \quad (6)$$

Next, the centre of gravity longitudinal position X can be defined from pitch momentum equation around the vehicle unit's first axle as following:

$$X[i] = \frac{\sum(F_z[i,j] \cdot L[i,j]) + F_{cz}[i-1] \cdot A[i] + F_{cz}[i] \cdot B[i]}{m[i] \cdot g}. \quad (7)$$

where L is axle distances from the first axle of the unit, A is distance from the unit's first axle to the centre of unit's front coupling and B is distance from the unit's first axle to the centre of unit's rear coupling.

Solving vertical equations like this, there are the same number of equations and unknowns.

3.2 Tyre models

Both compared PBS tools use modified version of Pacejka's "magic formula", which is defined as:

$$\text{Side force} = y(x) = D \cdot \sin(C \cdot \text{atan}(B \cdot x - E \cdot (B \cdot x - \text{atan}(B \cdot x)))) + S_v. \quad (8)$$

$$\text{Slip} = x = X + S_H. \quad (9)$$

(Bakker, et al., 1987)

3.2.1 Tyre models in OpenPBS

OpenPBS tyre models are presented in Ville Santahuhta's "Roll dynamics and tyre relaxation in heavy combination vehicle models for transient lateral manoeuvres" (Santahuhta, 2020). Linear tyre model will follow the simple equation for the lateral tyre force:

$$F_y = -C_{st} \cdot \alpha \quad (10)$$

where C_{st} is tyre cornering stiffness and α is slip angle of tyre.

The non-linear tyre model is a simplified version of Pacejka's "magic formula". Simplification aims to keep the number of parameters as low as possible. Tyre steady state lateral force will be:

$$F_{yt} = F_{ZT} \cdot u_y \cdot \sin \left(C \cdot \text{atan} \left(\frac{CC_y}{C \cdot u_y} \cdot \alpha_y \right) \right) \quad (11)$$

where F_{ZT} is tyre vertical force, u_y is the maximum lateral force coefficient (peak friction coefficient), C is a shape factor, CC_y is a tyre cornering coefficient and α_y is slip angle.

Tyre shape factor C is defined as:

$$C = 2 \left(1 + \frac{\text{asin}(u_2)}{\pi} \right). \quad (12)$$

where u_2 is the ratio between infinite slide friction and peak friction and has a value of 0.8.

The maximum lateral force gradient is modified by the vertical force:

$$u_y = u_{y0} \cdot \left(1 + u_{gy} \cdot \frac{F_{ZT} - F_{ZT0}}{F_{ZT0}} \right), \quad (13)$$

where u_{y0} is tyre maximum lateral force coefficient at nominal load and u_{gy} is the maximum lateral force gradient at actual load.

Tyre cornering coefficient CC_y can be defined from the cornering coefficient at nominal tyre force CC_{y0} and maximum cornering coefficient gradient ccg_y as follows:

$$CC_y = CC_{y0} \cdot \left(1 + ccg_y \cdot \frac{F_{ZT} - F_{ZT0}}{F_{ZT0}} \right). \quad (14)$$

Both tyre models, linear and non-linear also include the tyre relaxation, which is a delay how fast the tyre will reach steady state conditions. For the long combinations, this is important parameter for simulations, because the delay will propagate through the vehicle. In linear tyre model, the tyre relaxation is modelled as a first order filter to the tyre force generation as following:

$$\dot{F}_y = \frac{|v_x|}{L_r} \cdot (f(s_y, F_z, \mu, \dots) - F_y) \quad (15)$$

where L_r is the relaxation length and set to standard value of 0.4 and $f(s_y, F_z, \mu, \dots)$ is tyre force in steady state conditions.

Tyre relaxation in non-linear tyre model is applied to the tyre slip angle as follows:

$$\frac{d}{dt} a'_y = \frac{|v_x|}{L_r} \cdot (\alpha_y(v_x, v_y, \omega_z, \dots) - a'_y), \quad (16)$$

where $\alpha_y(v_x, v_y, \omega_z, \dots)$ is the tyre slip angle in steady state conditions and a'_y is the slip angle with the tyre relaxation.

3.2.2 Tyre model in the Reference tool

Tyre model of the Reference tool is simplified version of Pacejka's "Magic Formula 1989", in which the tyre lateral force is calculated as:

$$F_y = D \cdot \sin(C \cdot \text{atan}(B \cdot (1 - E) \cdot \alpha + E \cdot \text{atan}(B \cdot \alpha))) \quad (17)$$

where D is peak force, C is shape factor, B is slip stiffness and E is curvature and α is slip angle.

Magic formula factors are defined as follows:

$$D = a_2 \cdot F_z^2 + a_3 \cdot F_z, \quad (18)$$

$$C = a_1, \quad (19)$$

$$B = \frac{k \cdot a_4 \cdot \sin\left(2 \cdot \text{atan}\left(\frac{F_z}{a_5}\right)\right)}{C \cdot D} \text{ and} \quad (20)$$

$$E = a_7 \cdot F_z + a_8. \quad (21)$$

where F_z is tyre vertical force and k is a modification factor. Tyre parameters $a_1 \dots a_8$ are presented in Table 2.

Modification factor k is determined as:

$$k = \frac{KC_{nom} \cdot F_{znom}}{C_{\alpha}(F_{znom})} \quad (22)$$

where KC_{nom} is tyre cornering coefficient, F_{znom} is tyre nominal load value (see Table 7) and $C_{\alpha}(F_{znom})$ is cornering stiffness for reference tyre, which is defined as:

$$C_{\alpha} = a_4 \cdot \sin\left(2 \cdot \text{atan}\left(\frac{F_{znom}}{a_5}\right)\right) \cdot \frac{180^\circ}{\pi}. \quad (23)$$

Table 2. Parameters used in the simplified tyre magic formula in the Reference tool.

Parameter	Definition	Unit	Value
a1	Shape factor	-	1.3831733
a2	Load influence on lateral friction coefficient	1/kN	-2.8169800
a3	Lateral friction coefficient	-	800.49500
a4	Change of stiffness with slip	N/deg	7063.9207
a5	Change of progressivity of stiffness / load	1/kN	115.40403
a7	Curvature change with load	-	-0.0406229
a8	Curvature factor	-	1.2440395

Slip angle α_1 for truck's tyres is defined as:

$$\alpha_1 = sc_1 \cdot \delta - \frac{v_{y1} + \omega_{z1} \cdot l_1}{v_{x1}}, \quad (24)$$

where sc_1 is one for steerable axles and 0 for other axles, δ is steering angle, $x(1)$ is unit's lateral velocity, $x(2)$ is yaw rate of unit, l_1 is axle longitudinal position relative to CoG and v_x is longitudinal velocity.

Tyre slip angles for towable units are defined respectively as:

$$\alpha_{i+1} = sc_{i+1} \cdot \psi_i - \frac{v_{y_{i+1}} + \omega_{z_{i+1}} \cdot l_{i+1}}{v_{x_{i+1}}}, \quad (25)$$

where ψ is coupling angle between consecutive units, which is defined as:

$$\psi[i] = \varphi[i] - \varphi[i + 1], \quad (26)$$

where φ is yaw angle. (Aurell 2017, pp. 16-20)

3.2.3 Differences in tyre models

The following differences can be found in tyre models between compared tools:

- No tyre relaxation in the Reference tool
- OpenPBS non-linear tyre model is more simplified:
 - o No curvature factor E
 - o Less tyre parameters needed

Tyre model comparison is presented in Figure 5. Tyre lateral force curves are plotted as the function of the slip angle. Tyre lateral forces are defined for 50 %, 100 % and 150 % tyre vertical force of tyre nominal load value. Used nominal load value in comparison was 45 kN. The maximum slip angle in simulations of this thesis was 5°. Figure 5 shows, that the tyre curves do not match properly in

any case. Tyre lateral force becomes higher in OpenPBS tyre model almost every time.

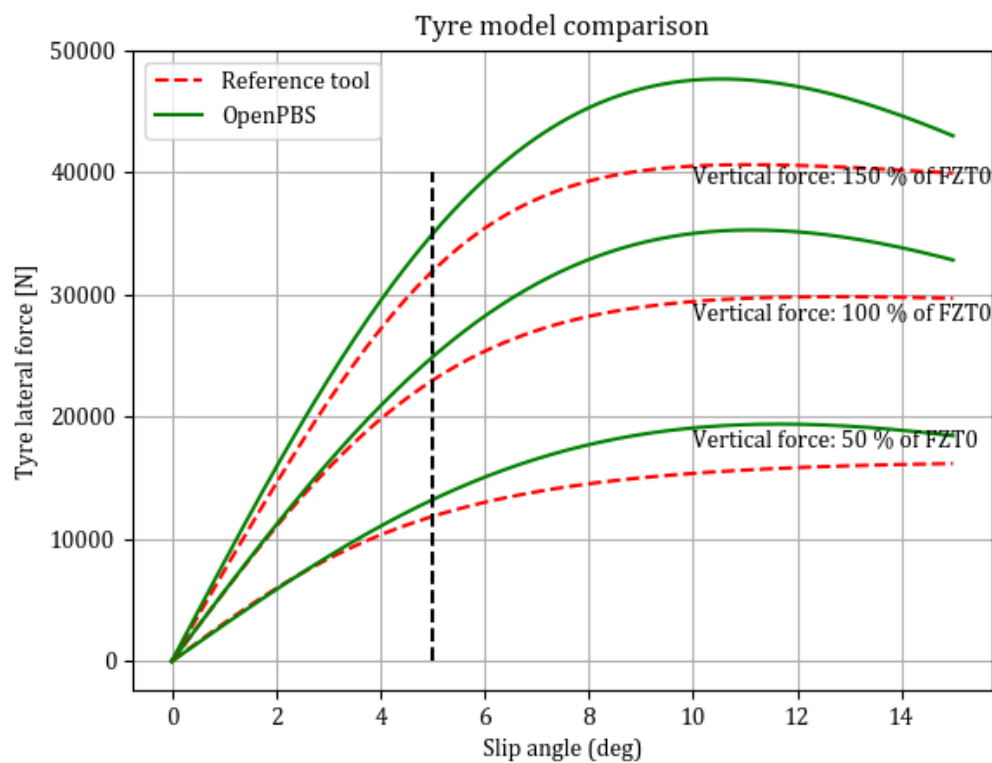


Figure 5. Tyre lateral forces as the function of the slip angle.

For the comparison, efforts were made to align the tyre models. This should be done modifying the tyre parameter of OpenPBS. First step was to align the cornering coefficients. On the left of Figure 6 are cornering coefficients with the existing tyre parameters in OpenPBS and on the right the aligned cornering coefficients. To reach the best possible match in the cornering coefficient, tyre parameter cc_{gy} was set from -0.1 to -0.25. The parameter cc_{gy} is the tyre maximum cornering coefficient gradient. Better match was not possible without modifications to tyre equations.

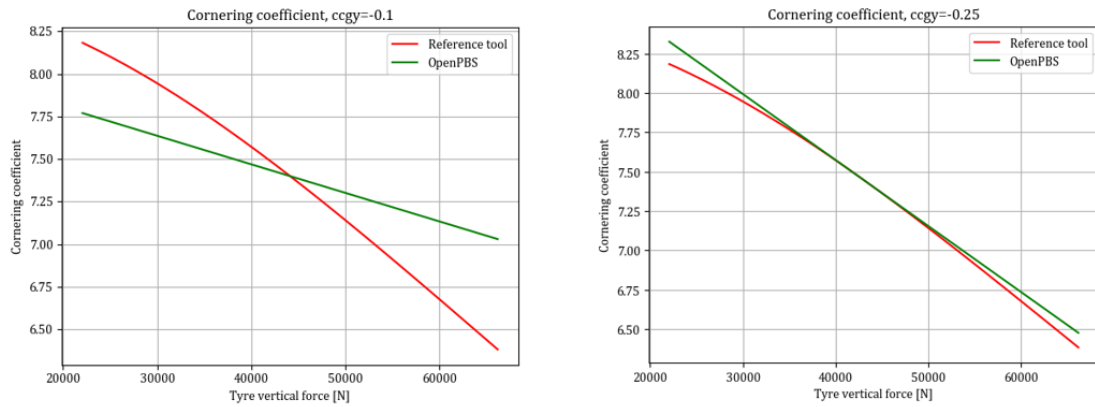


Figure 6. Cornering coefficients before and after parameter optimization.

After the cornering coefficients were aligned, the other tyre parameters in OpenPBS were chosen so the tyre lateral forces would match between the tools. The following modifications to the tyre parameters of OpenPBS were made compared with Santahuhta's thesis (Santahuhta, 2020):

- $ccg_y -0.1 \rightarrow -0.25$
- $ug_y -0.2 \rightarrow -0.18$
- $u_{y0} 0.8 \rightarrow 0.68$
- $CC_{y0} 7.4 \rightarrow 7.6$

The shape factor C of OpenPBS in (12) was modified to vary with load as follows:

$$C = C_0 + \frac{F_{ZT}}{F_{ZT0}}, \quad (27)$$

where C_0 is set to value of 0.7.

Tyre lateral forces as the function of the slip angle after parameter optimization are presented in Figure 7. Now the match between the tyre curves was sufficient.

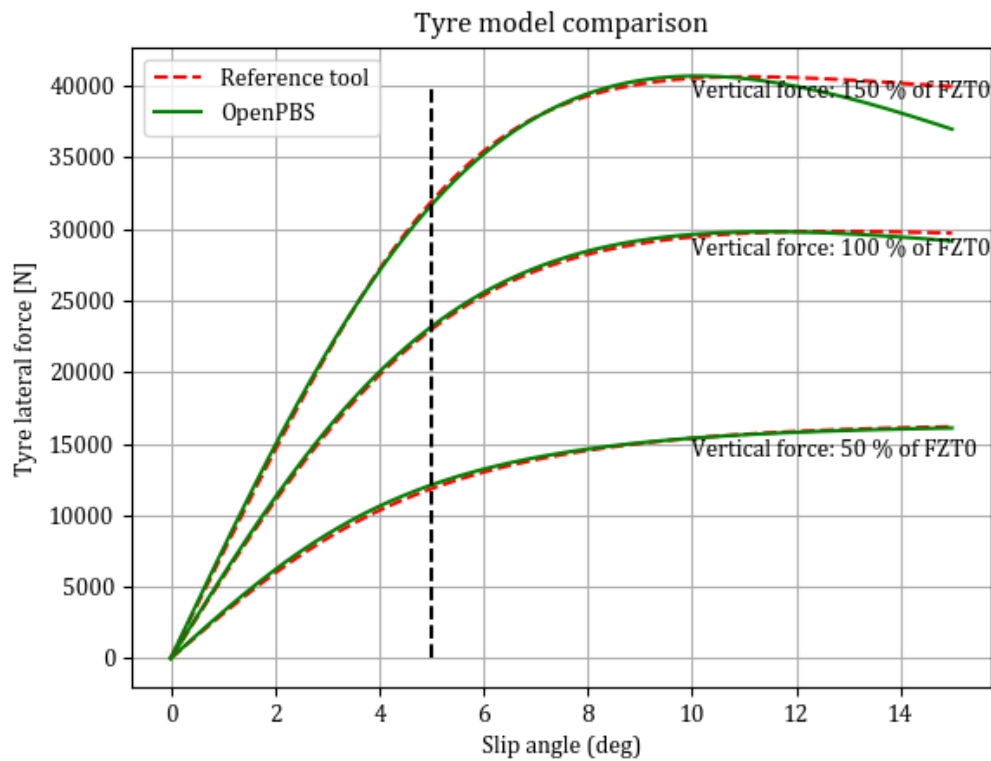


Figure 7. Aligned tyre models with the load depending C factor.

However, it was noticed that the match between the tyre models was not so good with the different tyre nominal load values. Therefore the tyre model of the Reference tool was implemented in the OpenPBS and the tools were run with the same tyre model for the sake of comparison.

3.3 Handling fulltrailers

OpenPBS can not handle fulltrailers as one unit, because it includes an articulation point (turntable). Therefore, fulltrailers are splitted to a dolly and semitrailer in Python code. First, all vehicle parameters are calculated for the whole fulltrailer. Then those defined OpenPBS vehicle parameters for fulltrailer are divided for a “virtual dolly” and semitrailer. To make this division possible, two assumptions are needed. The mass of the virtual dolly is assumed to be 2200 kg as in the Reference tool and the length of the virtual dolly is assumed to be 5 metres.

4 Parameter conversion

Conversion of the VTR parameters to OpenPBS vehicle parameters in Python script is presented in this chapter. Not all parameters can be specified straight from the VTR vehicle data, so lots of assumptions need to be made and nominal values considered. OpenPBS vehicle parameters are presented in Table 3.

Table 3. OpenPBS vehicle parameters and definitions. Vehicle parameters marked with an asterisk are added in this thesis to OpenPBS.

Parameter	Definition
nu	number of units
na	maximum number of axles per unit
L	Axle distances from first axle of unit
W	Track width of each axle
A	Distance to unit's front coupling from first axle of unit
B	Distance to unit's rear coupling from first axle of unit
FOH	Unit's front overhang from first axle of unit
ROH	Unit's rear overhang from first axle of unit
driven	Defines driven axle(s)
axlegroups	Defines which axlegroup each axle belongs to
I_z	Inertia of unit in Yaw-plane
I_x	Roll inertia of unit
h	Centre of gravity height of each unit
h _{RC}	Roll centre height of each unit
h _C	Height of each coupling
max_engine_power	Maximum engine power
max_thrust_force_vx0	Maximum thrust force from powertrain
k_s	Suspension roll stiffness per axle
c_s	Suspension roll dampening per axle
K_{cr}	Roll stiffness of each coupling
C_{cr}	Roll dampening of each coupling
FZT0	Tyre nominal force
CCy0	Cornering coefficient of each axle
L_r	Tyre relaxation length
uy0	Tyre maximum lateral force coefficient
u _{gy}	Tyre maximum lateral force gradient
u ₂	Tyre slide friction ratio
cc _{gy}	Tyre maximum cornering coefficient gradient
CYT*	Tyre lateral stiffness
CZT*	Tyre vertical stiffness
rfc*	Defines roll-free couplings
W_e^*	Effective track width for SRT calculations
F_z^*	Vertical axle loads
m_k^*	Vehicle kerb mass

4.1 VTR parameters

The parameters from the vehicle register are listed in the following Table 4. Strike through parameters are not used in parameter conversion.

Table 4. VTR unit parameters.

Parameter name	Description
vtr1	Total mass [kg]
vtr2	Coupling distances 1/2 [mm]
vtr3	Distance between couplings [mm]
vtr4	Load bay length [mm]
vtr5	Rear overhang [mm]
vtr6	Tyre amount
vtr7	Axle amount
vtr8	Distance between consecutive axles [mm]
vtr9	Tyre dimensions
vtr10	Vehicle length [mm]
vtr11	Vehicle height [mm]
vtr12	Vehicle width [mm]
vtr13	Kerb mass [kg]
vtr14	Tyre load index
vtr15	Max coupling load [kg]
vtr16	Engine power [kW]
vtr17	Number of driven axles

Coupling distances (vtr2) are defined from the front end of trucks and from the rear end of the trailers. Rear overhang (vtr5) is specified from the rear axle of unit.

4.2 Conversion to OpenPBS vehicle parameters

OpenPBS parameters are in matrix format, which size is defined with the parameters number of units nu and maximum number of axles na . In equations below, index i will be the number of unit and j the number of axle. Some parameters are determined for each unit when the parameter will be a vector with a size of nu .

4.2.1 Number of units (nu) and maximum number of axles (na)

Number of units (nu) is defined from the size of the vector, which consists of the registration numbers of the combination vehicle. This is defined in Python code e.g. as:

```
karscode= (["XXX123", "YYY456", "ZZZ789"] )  
nu=len(karscode)  
→ nu=3
```

It should be noted, that number of units will be 3 also for Nordic combination with a fulltrailer, because there is a “virtual dolly” (see Chapter 3.3). Maximum number of axles (na) is defined with the operator “max” of the vtr7 vector, which consists numbers of axles of each unit involved. For instance, the maximum number of axles could be:

```
vtr7=( [4, 2, 3] )  
na=max(vtr7)  
→ na=4
```

4.2.2 Axle positions from first axle of unit (L)

Axle distances are defined in VTR between consecutive axles. In OpenPBS vehicle parameter L is an axle position relative to unit’s first axle and positive in forward direction. From Figure 8 follows, that the axle positions L can be defined as:

$$\begin{aligned} L[1,1] &= 0 \\ L[1,2] &= 0 - vtr8(1) \\ L[1,3] &= 0 - (vtr8(1) + vtr8(2)) \\ L[1,4] &= 0 - (vtr8(1) + vtr8(2) + vtr8(3)) \end{aligned} \tag{28}$$

and so on for each unit. It should be noted that values should be also converted to meters, and if some column of vtr8 is zero, the corresponding L value should be zero too.

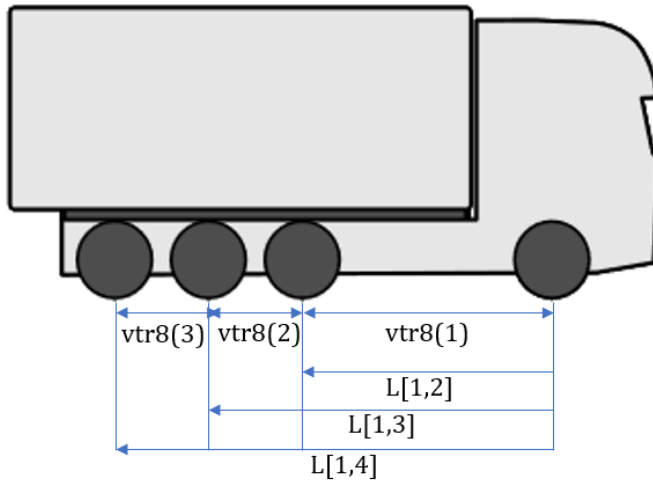


Figure 8. Axle positions in VTR and OpenPBS. Parameter L defines axle positions relative to the first axle and the positive direction is forward, measured from first axle.

4.2.3 Track width of each axle (W)

Track width W is calculated as in Aurell's report (Aurell, 2017). For the axles with single tyres, track width will be

$$W[i, j] = A_w - S, \quad (29)$$

where A_w is axle width, which is defined to be 2.5 meters and S is tyre width (from vtr9).

Respectively, track width W for the axles with double tyres is defined as:

$$W[i, j] = A - 2 \cdot S - E, \quad (30)$$

where E is the gap between the double tyres and is assumed to 0.03 m.

Number of tyres of each unit are usually known from VTR, but which axles have double tyres, is not specified. Number of tyres is not OpenPBS vehicle parameter, but it is used to calculate other input parameters. It can be defined from the nominal load values of tyres FZT0 (see Table 7). If FZT0 times 2 is smaller than maximum axle load, there must be double tyres, otherwise single tyres. This is defined in the following:

if $2 \cdot FZT0 < Fz_{max} \rightarrow$ number of tyres = 4,
else if $FZT0 = 0 \rightarrow$ number of tyres = 0,
else number of tyres = 2.

4.2.4 Coupling distances (A and B)

OpenPBS parameters A and B are the coupling distances relative to the first axle, A is to the front coupling and B to the rear coupling. VTR coupling distances (vtr2) are defined from the front end to the centre of coupling for the trucks/tractors and from the rear end to the centre of coupling for trailers. If a vehicle has two couplings (dolly or link-trailer), both distances should be from the rear end of vehicle to the centre of coupling. Also, a tractor can have two coupling distances, which means there is a transferable fifth wheel. Then the mean value is used for calculations. For the older vehicles (Nationellt), coupling distances from the first axle to the centre of coupling. (TSFS 2009:59, p. 19-20) For the trucks/tractors A=0 and for the towable vehicles, A can be defined as:

$$A[i] = vtr2_1[i] - vtr5[i] - \sum_{j=1}^{j=na} vtr8[i, j]. \quad (31)$$

Parameter B can be defined as follows for trucks/tractors:

$$B[i] = - \sum_{j=1}^{j=na} vtr8[i, j] + vtr5[i] - vtr10[i] + vtr2[i], \quad (32)$$

and if a tractor has a transferable fifth wheel, the mean value of vtr2 should be used in (32). For semitrailers, link-trailers and centre-axle trailers the parameter B should be defined respectively:

$$B[i] = - \sum_{j=1}^{j=na} vtr8[i, j] + vtr5[i] - vtr2[i], \quad (33)$$

and for dollies:

$$B[i] = - \frac{\sum_{j=1}^{j=na} vtr8[i, j]}{2}. \quad (34)$$

4.2.5 Front and rear overhang (FOH and ROH)

FOH and ROH are parameters which describe the front and rear overhang of the vehicle as a distance from the first axle of vehicle. $vtr5$ describes the rear overhang from the rear axle of vehicle, so ROH can be defined for each unit as follows:

$$ROH[i] = -(vtr5[i] + \sum_{j=1}^{j=na} vtr8[i,j]). \quad (35)$$

FOH can be defined using equation (35):

$$FOH[i] = vtr10[i] + ROH[i]. \quad (36)$$

4.2.6 Vertical axle loads (F_z)

Vertical axle loads F_z are defined by user. In this thesis, the following axle loads are used:

- Nordic combination with a converted dolly:
 $F_z = [8, 9, 9, 6; 9, 9, 0, 0; 8, 8, 8, 0] \cdot 1000 \text{ kg} \cdot g$
- Nordic combination with a fulltrailer:
 $F_z = [8, 9, 9, 6, 0; 9, 9, 8, 8, 8] \cdot 1000 \text{ kg} \cdot g.$
- B-double:
 $F_z = [7, 11.5, 7.5; 8, 8, 8; 8, 8, 8] \cdot 1000 \text{ kg} \cdot g,$

Thereby, the total mass of the vehicle will be the maximum allowed 74 tons and should simulate the worst scenarios.

4.2.7 Driven and axlegroups

Number of driven axles is specified in VTR, but not which axles are driven.

Hence, assumptions are made as follows:

- If there are two driven axles in truck, driven axles are always ones with the double tyres
- When there is one driven axle, it is assumed to be always the front one of the double mounted axles, because that is most common
- Trucks with only one axle with double tyres, it is always the driven axle.
- In special cases, if truck has driven axle with single tyres or front wheel drive, driven axles must be defined by user.

Parameter driven defines driven axles in a matrix format with Boolean values. In Modelica format, parameter driven for the Nordic combination could be:

$driven = [false, true, true; false, false, false; false, false, false]$.

The parameter called axlegroups is an integer determining which axlegroup each axle belongs to. In practice, rear axles of truck belong to second axlegroup, and every other axles of the vehicle belong to the first axle group. When there is no axle, the value will be zero. Parameter axlegroup for the Nordic combination above will be:

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

4.2.8 Inertia of unit in Yaw-plane (I_z)

Inertia on the yaw plane of each unit is calculated using the same assumptions as Aurell in his report (Aurell, 2017) as follows:

$$I_z[i] = I_l[i] + m_l[i] \cdot (x_l[i] - x_0[i])^2 + I_r[i] + m_r[i] \cdot (x_r[i] - x_0[i])^2 + I_{hm}[i] + m_{hm}[i] \cdot (x_{hm}[i] - x_0[i])^2 + \sum(I_a[i] + m_a[i] \cdot (x_a[i] - x_0[i])^2), \quad (37)$$

where I_l , m_l , and x_l are inertia, mass and CoG position of load and respectively I_r , m_r and x_r for frame, I_a , m_a and x_a for axles and I_{hm} , m_{hm} and x_{hm} for cabin-engine inertia, mass and CoG position.

The additional inertia variables I_l , I_r , I_a and I_{hm} in Equation (37) are defined as:

$$\begin{aligned} I_l[i] &= \frac{m_l[i]}{12} \cdot (vtr4[i]^2 + vtr12[i]^2) \\ I_r[i] &= \frac{m_r[i]}{12} \cdot (vtr10[i]^2 + br[i]^2) \\ I_{hm}[1] &= 2000 \\ I_a[i, j] &= m_a[i, j] \end{aligned} \quad (38)$$

where br is the width of frame and is set to be $vtr12$ for trucks and 1 m for trailers. Only trucks have a cabin-engine inertia, so for the trailers I_{hm} is zero. It should be noted that Equations (37) and (38) do not apply to dollies, and its inertia on yaw plane is set to be the same as the mass of dolly ($vtr13$).

The load masses m_l of each unit are defined in Equations (1) and (2). The mass of cabin-engine m_{hm} is assumed to be 3500 kg and the frame masses m_r are assumed as:

$$\begin{aligned}
m_r[i] &= 2500 \text{ kg for tractors} \\
m_r[i] &= vtr13[i] - \sum_{j=1}^{j=na} m_a[i, j] - m_{hm} \text{ for trucks} \\
m_r[i] &= vtr10[i] \cdot 250 \text{ for semi-, link-, and fulltrailers} \\
m_r[i] &= vtr10[i] \cdot 200 \text{ for dollies and CATs}
\end{aligned} \tag{39}$$

The axle masses m_a for each axle of truck are defined as:

$$m_a[i, j] = \begin{cases} 700 \text{ kg if single tyres} \\ 900 \text{ kg if double tyres,} \\ 1300 \text{ kg if driven axle} \end{cases}$$

and for trailers:

$$m_a[i, j] = \begin{cases} 700 \text{ kg if single tyres} \\ 800 \text{ kg if double tyres} \end{cases}$$

Vehicle sprung mass m_{fj} is calculated as following:

$$m_{fj}[i] = m_{hm}[i] + m_r[i]. \tag{40}$$

The CoG positions of the different masses in (37) are based on the following assumptions. The CoG position of the load will be:

$$\begin{aligned}
x_l[i] &= - \left(\sum_{j=1}^{j=na} vtr8[i, j] + vtr5[i] - \frac{vtr4[i]}{2} \right) \text{ for trucks} \\
x_l[i] &= \frac{vtr4[i]}{2} - vtr5[i] - \sum_{j=1}^{j=na} vtr8[i, j] \text{ for semi-, full - and CAT} \\
x_l[i] &= vtr10[i] - \frac{vtr4[i]}{2} - vtr5[i] - \sum_{j=1}^{j=na} vtr8[i, j] \text{ for linktrailers} \\
x_l[i] &= 0 \text{ for tractors and dollies}
\end{aligned} \tag{41}$$

The CoG position of cabin-engine mass x_{hm} is assumed to be zero and the axle mass CoG position x_a is the same as parameter L (see Equation (28)). The frame CoG position x_r can be defined as:

$$\begin{aligned}
x_r[i] &= - \left(\sum_{j=1}^{j=na} vtr8[i, j] + vtr5[i] - \frac{vtr10[i]}{2} \right) \text{ for trucks} \\
x_r[i] &= \sum_{j=1}^{j=na} vtr8[i, j] + vtr5[i] - \frac{vtr10[i]}{2} \text{ for trailers} \\
x_r[i] &= - \frac{\sum_{j=1}^{j=na} vtr8[i, j]}{2} \text{ for dollies}
\end{aligned} \tag{42}$$

The centre of gravity position of each vehicle is calculated in OpenPBS in Vertical Forces model (see Chapter 3.1), but because it is needed in yaw inertia equations, it is calculated in Python as follows:

$$x_0[i] = \frac{m_l[i] \cdot x_l[i] + m_r[i] \cdot x_r[i] - m_{hm}[i] \cdot x_{hm}[i] + \sum_{j=1}^{j=na} m_a[i,j] \cdot x_a[i,j]}{m_l[i] + m_r[i] + m_{hm}[i] + \sum_{j=1}^{j=na} m_a[i,j]} \quad (43)$$

4.2.9 Roll inertia of unit (I_x)

Inertia on the roll plane is calculated from a single equation from Jacobson's Vehicle Dynamics compendium (Jacobson, 2019) for each unit as follows:

$$I_x[i] = \frac{m[i] \cdot (vtr12[i]^2 + h[i]^2)}{12} \quad (44)$$

4.2.10 CoG, roll-centre and coupling heights of each unit (h , h_{RC} , h_C)

The centre of gravity height of each unit h is calculated from the following equation:

$$h[i] = \frac{\sum_{j=1}^{j=na} h_a[i,j] \cdot m_a[i,j] + h_{fj}[i] \cdot m_{fj}[i] + h_l[i] \cdot m_l[i]}{\sum_{j=1}^{j=na} m_a[i,j] + m_{fj}[i] + m_l[i]}, \quad (45)$$

where h_a is the axle CoG height, h_{fj} is the sprung mass CoG height and h_l is the load CoG height.

The axle CoG height h_a is the same as a tyre dynamic radius. Tyre radius is calculated using VTR parameter $vtr9$, which consists of the tyre dimensions tyrewidth, tyreprofile and wheel diameter. Tyre radius r_t for each axle will be:

$$r_t[i,j] = \frac{\left(\frac{\text{tyrewidth}[i,j]}{1000} \cdot \frac{\text{tyreprofile}[i,j]}{100} + \text{wheeldiameter}[i,j] \cdot \frac{25.4}{1000} \right)}{2} \quad (46)$$

and the tyre dynamic radius r_d is defined for each axle as:

$$r_d[i,j] = h_a[i,j] = r_t[i,j] - 0.04 \text{ m}. \quad (47)$$

Sprung mass CoG height h_{fj} for each unit can be defined using Equations (46) and (47) as following:

$$h_{fj}[i] = \frac{\sum_{j=1}^{j=na} (r_t[i,j] + r_d[i,j])}{vtr7[i]}. \quad (48)$$

The CoG height of the load for each unit h_l is calculated as:

$$h_l[i] = \frac{(h_f[i] + h_{fj}[i] + h_{hj})}{2}, \quad (49)$$

where h_f load height from the ground to the top of the load and it is defined by user. In the simulations of this thesis, the maximum allowed height, 4.4 m is used as a height of the load. Height of the subframe h_{hj} is set to be 0.2 meter.

Parameter h_{RC} is the roll-centre height of each unit and is defined as a mean value of h_a of each unit:

$$h_{RC}[i] = \frac{\sum_{j=1}^{j=na} h_a[i,j]}{vtr7[i]}. \quad (50)$$

Parameter h_c describes the height of each coupling between units. For the fifth wheel coupling the height should be 1 meter, and 0.5 meters for drawbar coupling. Thus, h_{RC} will be e.g. for Nordic combination {0.5, 1.0} in Modelica format.

4.2.11 Maximum engine power and maximum thrust force

Vehicle maximum engine power comes directly from $vtr16$, but it should be converted from kW to W. The maximum force on the driven wheels from the powertrain can be defined as:

$$max_thrust_force_{vx0} = \frac{maxenginepower}{2 \cdot \pi \cdot \frac{w_{Tmax}}{60}} \cdot \frac{Ratio}{r_t} \quad (51)$$

where w_{Tmax} is the engine speed at maximum engine torque and is assumed to 1300 rpm. Ratio is the whole powertrain ratio from engine to wheel and used the typical value of 30 (Jacobson 2016, Table 1.6) and r_t is the wheel radius.

4.2.12 Suspension and coupling roll stiffness and damping (k_s , c_s , K_{cr} and C_{cr})

The suspension roll stiffness k_s of each axle are based on the first PBS project. For the front axle of trucks, k_s will be 400 kNm/rad and for the rear axles 1400 kNm/rad. For the axles of trailers k_s will be 1500 kNm/rad.

The suspension roll damping c_s will be based on the design rule of vertical oscillation damping. The design values for axle loads are used in (52) and (53) and the values are presented in Table 5.

Table 5. Design values of axle loads for suspension roll damping calculations and resulting values of roll damping.

Vehicle type	Design values for axle load	Resulting values of roll damping [kg·m ² /s]
4-axle rigid truck with tridem	(8e3 9e3 9e3 6e3) kg	(7.0e3 7.9e3 7.9e3 5.3e3)
4-axle rigid truck with 2 front axles	(7.5e3 7.5e3 9.5e3 9.5e3) kg	(6.6e3 6.6e3 8.4e3 8.4e3)
3-axle tractor with tandem	(7e3 9.5e3 9.5e3) kg	(6.2e3 8.4e3 8.4e3)
3-axle tractor with one axle driven	(7e3 11.5e3 7.5e3) kg	(6.2e3 10e3 6.6e3)
5-axle fulltrailer	(9e3 9e3 8e3 8e3 8e3) kg	(9.8e3 9.8e3 8.7e3 8.7e3 8.7e3)
3-axle semitrailer	(8e3 8e3 8e3e) kg	(8.7e3 8.7e3 8.7e3)
2-axle dolly	(9e3 9e3) kg	(9.8e3 9.8e3)
3-axle link-trailer	(8e3 8e3 8e3) kg	(8.7e3 8.7e3 8.7e3)

First the vertical stiffness per side is calculated as:

$$k_z[i, j] = 0.5 \cdot (2\pi \cdot f)^2 \cdot axleload[i, j] \quad (52)$$

where f is assumed to be 1.4 Hz for trucks and 1.2 Hz for trailers.

Then the vertical damping per side can be calculated assuming damping ratio as 20 %:

$$c_z[i, j] = 0.2 \cdot \sqrt{2 \cdot k_z[i, j] \cdot axleload[i, j]}. \quad (53)$$

Using Equations (52) and (53) the suspension roll damping will be:

$$c_s[i, j] = 2 \cdot c_z \cdot \left(\frac{W_d}{2}\right)^2, r \quad (54)$$

where W_d is distance between dampers and is assumed to be 1 m for trucks and 1.2 m for trailers.

Coupling roll stiffness K_{cr} is defined as a vector with a size of number of units minus 1. For the fifth wheel couplings, used K_{cr} value is 10^9 Nm/rad and zero for drawbar couplings. Coupling roll damping C_{cr} is defined respectively and used the value of zero for all couplings.

4.2.13 Tyre parameters (FZT0, CC_{y0}, L_r, u_{y0}, u_{gy}, u₂, cc_{gy}, CYT and CZT)

Tyre parameters CC_{y0}, L_r, u_{y0}, u_{gy}, u₂, cc_{gy} and CYT are in matrix format and have the size of [nu, na]. This is defined with one input variable, with zeros where there are no axes. Values used in this thesis are presented in Table 6.

Table 6. Tyre parameters with a certain value.

Tyre parameter	Definitions	Used value
CC _{y0}	Cornering coefficient [(N/rad)/N]	7.4
L _r	Relaxation length [m]	0.4
u _{y0}	Maximum lateral force coefficient [-]	0.8
u _{gy}	Maximum lateral force gradient [-]	-0.2
u ₂	Peak friction ratio on dry asphalt road [N/N]	0.8
cc _{gy}	Maximum cornering coefficient ratio [-]	-0.1
CYT	Lateral stiffness [kN/m]	300

The nominal load value FZT0 is defined for each tyre dimension. Values come from Aurell's report (Aurell, 2017). Also tyre vertical stiffnesses CZT are defined for each tyre dimension, and the values of CZT are based on the received data from Nokian Tyres. CZT values are valid for nominal inflation pressure. Values of FZT0 and CZT are presented in Table 7.

Table 7. Nominal load values FZT0 and vertical stiffnesses CZT for each tyre dimension (Aurell, 2017, Nokian Tyres).

Tyre dimension	Load Index	Inflation pressure [kPa]	FZT0 [kN]	CZT [kN/m]
245/70R17.5	143/141	875	26.7	730.4
255/60R19.5	143/141	900	26.7	790.4
265/70R19.5	143/141	850	26.7	765.3
285/70R19.5	150/148	900	32.9	819.8
275/70R22.5	148/145	900	30.9	853.8
295/80R22.5	154/149	850	36.8	855.3
315/70R22.5	156/150	900	39.2	941.3
315/80R22.5	156/150	850	39.2	880.3
355/50R22.5	156	900	39.2	1004.4
385/55R22.5	160	900	44.1	1058.6
385/65R22.5	160	900	44.1	1033.4
425/55R19.5	160	900	44.1	1009.4
425/65R22.5	165	825	50.5	1022.7
435/50R19.5	160	900	44.1	1027.9
445/45R19.5	165	850	50.5	1046.2

5 Compared PBS measures

In the Reference tool the following PBS measures are defined, which are also used for the comparison of the two PBS tools in this thesis

- Steady state Rollover Threshold (SRT): *SRT is motivated by the risk of that HCTs rolls over in long curves and thereby creates accidents*
- Gradeability: *Motivated by the risk of that HCTs drives very slow in uphill and thereby creates traffic congestion*
- Tracking Ability on Straight Path (TASP): *Motivated by the risk of that HCTs deviate too much laterally when driving on relatively straight roads and thereby creates accidents*
- Rearward Amplification (RWA): *Is motivated by the risk of that HCTs becomes difficult to manoeuvre laterally in high speed and thereby creates accidents.*
- Yaw Damping (YD): *YD is motivated by the risk of that HCTs becomes laterally unstable after changing lane and thereby creates accidents.*
- ~~(D, Dc and V-values for coupling)~~
 - not implemented in OpenPBS

(Jacobson, B. et al., 2017)

5.1 Steady state rollover threshold (SRT)

Steady-state rollover threshold (SRT) is the maximum lateral acceleration that the vehicle can take in steady-state cornering without rolling over. Passenger cars are often designed that those should be impossible to rollover, but for the heavy trucks it is possible on high road friction conditions. The rollover happens due to a high centre of gravity versus narrow track width. (Jacobson 2019, pp. 268-270)

5.1.1 Implemented SRT model

The following SRT model is based on the report “Performance based standards for high capacity transports in Sweden” (Kharrazi, S. et al., 2017) and it is considered for publication as an ISO standard. The model is similar to ECER111 (UNECE 2001) regulation for the calculation of SRT, but the tyre lateral stiffness is added, which has an influence on the effective track width.

To implement this model in OpenPBS, there is a need for four new vehicle parameters which are presented in Table 8.

Table 8. Added parameters for SRT calculations.

Parameter	Definition
rfc	Defines roll-free couplings e.g. $\{true, false\}$
CZT	Tyre vertical stiffness [N/m]
CYT	Tyre lateral stiffness [N/m]
W_e	Effective track width [m]

Parameter rfc defines roll-free couplings and it is used to define how many candidate SRTs will be calculated. The number of roll-coupled units nrc is defined in Modelica as follows:

$$nrc = 1 + \text{Modelica.Math.BooleanVectors.countTrue}(rfc)$$

Each roll-coupled unit gives one candidate of SRT and the vehicle's SRT is then

$$SRT = \min_i (SRT[i]) \quad (55)$$

where index i is the number of roll-coupled unit (nrc).

SRT is calculated for each roll-free coupled unit, so vehicle units which are coupled via fifth wheel, are treated as a single unit. For B-doubles, only one value of SRT is calculated, because both of combination's couplings are connected via fifth wheel. In Nordic Combination's case, first SRT is calculated for the rigid truck, and second is for dolly-semitrailer. It should be noted that the fifth wheel roll stiffness is included in equations only for tractors, when the rest of tractor is neglected.

The effective track width of each axle W_e is needed as a vehicle parameter for SRT calculations. If axle has single tyres, the effective track width is same as nominal track width W (see Figure 9.). If tyres are double-mounted, the effective track width for each axle should be calculated as

$$W_e = \sqrt{W^2 + (T + E)^2}, \quad (56)$$

where W is nominal track width, T is tyrewidth and E is gap between the double-mounted tyres.

The equivalent suspension roll stiffness of each axle with the respect of ground plane can be calculated as

$$k_{ES}[i,j] = k_s[i,j] \cdot \left(\frac{h_s[i]}{h_s[i] - h_{RC}[i]} \right)^2, \quad (57)$$

where k_s is suspension roll stiffness with the respect of the axle roll centre, h_s is height of sprung mass centre of gravity and h_{RC} is suspension roll centre above the ground.

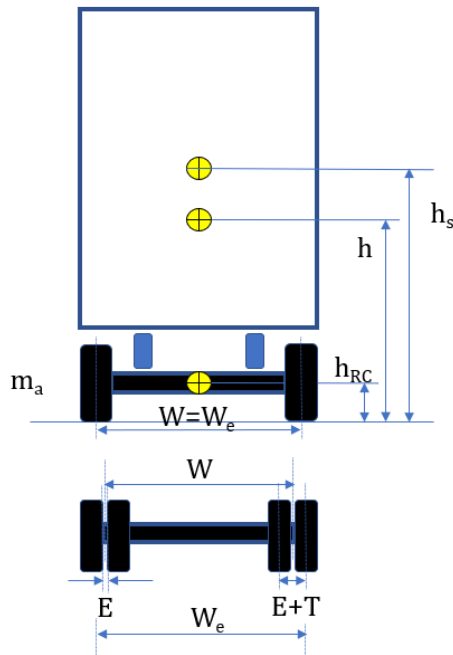


Figure 9. Definitions for effective track width W_e and sprung mass CoG height h_s .

The suspension roll stiffness k_s and suspension roll centre h_{RC} are OpenPBS vehicle parameters, and height of sprung mass CoG h_s can be calculated as

$$h_s = \frac{h \cdot m - m_a \cdot h_{RC}}{m - m_a}. \quad (58)$$

The roll stiffness from the tyre vertical stiffness is calculated as following:

$$k_{Ti} = \frac{CZT_i \cdot W_e^2}{2}, \quad (59)$$

and then the roll stiffness of an axle using Equations (57) and (59) will be:

$$k_i = \frac{k_{ESi} \cdot k_{Ti}}{k_{ESi} + k_{Ti}}. \quad (60)$$

Then the vehicle's roll stiffness is sum of (60):

$$k_v = \sum_{i=1}^{na} k_i. \quad (61)$$

but if the vehicle consists of roll-coupled units, then the fifth wheel representative roll stiffness K_{cr} should be added to (61):

$$k_v = \sum_{i=1}^{na} k_i + K_{cr}, \quad (62)$$

where $K_{cr} = 4 \cdot F_{cz}$, F_{cz} is fifth wheel's vertical force and factor 4 is expressed in m/rad.

The effective tyre lateral stiffness CYT_e of vehicle is a sum of tyre lateral stiffness of all axles:

$$CYT_e = \sum_{i=1}^{na} CYT_i. \quad (63)$$

The vehicle effective track width should be calculated as:

$$We_v = \frac{\sum_{i=1}^{na} We_i F_{z_i}}{\sum_{i=1}^{na} F_{z_i}}, \quad (64)$$

but if the vehicle consists fifth wheel coupled units, a fifth wheel representative track width W_{fw} should be included to the vehicle effective track width (64). The fifth wheel representative track width is defined as:

$$W_{fw} = \frac{\sum_{i=1}^{na} We_i}{na}, \quad (65)$$

when the effective track width will be:

$$We_v = \frac{\sum_{i=1}^{na} We_i F_{z_i}}{\sum_{i=1}^{na} F_{z_i} + F_{cz}} + \frac{W_{fw} F_{cz}}{\sum_{i=1}^{na} F_{z_i} + F_{cz}}. \quad (66)$$

Lateral acceleration, when each wheel lift-off happens, is

$$\frac{a_{yl_i}}{g} = \frac{F_{z_i} We_v}{2 \cdot (D_{Ki} \sum_{i=1}^{na} F_{z_i} h + \frac{(D_{Ki} (\sum_{i=1}^{na} F_{z_i} - Fu) hs)^2}{k_i - D_{Ki} \sum_{i=1}^{na} F_{z_i} hs} + \frac{F_{z_i}^2}{CYT_i})}, \quad (67)$$

where h is vehicle's CoG height and DK is distribution of the vehicle roll stiffness of each axle, which is defined as:

$$D_{Ki} = \frac{k_i}{k_v}, \quad (68)$$

and F_u is vehicle unsprung load, which is defined from the axle masses as follows:

$$F_u[i] = \sum_{j=1}^{j=na} m_a[i, j] \cdot g. \quad (69)$$

The vehicle theoretical maximum lateral acceleration is respectively:

$$\frac{a_{ym}}{g} = \frac{\sum_{i=1}^{na} F_{z_i} W e_v}{2 \cdot \left(\sum_{i=1}^{na} F_{z_i} h + \frac{\left(\sum_{i=1}^{na} F_{z_i} - F_u \right) h s}{k_v - D_{Ki} \sum_{i=1}^{na} F_{z_i} h s} + \frac{\sum_{i=1}^{na} F_{z_i}^2}{C Y T e} \right)}. \quad (70)$$

The steady state rollover threshold can be calculated as a linear interpolation between the lateral acceleration at the first wheel lifts off (67) and the theoretical maximum lateral acceleration (70):

$$a_{SRT} = \min_i \left(a_{ym} - \left(a_{ym} - a_{yl_i} \right) \frac{F_{z_i}}{\sum_{i=1}^{na} F_{z_i}} \right). \quad (71)$$

5.1.2 Reference SRT model

The following SRT model is used in the Reference tool. The model calculates two SRTs for Nordic combination, one for truck and one for dolly-semitrailer. Correspondingly, one value of SRT is defined for B-double. A simple roll-model of vehicle which is just about to turn over is presented in Figure 10. The equilibrium from Figure 10 can be determined as

$$m \cdot a_y \cdot H + m \cdot g \cdot H \cdot \varphi = m \cdot g \cdot \frac{T}{2}, \quad (72)$$

where H is centre of gravity height from ground [m], W is track width [m], φ is roll-angle [rad], m is mass [kg] and a_y is lateral acceleration [m/s²].

The method assumes that the axle stiffnesses are proportional to axle load. Vehicle's flexibility from tyres, suspension and frame is thought to be concentrated in one place, which is expressed by the rollgradient φ_g [rad/g]. Then the roll angle φ can be written:

$$\varphi = \varphi_g \cdot \frac{a_y}{g}. \quad (73)$$

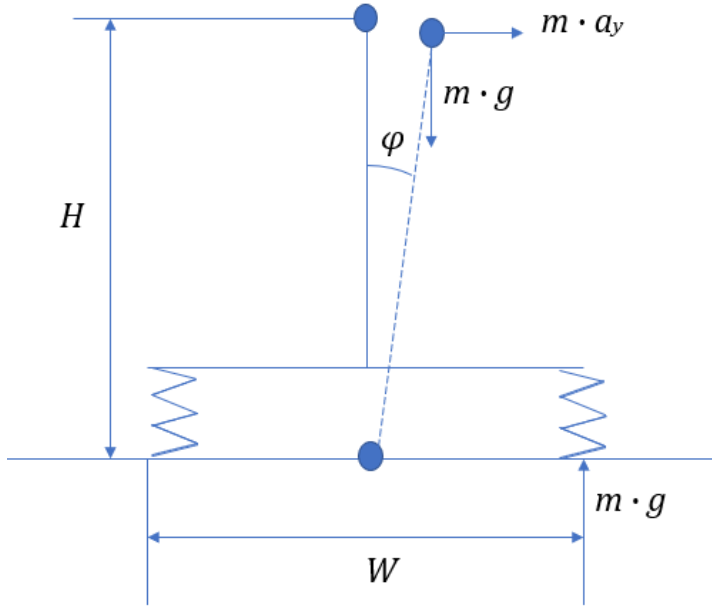


Figure 10. Free-body diagram of the vehicle that is just about to roll over.

The rollgradient also depends on the centre of gravity height H and the normalized rollgradient φ_{gH} [rad/g m]:

$$\varphi_g = \varphi_{gH} \cdot H. \quad (74)$$

The value of φ_{gH} is set to 0.15 for the trucks and 0.11 for trailers. Steady state rollover threshold can be solved from equations (72), (73) and (74) as following:

$$SRT = \frac{W}{2 \cdot H} \cdot \frac{1}{1 + \varphi_{gH} \cdot H} \cdot g. \quad (75)$$

5.2 Gradeability (GA)

Gradeability is the maximum slope on which the vehicle can be driven at a certain speed and given coefficient. It is calculated in the same simple way in OpenPBS and in the Reference tool.

The air resistance is:

$$F_d = \frac{1}{2} \cdot C_d \cdot A \cdot \rho \cdot v_x^2, \quad (76)$$

where C_d is air resistance coefficient, A is frontal area [m²], ρ is air density [kg/m³] and v_x is longitudinal velocity [m/s²].

Force on driven wheels from powertrain is calculated as:

$$F_m = \frac{\eta \cdot P_{max}}{v_x}, \quad (77)$$

where η is total efficiency of powertrain and P_{max} is max engine power [W].

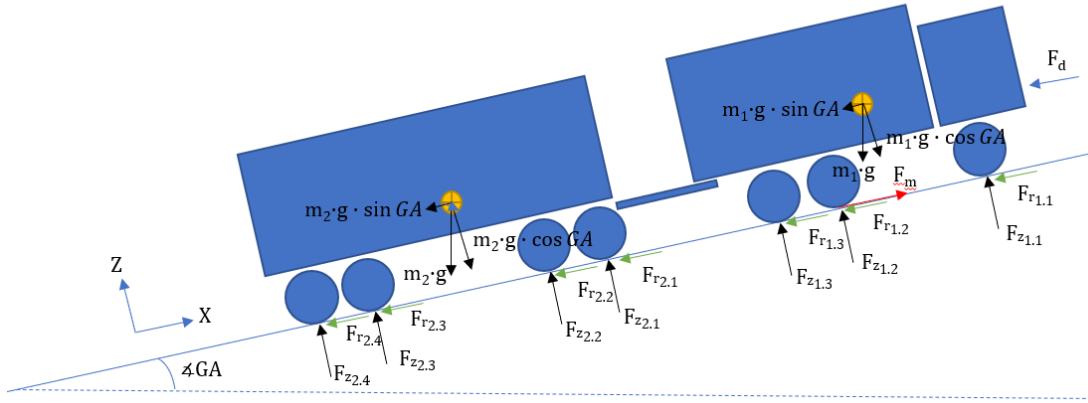


Figure 11. Free-body diagram for gradeability.

In Figure 11 is presented free-body diagram for Nordic combination for gradeability calculations at constant longitudinal velocity. The equilibrium for the X-axis will be:

$$F_m - F_d - \sum F_r - (m_1 + m_2) \cdot g \cdot \sin(GA) = 0, \quad (78)$$

where $\sum F_r = f_r \cdot (m_1 + m_2) \cdot g \cdot \cos(GA)$ and f_r is rolling resistance coefficient.

Solved gradeability from (78) will be:

$$\Delta GA = \frac{F_m - F_d}{m \cdot g} - f_r. \quad (79)$$

Parameters for gradeability calculations are presented in Table 9 (the Reference tool parameters).

Table 9. Fixed vehicle and manoeuvre parameters and values for gradeability calculations.

Symbol	Definition	Value
C_d	Air resistance coefficient	0.8
A	Vehicle's frontal area	10 [m ²]
ρ	Air density	1.3 [kg/m ³]
η	Total powertrain efficiency	0.85
f_r	Rolling resistance coefficient	0.01
v_x	Longitudinal velocity	70 km/h = 19.444 m/s

5.3 Tracking-Ability on Straight Path (TASP)

Tracking-ability on straight path (TASP) is the swept width between the first and last axle centres of the vehicle when driving at a certain speed and a certain road friction and a certain lateral road inclination.

In OpenPBS, TASP is defined solving first the lateral velocity of combination's last axle as follows (explained in Figure 12):

$$v_y = v_x \cdot \cos(p_z) - v_{ys} + L_{CoG} \cdot w_z \cdot \cos(p_z) \quad (80)$$

where r_y is axle lateral position [m], v_x is longitudinal velocity [m/s], p_z is yaw angle [rad], v_{ys} is lateral velocity at vehicle CoG [m/s], L_{CoG} is the axle's distance to CoG [m] and w_z is yaw rate [rad/s].

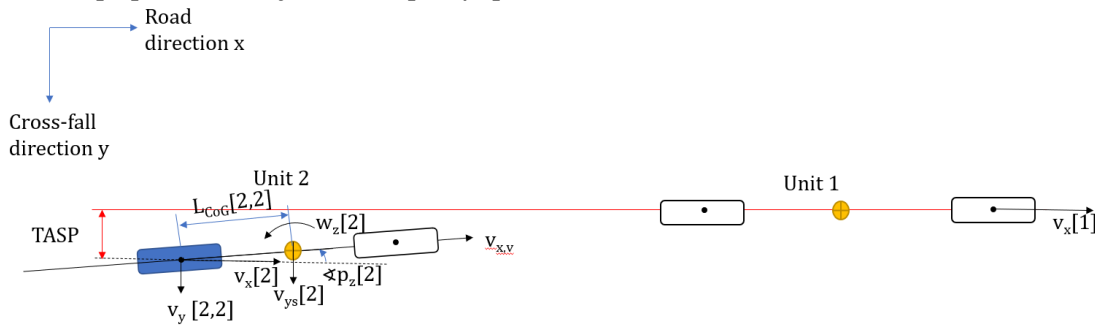


Figure 12. Free-body diagram of TASP calculations in OpenPBS.

Then the TASP value will be the time integral for solved lateral velocity of the rear axle:

$$TASP = \int v_y dt. \quad (81)$$

TASP is calculated in the Reference tool as following:

$$\Delta y = axfs \cdot \frac{1}{KC} \cdot 0.05 \quad (82)$$

where axfs is distance between the first and last axle of combination [m], KC is tyre cornering coefficient on each axle and 0.05 [rad] is the lateral road inclination.

5.4 Rearward Amplification (RWA)

In a sudden lateral movement such as a lane change, each unit of HCT vehicle is affected by a different lateral acceleration. This amplifies towards the rearmost unit of the vehicle combination and can make it difficult to handle. A wide lateral response of the vehicle may increase the risk of hitting other objects and even cause the rear unit's rollover. Lower RWA value indicates better performance of the vehicle. Rearward amplification is the amplification of movement of the last unit of combination compared with the first unit of vehicle. (Kati, 2013, ISO 14791: 2000 (E)) In both tools, RWA is defined as a ratio of maximum values of yaw rate between the first and last unit in a single lane change manoeuvre as follows:

$$RWA = \max_i \left(\frac{\max_t (|\omega_z[i](t)|)}{\max_t (|\omega_z[1](t)|)} \right) \quad (83)$$

Single lane change manoeuvre is defined in both tools with a path-following lateral acceleration input. Longitudinal velocity is 80 km/h. Lateral acceleration is defined for the front axle of vehicle with the value of 1.5 m/s². The path in single lane change will be thus:

$$\begin{aligned} y(t) &= 0, \quad t < st \\ y(t) &= \iint a_y \cdot \sin(2 \cdot \pi \cdot f \cdot t) dt, \quad st \leq t < st + \frac{1}{f} \\ y(t) &= \frac{a_y}{2 \cdot \pi \cdot f}, \quad t \geq st + \frac{1}{f} \end{aligned} \quad (84)$$

where st is start time of lane change, a_y is the lateral acceleration input and f is the lane change frequency.

There is a "PID" driver model in the Reference tool for following the path defined in (84). From the Figure 13 one can see that the manoeuvres are almost the same although there is no driver model in OpenPBS. Closed-loop driver model is not

needed in OpenPBS, because an "ideal driver model" is assumed by prescribing the path of the first axle. This is possible since a Modelica tool can compute the inverse dynamics.

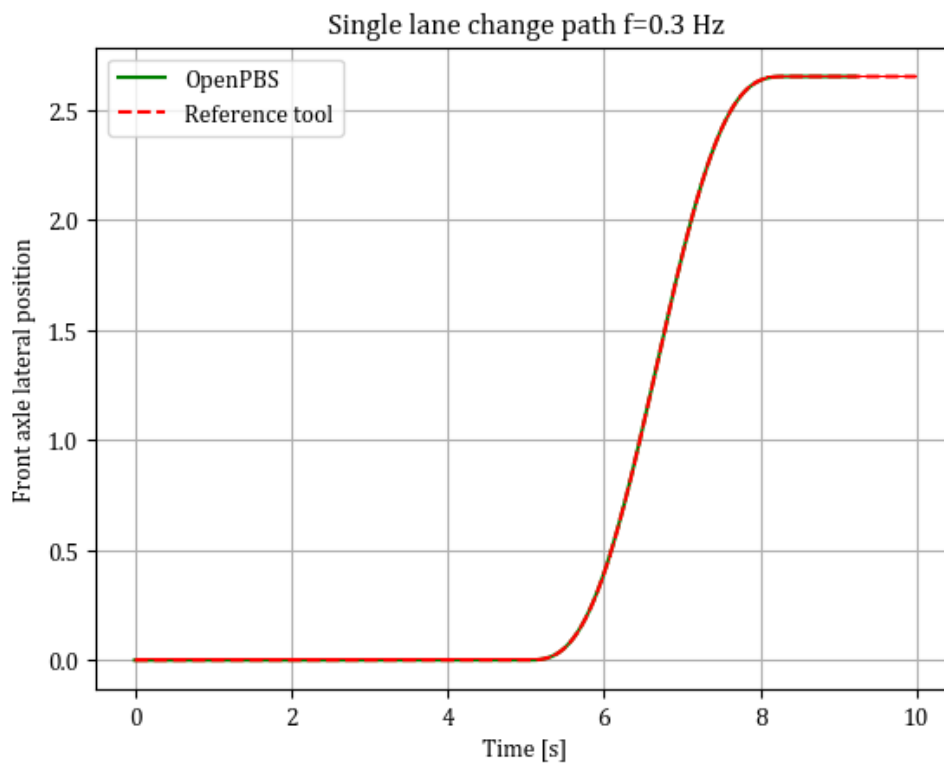


Figure 13. Front axle lateral position in single lane change with the frequency of 0.3 Hz.

The Reference tool uses a frequency sweep in a lane change frequency to find the maximum value of rearward amplification. Frequency varies in 0.05 Hz increments and the rearward amplification will be the highest value. In the current version of the OpenPBS tool the RWA is calculated for a fixed frequency. From the RWA results of the Reference tool was noticed that usually frequency of 0.3 Hz gave maximum RWA for B-doubles, and frequency of 0.4 Hz for Nordic combinations. These values are used in comparison runs (Chapter 6).

5.5 Yaw Damping (YD)

Yaw damping indicates how quickly yaw oscillations in articulation points settle after a certain manoeuvre. Taking a longer time to decay those oscillations may occur in a higher safety risk on the roads due to the driver's increased workload. In extreme conditions, longer time to decay can cause rollover of the vehicle or a

crash with a vehicle in opposite lane. Requiring an acceptable decay of oscillations of articulation points aims to improve road safety. (Kati 2013, pp. 11)

In both tools, yaw damping is defined in a single sine-steering manoeuvre. Vehicle longitudinal velocity is 80 km/h and the manoeuvre is defined as a period sine of steering angle, with the amplitude of 0.04 rad and the frequency will be the same, which gave the maximum rearward amplification.

Yaw damping model in OpenPBS is modified to use same equations as in the Reference tool. The following model is the same as presented in ISO standard (ISO 14791: 2000 (E), pp. 13-14). Yaw damping in the Reference tool is defined from articulation angles from the articulation point which gives the lowest value. OpenPBS calculates YD only for the rear coupling. From the absolute values of positive and negative amplitudes of articulation angles, mean value r is defined as:

$$r = \frac{1}{n-2} \cdot \frac{A_1+A_2}{A_2+A_3} + \frac{A_2+A_3}{A_3+A_4} + \frac{A_3+A_4}{A_4+A_5} + \dots + \frac{A_{n-2}+A_{n-1}}{A_{n-1}+A_n} \quad (85)$$

The value of r is calculated for as many amplitudes as $A_{n-1}+A_n$ is at least 10 % of A_1+A_2 or at least 7 amplitudes, if the 10% is not reached before. Then yaw damping for the articulation joint which gives the lowest ratio is:

$$YD = \frac{\ln(r)}{\sqrt{\pi^2 + \ln(r)}} \quad (86)$$

6 Comparison results

Three comparison runs for chosen 100 vehicles were performed during the thesis. The purpose of three runs was to improve the comparison code and simulation models for each comparison. The final comparison results are presented in this chapter.

6.1 Gradeability

OpenPBS and the Reference tool use the same equations to solve gradeability. Thus, the results of Gradeability are the same if the same vehicle and manoeuvre parameters are used (see Table 9). Before that, some errors from OpenPBS Longitudinal vehicle model were fixed. Gradeability results are presented in Figure 14. The figure is marked with a lower limit of 1 % that the vehicle should exceed. This minimum value of GA is used in the Reference tool. It should be noted that the Longitudinal model of OpenPBS defines two other PBS measures on the same simulation, Startability and Acceleration Capability. Thereby, OpenPBS is more comprehensive tool for evaluating the longitudinal dynamics.

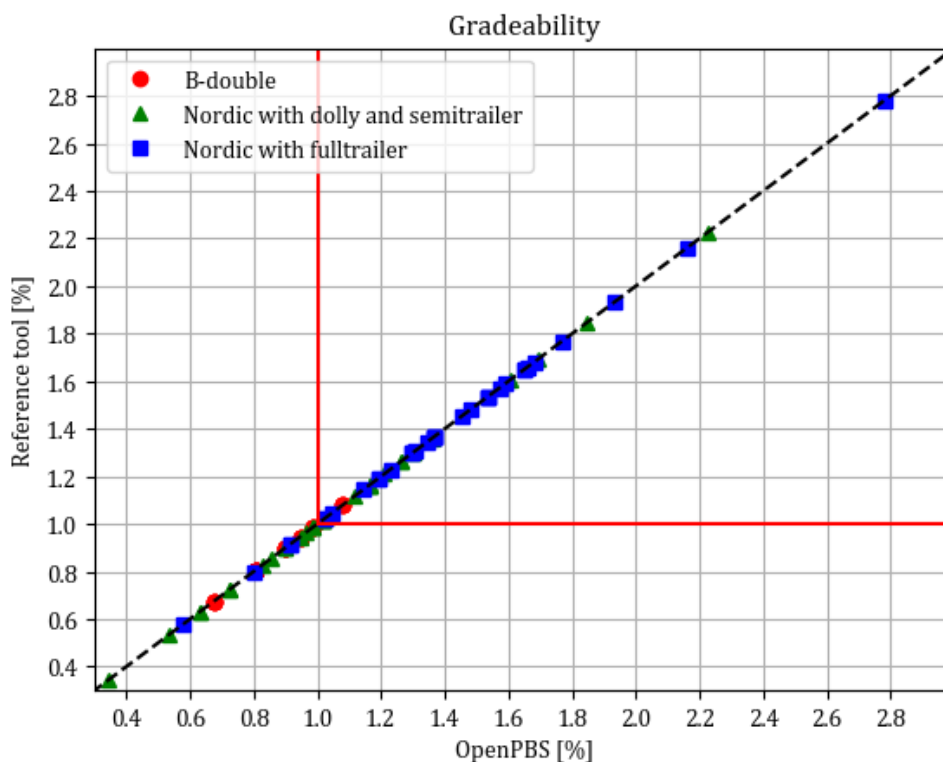


Figure 14. Gradeability results.

6.2 Tracking Ability on Straight Path

Tracking ability on straight path results were surprisingly similar despite the very simple model in the Reference tool versus the simulation model in OpenPBS. The comparison results of TASP are shown in the Figure 15. The Reference tool estimates all the TASP values to be a little higher than OpenPBS. Especially the TASP values of Nordic combinations with a fulltrailer are a bit higher in the Reference tool. This could be since the Reference tool does not consider double tyres unlike OpenPBS. The maximum value of TASP is set at 0.4 metres in the Reference tool, which is much higher than the calculated TASP for all the considered vehicles. Hence, TASP is not an important PBS measure for the compared vehicles.

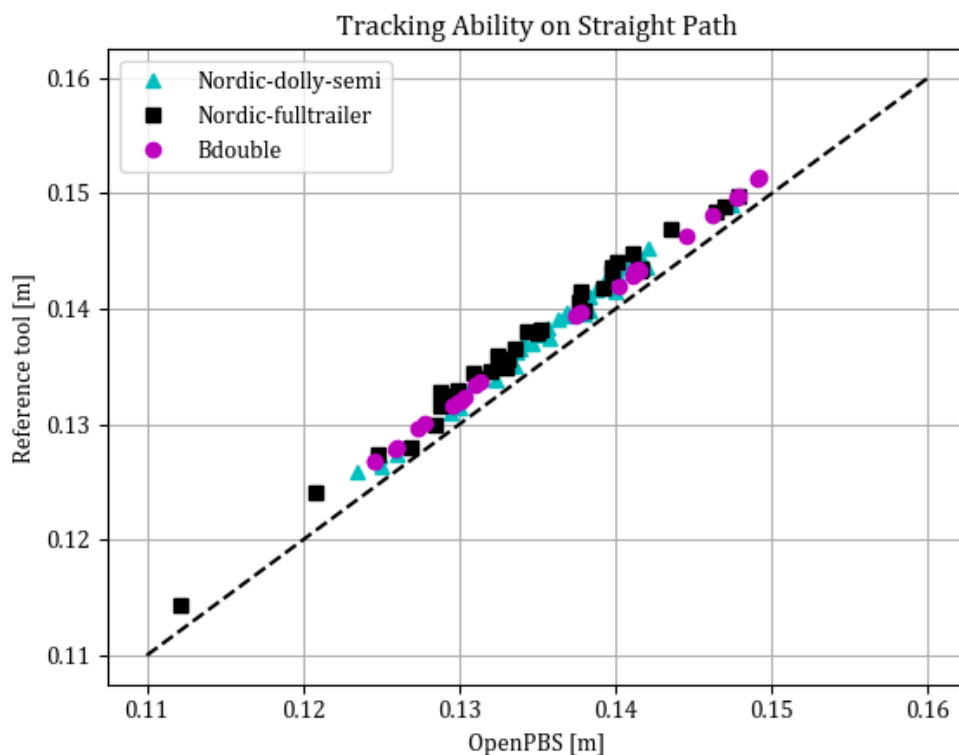


Figure 15. Comparison results for TASP.

The difference between the results is accentuated with the various axle loads as shown in Figure 16. The TASP model in the Reference tool does not consider the axle loads.

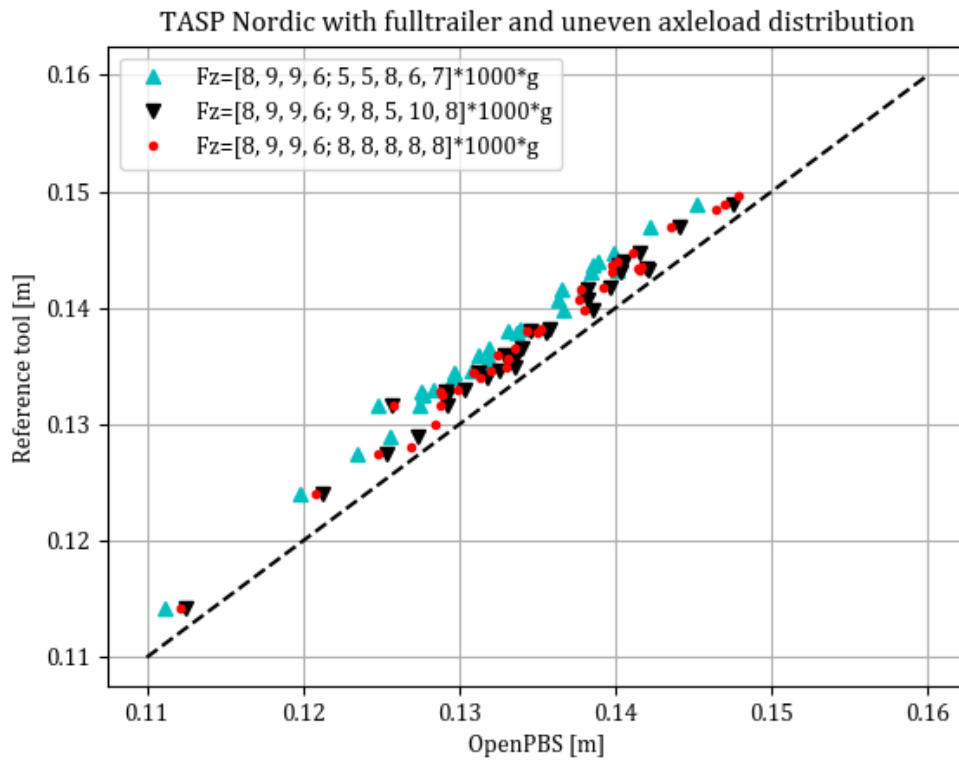


Figure 16. TASP results for Nordic combination and with uneven axleload distribution.

Despite the simple TASP model of the Reference tool, the results are rather accurate. However, a little overestimation in the Reference tool might be highlighted for the longer vehicle combinations. Thus, using the more accurate TASP model of OpenPBS would be appropriate.

6.3 Rearward Amplification

Rearward amplification values were always higher in OpenPBS as seen in Figure 17. None of the compared vehicles exceeded the maximum RWA value of 2.4, which is used as a limit in the Reference tool. On the other hand, typical acceptable maximum RWA value is 2.0 (ITF, 2019). If this value is used as a limit, it should be noted that the Reference tool always gave smaller RWA than 2.0. Respectively, OpenPBS gave the higher than 2.0 RWA result for 24 vehicles.

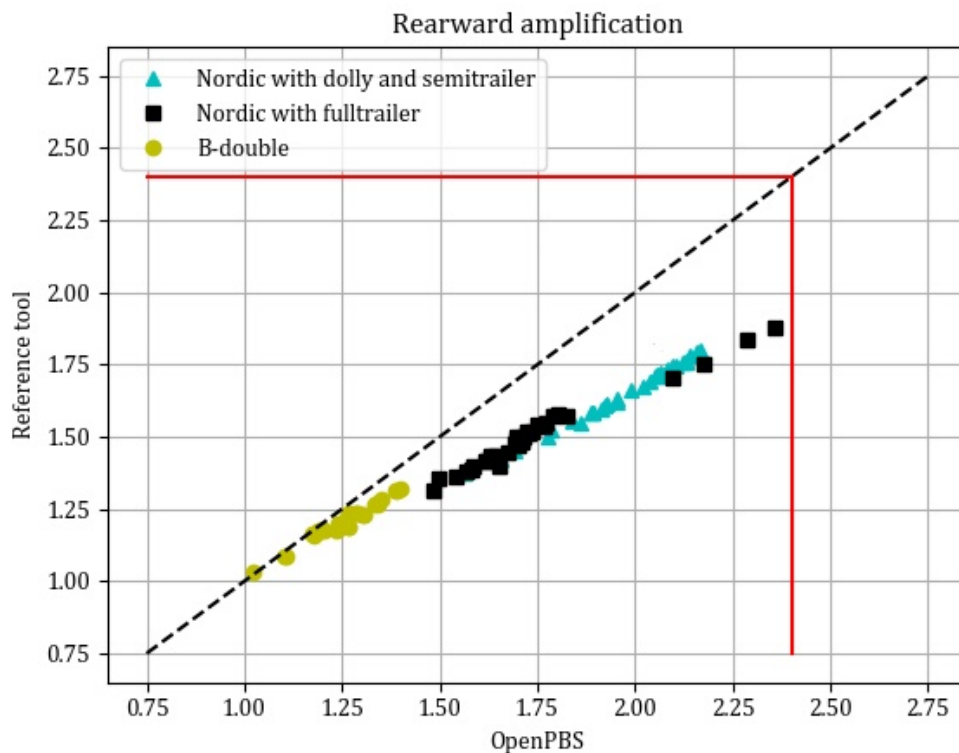


Figure 17. Rearward Amplification results using same tyre models and tyre relaxation in OpenPBS.

In Figure 18 the RWA results without the tyre relaxation in OpenPBS are presented. RWA results are still higher in OpenPBS, but the values are closer to each other between the tools. Tyre relaxation is not considered in the Reference tool. The significant influence of tyre relaxation can be noticed comparing Figure 17 with Figure 18.

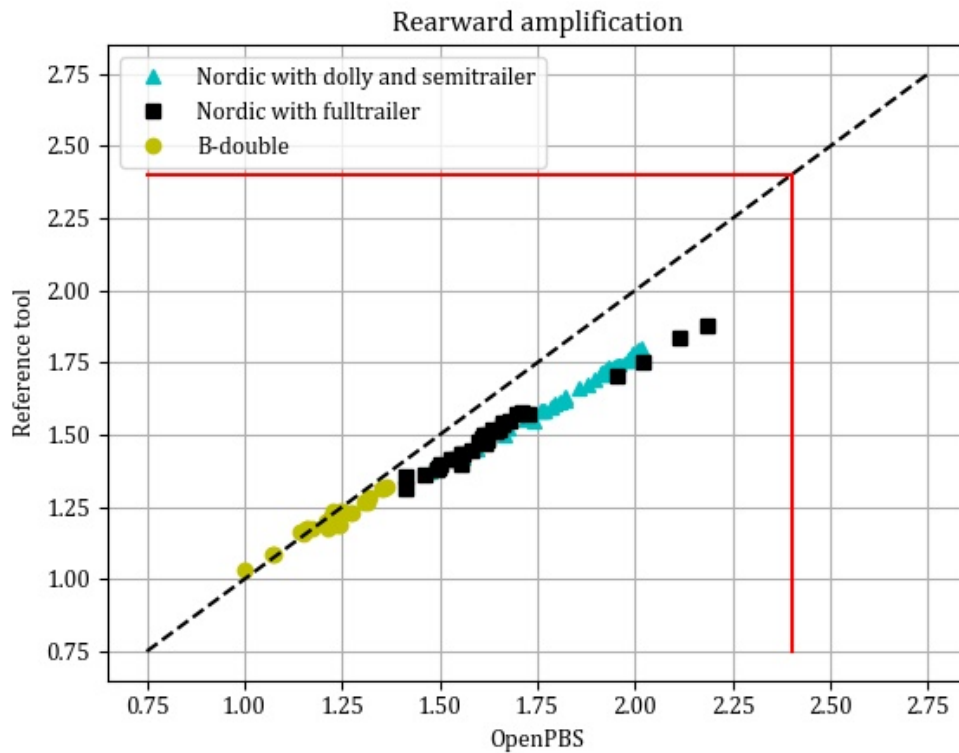


Figure 18. RWA results with the same tyre models and without tyre relaxation in OpenPBS.

Another reason for the higher RWA values of OpenPBS could be the considered roll dynamics. The influence of CoG height is better captured in OpenPBS due to included roll dynamics in simulation models. Roll dynamics is not considered in the Reference tool.

6.4 Yaw Damping

The comparison results of Yaw damping are shown in Figure 19. The Reference tool seems to give higher value of YD almost in all cases. As in Chapter 6.3, the results without tyre relaxation in OpenPBS are presented in Figure 20. Without tyre relaxation YD values are closer to each other between the tools, but still higher in the Reference tool. This could be due to included roll dynamics in OpenPBS.

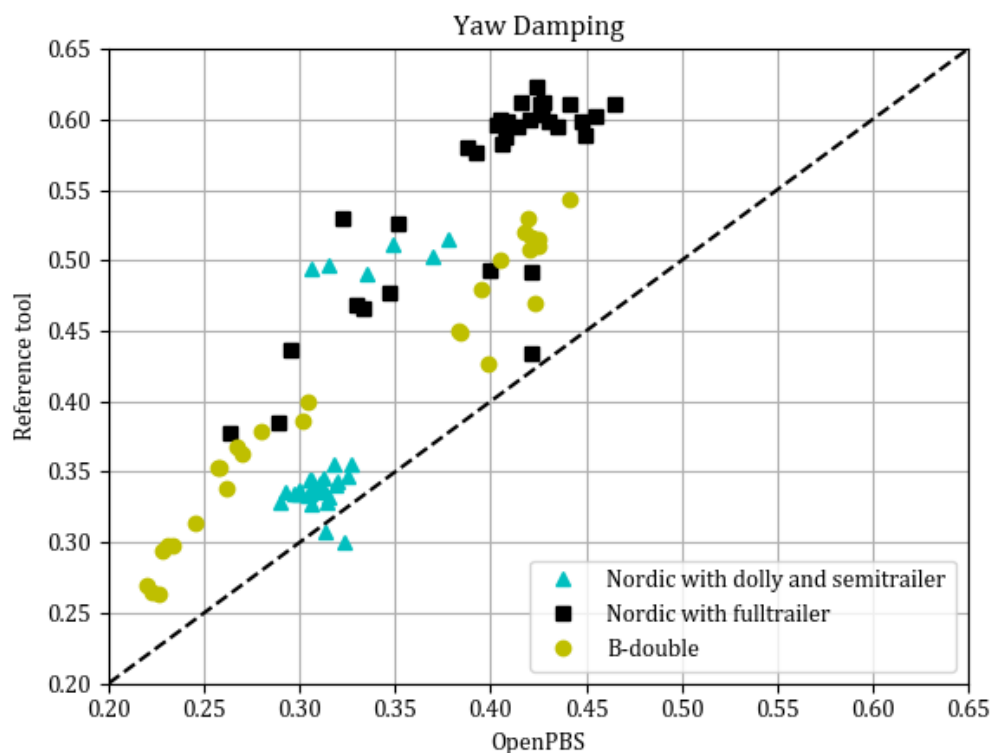


Figure 19. Yaw Damping results with same tyre models and with tyre relaxation in OpenPBS.

The minimum value of YD is set to 0.15 in the Reference tool. All the chosen combinations passed this limit. Nonetheless, the overestimation of YD in the Reference tool could be more significant for the longer and heavier vehicles. Thereby, using OpenPBS for evaluating HCT's lateral stability would be justified due to better-captured vehicle dynamics and more accurate vehicle models.

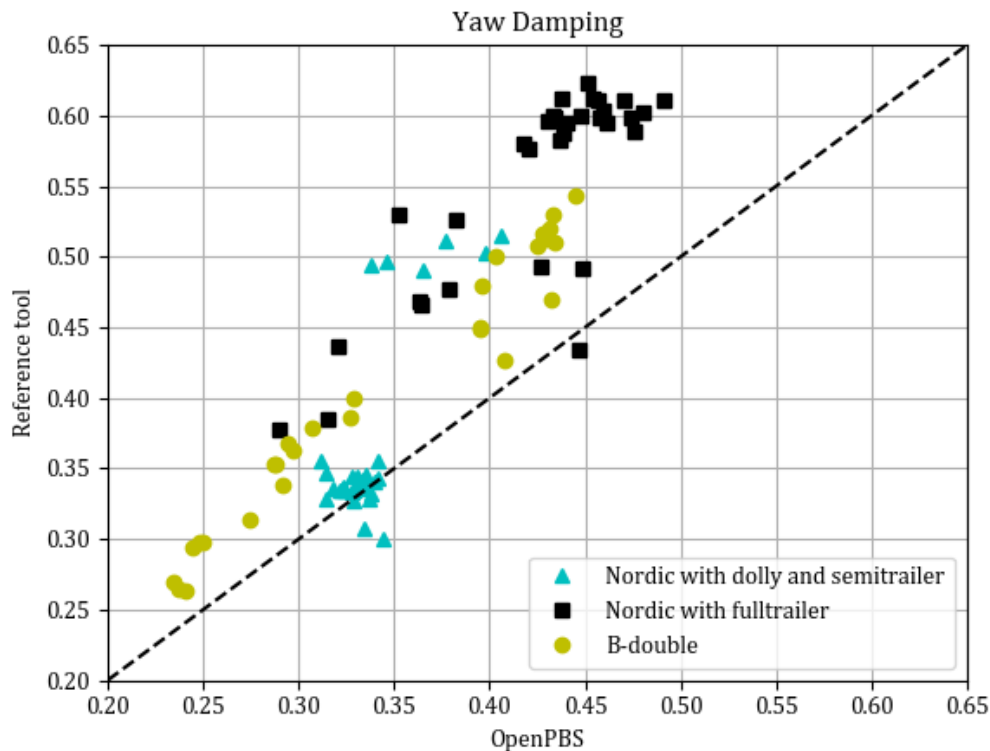


Figure 20. Yaw damping results without tyre relaxation in OpenPBS.

6.5 Steady state Rollover Threshold

None of the chosen 100 vehicles passed the limit 3.5 m/s^2 of SRT as seen in Figure 21. This could be due to chosen user defined inputs, the worst case scenarios were simulated using maximum allowed total masses and height of the vehicle.

Figure 21 shows the high influence of tyre lateral stiffness included in OpenPBS. In Nordic combinations, a trailer was the first one to roll over. All chosen semitrailers were with single tyres and the SRTs for Nordic with a converted dolly were higher in the Reference tool. Also, Nordic combinations with a fulltrailer and single tyres, got higher value in the Reference tool. Unlikely fulltrailers with double tyres got higher values of SRT in OpenPBS. This is because the tyre lateral stiffness will be twice as high for double tyres as single tyres. Values of lateral stiffness of the tyre are difficult to obtain. In the first PBS project, the values of 200 and 400 kN/m were used for validation of the proposed SRT model (Kharrazi, et al., 2017, pp. 44).

After experimental simulations during the thesis work, the value of 300 kN/m was chosen to tyre lateral stiffness (see Figure 22).

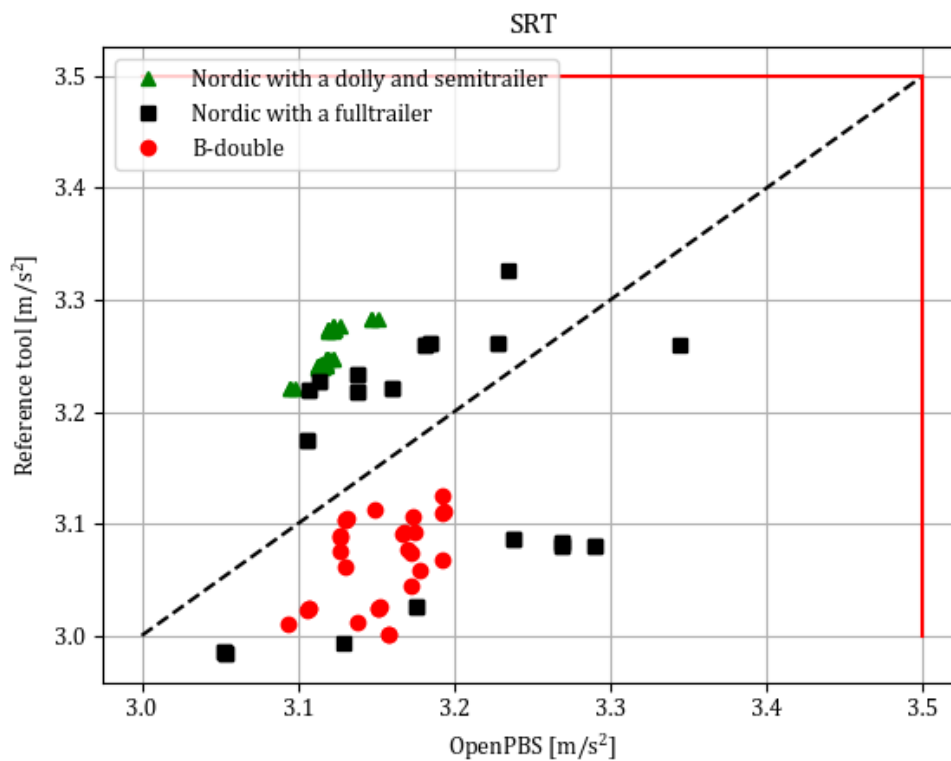


Figure 21. Comparison results of SRT.

OpenPBS gives higher SRT for B-doubles than the Reference tool even if the semitrailers are with single tyres. This could be due to the included roll stiffness of fifth wheel in OpenPBS. The first PBS project does not mention how the SRT should be calculated for B-double. The Reference tool calculates one value for the whole combination. In Figure 21, SRT was calculated for B-double considering a link-trailer and semitrailer as one unit and the tractor was replaced with the equivalent fifth wheel roll stiffness. If calculate SRT only for the semitrailer with the fifth wheel roll stiffness, SRT will be higher as shown in Figure 22. Reason for that could be that the contribution of the fifth wheel roll stiffness decreases when considering link-trailer and semitrailer as one unit.

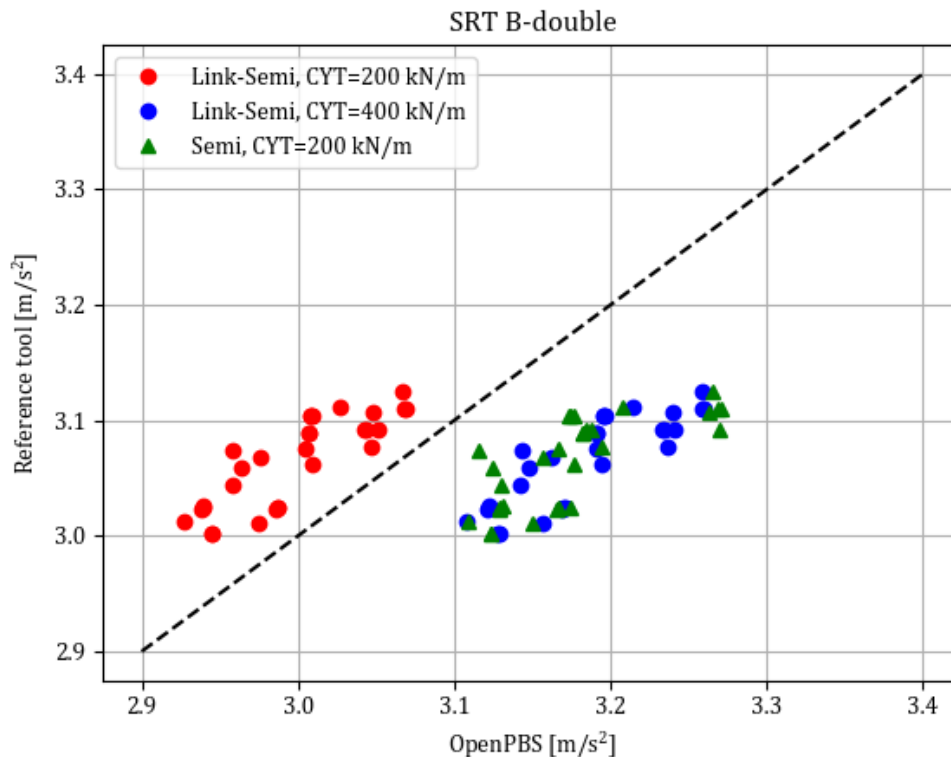


Figure 22. SRT results for B-double with different tyre lateral stiffnesses and evaluation methods. Red and blue dots consider link-trailers and semitrailers as one unit and tractors are replaced with the equivalent fifth wheel roll stiffness. Green ones are defined only for the rear semitrailer of B-double including the coupling roll stiffness.

Dispersion of SRT results in Figure 21 is significant. This is due the very simple SRT model of the Reference tool. It ignores the affect of double tyres and roll stiffness of fifth wheel. Also, the roll stiffnesses from tyres, suspension and frame are considered as a constant gradient. The implemented SRT model in OpenPBS is more accurate, it specifies the roll stiffnesses of coupling, suspension, and tyres individually. As can be seen from the results, using OpenPBS to define SRT instead of the Reference tool would be consistent due to capturing the characteristics of the vehicle.

7 Conclusions and Future work

The conclusions of the comparison between two existing PBS tools in Sweden are presented in this chapter. OpenPBS is not ready yet, so the recommendations for future work are also listed.

7.1 Conclusions

The comparison was made using the same inputs, registration numbers, vertical axle loads and height of the load. The manoeuvres and fixed simulation parameters were aligned, and the same non-linear tyre model was used. The following conclusions of the compared PBS measures were made:

- GA results were the same
- TASP results were almost the same
- RWA gets lower values in the Reference tool
- YD gets higher values in the Reference tool
- SRT model is too simplified in the Reference tool

Gradeability results were the same due to the same models for calculation. OpenPBS's Longitudinal model defines Startability and Acceleration Capability measures too. Thereby, OpenPBS is would be more attractive choice for evaluating the longitudinal performance of HCT, since it is more comprehensive.

TASP results were surprisingly close between the tools despite the very simple model in the Reference tool compared with the simulation model of OpenPBS. However, the Reference tool does not consider axle loads or number of tyres per axle in TASP calculations. The difference might be accentuated for the longer and heavier vehicles.

The lateral stability measures, RWA and YD were always assessed more optimistically in the Reference tool than in OpenPBS. As shown in the results, one reason is the involved tyre relaxation in OpenPBS. Another reason could be the better-captured influence of CoG height as an included roll dynamics in OpenPBS. Also, more accurate vehicle models and parameters are used in OpenPBS. Thus, using the more accurate and realistic tool for evaluating these critical measures would be well justified.

The high dispersion in SRT results indicates, that using proposed SRT model from the first PBS project would be consistent due to many variations in vehicle characteristics. The effect of added tyre lateral stiffness was significant in the implemented SRT model to OpenPBS. The SRT model of the Reference tool does not consider double tyres or coupling roll stiffness and the roll stiffness from tyres, suspension and frame is handled as a constant value.

As a drawback of OpenPBS could be mentioned, that Modelica is object oriented. This means that a model is often used in different manouvres, so changes can affect in more places than expected. Using Modelica also requires more special competence in dynamic modelling, since it is equation oriented (acausal modelling). Also, Modelica compiles after each model update, so during model development a Python model is typically faster.

According to this thesis, OpenPBS would be the more suitable tool for assessing HCT vehicles. OpenPBS defines more PBSs and handles more combination types than the Reference tool. OpenPBS is also more easily expanded due to its package structure and vectorized models. OpenPBS is more accurate and realistic and captures the most important phenomena of vehicle dynamics. OpenPBS has the advantage of being based on an international standard for dynamic models, Modelica, and the widely used open programming language Python.

7.2 Future work

OpenPBS is not ready yet for evaluating all the PBS measures and vehicle types. Also, some things were better made in the Reference tool. Future work regarding the development of OpenPBS are listed as follows:

- Implement frequency sweep in OpenPBS to find the maximum value of RWA as in the Reference tool. One way could be to let the Modelica tool generate system matrices (A,B,C,D) and then compute the transfer functions in a Python script
- Calculate Yaw damping for each coupling to find the minimum value
- Implement the low-speed steerable and liftable axles in Low-Speed Manoeuvres

- Model the physical difference between the fulltrailers and dolly-semitrailer in OpenPBS. A first step could be to define a boolean vectors telling whether each coupling is pitch moment free or not and whether it can take vertical force or not.
- Compare the implemented SRT model with a real physical simulation model of SRT
- Compare the simple TASP model of the Reference tool with OpenPBS model for longer and heavier combinations (consisting of 4 units), if the simple TASP model is accurate enough for LVCs
- Verify models for AB-double and B-tripple
- Decide the fixed vehicle model parameters which are not depending on the vehicle unit individual to the final version of OpenPBS
- Implement the final tyre model proposed by PBS II project.

8 References

- Aurell, J., 2017. *Beskrivning av stabilitetsanalys av tunga fordonskombinationer, rapport*. Göteborg.
- Bakker, E., Nyborg, L., Pacejka, H.B., 1987. Tyre Modelling for use in Vehicle Dynamics Studies. 1987. SAE Paper No. 870495
- Chalmers University of technology, 2020. Performance Based Standards II. [Online] Available at: <https://research.chalmers.se/project/8350> [Accessed 19 February 2020].
- ECE 111. 2001. UNIFORM PROVISIONS CONCERNING THE APPROVAL OF TANK VEHICLES OF CATEGORIES N AND O WITH REGARD TO ROLLOVER STABILITY. u.o. : UNITED NATIONS, 2001. Regulation No. 111.
- EUR-Lex, 2020. Access to European Union law. [Online] Available at: https://eur-lex.europa.eu/legal-content/EN/TXT/?uri=uriserv:OJ.L_.2018.153.01.0179.01.ENG [Accessed 26 February 2020].
- Finlex, 2020. Lagstiftning. Ursprungliga författningar. 31/2019. Statsrådets förordning om ändring av förordningen om användning av fordon på väg. [Online] Available at: <https://www.finlex.fi/fi/laki/alkup/2019/20190031> [Accessed 19 February 2020].
- ITF, 2019. High Capacity Transport: Towards efficient, Safe and Sustainable Road Freight. *International Transport Forum Policy Papers*, 69 (OECD Publishing, Paris)
- ISO 14791: 2000 (E), Road Vehicles - Heavy commercial combination vehicles and articulated buses - Lateral stability test methods. The International Organization for Standardization.
- ISO 18868:2013, *Commercial road vehicles - Coupling equipment between vehicles in multiple combination vehicles - Strength requirements*. The International Organization for Standardization
- Jacobson, B., 2016. Vehicle Dynamics Compendium for Course MMF062. Gothenburg: Chalmers University of Technology.
- Jacobson, B. et al., 2017. An Open Assessment Tool for Performance Based Standards of Long Combination Vehicles. [Online] Available at: <https://research.chalmers.se/en/publication/251269> [Accessed 14 February 2019].

Kati, Maliheh Sadeghi. 2013. Definitions of Performance Based Characteristics for Long Heavy Vehicle Combinations. Signals and Systems. u.o. : Chalmers University of Technology, 2013. ISSN 1403-266x.

Kharrazi, S., Karlsson, R., Sandin, J. & Aurell, J., 2015. Performance based standards for high capacity transports in Sweden, Linköping: Swedish National Road and Transport Research Institute (VTI).

Kharrazi, S., Bruzelius F. & Sandberg U., 2017. Performance based standards for high capacity transports in Sweden, FIFFI project 2013-03881 – Final Report” VTI report 948A.

Modelica Association, 2020. Modelica language. [Online] Available at: <https://www.modelica.org/modelicalanguage> [Accessed 31 June 2020].

OpenModelica, 2019. OpenModelica. [Online] Available at: <https://openmodelica.org/> [Accessed 10 June 2020]

Santahuhta, Ville. *Roll dynamics and tyre relaxation in heavy combination vehicle models for transient lateral manoeuvres*, Master's thesis, Chalmers and Oulu University, 2019, <https://odr.chalmers.se/handle/20.500.12380/300379>

Transport Styrelsen, 2009. TSFS 2009:59, Transportstyrelsens föreskrifter om fordonsuppgifter i vägtrafikregistret. [Online] Available at: <https://transportstyrelsen.se/sv/Regler/ts-foreskrifter-i-nummerordning/2009/> [Accessed 28 April 2020].

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