Performance based standards for high capacity transports in Sweden

FIFFI project 2013-03881 – Final report

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Abstract

Project “Performance Based Standards for High Capacity Transports in Sweden” started at the end of 2013 to investigate applicability of PBS in Sweden, and ended in Autumn 2017. The purpose of the project was to propose a performance based regulation of HCT vehicles and their access to the road network; under a PBS approach to regulation, standards would specify the performance required from the vehicle, rather than mandating prescriptive length and weight limits. In this project, all the three domains of safety, infrastructure and environment were addressed, but the focus has been on safety for which extensive testing, simulations and analysis were performed. This report gathers the outcomes of the project.

Titel: Prestandabaserade kriterier för högkapacitetstransporter i Sverige
FIFFI projekt 2013-03881 – Slutrapport

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Preface

This report includes the outcomes of the project: “Performance Based Standards for High Capacity Transports in Sweden”, supported by Vinnova with the reference number: 2013-03881. The project was coordinated by the Swedish National Road and Transport Research Institute (VTI); other parties involved in the project were Chalmers University of Technology, Volvo Group Trucks Technology, Scania, Parator Industri AB, Swedish Transport Administration (Trafikverket) and Swedish Transport Agency (Transportstyrelsen).

I would like to acknowledge certain individuals from the project partners who were of great help during the project:

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- Per Olsson (Parator).

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All the pictures in the report are provided by Volvo, if not mentioned otherwise.

Linköping, December 2017

Sogol Kharrazi
Project leader
Quality review

External peer review was performed on 5 December 2017 by Jolle Ijkema, Scania and Niklas Fröjd, Volvo. Sogol Kharrazi has made alterations to the final manuscript of the report. The research director Jonas Jansson examined and approved the report for publication on 12 December 2017. The conclusions and recommendations expressed are the authors’ and do not necessarily reflect VTI’s opinion as an authority.

Kvalitetsgranskning

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Summary


by Sogol Kharrazi (VTI), Fredrik Bruzelius (VTI) and Ulf Sandberg (VTI)

The transport sector is facing a major challenge to reduce energy consumption and limit environmental impact; therefore, there is a great interest in increasing the efficiency of the transport system in Sweden, which makes the High Capacity Transports (HCT) an attractive solution. The existing legislation in Sweden, allows heavy vehicle combinations with maximum length of 25.2 meter and maximum weight of 64 tons on the road network. However, the government is considering allowing heavier vehicles up to 74 tons on a designated part of the road network.

To introduce HCT vehicles in Sweden, the existing regulations should be modified and a proper way of regulating HCT vehicles and their access to the road network should be developed to ensure that a certified HCT vehicle would not have negative effects on traffic safety, infrastructure and the environment. One approach is to use performance based standards (PBS) for regulation of heavy vehicles access to the road network; under a PBS approach to regulation, standards would specify the performance required from vehicle, rather than mandating prescriptive length and weight limits. The inherent flexibility in the PBS approach allows industry to develop innovative vehicles optimized for different applications. PBS has been implemented in Australia, Canada, and New Zealand, and is under trial in South Africa.

In this scope, the project “Performance Based Standards for High Capacity Transports in Sweden” started at the end of 2013 to investigate the applicability of PBS in Sweden. The project objective was to propose a regulatory framework based on PBS by identifying a set of performance based standards suitable for Sweden, with attention to winter road conditions. The project ended in September 2017 and this report presents its main outcomes.

In the PBS project all the three domains of safety, infrastructure and environment were addressed; however, the focus has been on safety and manoeuvrability. All the existing regulations with respect to environmental aspects of heavy vehicles are already performance based; thus, proposing new standards for HCT vehicles was deemed to be unnecessary. However, to address the concerns about the noise emissions of HCT vehicles, a short simulation study was performed on comparison of noise emissions of a conventional and an HCT vehicle. The study showed that the resulting noise emission for a certain transport task is very similar for both vehicles. The infrastructure aspects, which has been focused on road network categorization with respect to 74 tons vehicles, was mostly performed by Trafikverket, results of which can be found in Trafikverket reports on the topic.

During the PBS project, extensive research was performed on safety aspects of HCT vehicles, resulting in a proposal of a PBS scheme, using test track experiments, offline simulations and driving simulator studies. The achieved results showed that a PBS scheme is a better means to evaluate performance of heavy vehicles than length or weight. Furthermore, the driving simulator study displayed a strong correlation between the perceived performance of the vehicles by the drivers and the studied performance measures.

One of the main objectives of the PBS project was to investigate the applicability of PBS in Sweden with attention to winter road conditions. Thus, the safety aspects which should be considered with respect to winter conditions were investigated, resulting in proposals for safe performance levels. Furthermore, the required complexity of models for accurate assessment of heavy vehicles with respect to the performance measures in the proposed PBS scheme were identified.
Sammanfattning

Prestandabaserade kriterier för högkapacitets transporter i Sverige. FIFFI projekt 2013-03881. Slutrapport
av Sogol Kharrazi (VTI), Fredrik Bruzelius (VTI) och Ulf Sandberg (VTI)


Under PBS-projektet genomfördes en omfattande studie om säkerhetsaspekter av HCT-fordon som resulterade i ett förslag till PBS-system, med hjälp av försök på provbana, simuleringar och körsimulatorstudier. Resultaten visade att ett PBS-system är ett bättre sätt för att utvärdera prestanda hos tunga fordon än fasta krav på längd eller vikt. Vidare visade körsimulatorstudien att det finns en stark korrelation mellan lastbilsförarna uppfattade prestanda av fordonen och de studerade prestandamåtten.

Ett av huvudsyftena med PBS-projektet var att undersöka användbarheten av PBS i Sverige med hänsyn till vintervägarna. Säkerhetsaspekter som bör beaktas med hänsyn till vinterförhållanden undersöktes vilket har resulterat i förslag på säkra prestandanivåer. Dessutom identifierades erforderlig modellkomplexitet för korrekt bedömning av tunga fordon med avseende på prestandamåtten i det föreslagna PBS-systemet.
1. Introduction

The large increase in the goods transport demands, the growing congestion problem and the environmental concerns over transportation emissions and fuel consumption, make High Capacity Transport (HCT) vehicles an attractive alternative to the conventional heavy vehicle combinations on the road; an alternative which is also expected to result in significant economic benefits. HCT refers to introduction of heavy vehicle combinations with higher capacity (longer and/or heavier vehicles) than the existing vehicles on the roads. With HCT vehicles, the existing capacity in the road infrastructure can be utilized efficiently without requiring too high investments, and the goods can be transported with fewer vehicles. It is expected that this will result in a reduction in the transport cost, fuel consumption, emissions and the traffic congestion.

The existing legislation in Sweden, allows heavy vehicles with maximum length of 25.25 m and maximum weight of 64 t. However, the government is considering allowing heavier vehicles up to 74 t on a designated part of the road network, which will be classified as a new category of roads with higher bearing capacity. The new road class, BK4, will be added to the existing three classes with bearing capacities BK1-3 (Trafikverket 2016).

Dispensations of longer and heavier HCT vehicles for trial periods have been granted in the recent years, which have shown considerable CO₂-reduction, fuel saving and improved transport economy (Cider & Ranäng 2013, Skogforsk 2013, Adell et al. 2014). According to Skogforsk website, as of Autumn 2017, 50 vehicles have been operating in the dispensation program, saving about 10 million litres of diesel and 25 000 tons of CO₂, since 2009.

To gain more knowledge about HCT vehicles and their effects on traffic safety, infrastructure and environment, the Swedish government is undertaking a large research program focused on HCT vehicles in Sweden. One of the projects in the HCT program is “Performance based standards for high capacity transport in Sweden” project. Performance Based Standards (PBS) is a way of regulating HCT vehicles and their access to the road network. Under a PBS approach, standards would specify the performance required from the vehicle operations rather than mandating prescriptive length and weight limits. The inherent flexibility in the PBS approach allows development of innovative vehicles optimized for different applications, without negative effects on safety, infrastructure and environment. PBS for regulation of heavy vehicles has been implemented in Australia, Canada, and New Zealand, and is under trial in South Africa. A review of these PBS approaches, and other relevant literature and regulations is published in the first report of the PBS project (Kharrazi et al. 2015).

1.1. PBS project

The project “Performance based standards for high capacity transport in Sweden” started at the end of 2013 to investigate the applicability of PBS in Sweden and to propose a regulatory framework based on PBS by identifying a set of performance based standards suitable for Sweden, with attention to winter road conditions. The project ended in September 2017; this report presents its main outcomes.

The PBS project was financed by Vinnova, reference number 2013-03881, and was part of the Swedish HCT program. The project was coordinated by VTI with partners from academia, industry and authorities. The project partners are:

1. The Swedish National Road and Transport Research Institute (VTI)
2. Chalmers University of Technology
3. Volvo Group Trucks Technology
4. Scania
5. Parator Industry AB
6. Trafikverket
7. Transportstyrelsen
In the PBS project all the three domains of safety, infrastructure and environment were addressed; but the focus has been on safety and manoeuvrability, for which extensive testing, simulations and analysis were conducted. In the following sections a brief overview of the performed tasks with respect to each domain is provided.

### 1.2. Infrastructure aspects of HCT vehicles

The main pavement deterioration mechanisms and their relationship to heavy loads, as well as bridge bearing capacity calculations were described in the first report of the PBS project (Kharrazi et al. 2015). It was also recommended that the HCT vehicles should comply with the existing load axle regulations; however, a need for investigating effects of multiple heavy load passage on the road was identified. This issue has been investigated in another project, results of which can be found in the paper by Erlingsson et al (Erlingsson et al. 2018). Erlingsson et al conducted Indirect Diametrical Tensile (IDT) fatigue tests and Confined Triaxial Tests (CTT) in a laboratory to evaluate the impact of three heavy vehicle combinations with different axle configurations. The investigated vehicles were a 64t Nordic combination allowed on the Swedish roads, a 74 t Bdouble and a 74 t Adouble. The results demonstrated that when damage due to permanent deformation is concerned, the Bdouble and Adouble vehicles cause less damage per ton of transported goods compared to the Nordic combination. Furthermore, the Adouble caused less fatigue damage per ton compared to the Nordic combination. In other words, the load distribution and axle configuration play a more important role in the resulting road damage than the total weight of the vehicle. Further studies on measuring and comparison of damage to the road caused by conventional and HCT vehicles are planned. Four instrumented test road structures have been built in northern Sweden about 100 km north of the Arctic Circle, with cold winters, mild summers and a long spring thaw period. 16% of the traffic passing the instrumented road consists of heavy vehicles, and it is on the route of 90 t HCT vehicles of ore mining industry (Erlingsson et al. 2017). Additional instrumented road structures have been built on two local roads in North of Sweden, where timber vehicles pass.

One of the main tasks with respect to infrastructure in the PBS project, was the categorization of the Swedish road network with respect to HCT vehicles; Trafikverket was the main partner responsible for this task. Trafikverket efforts have mainly been focused on road network categorization with respect to 74t vehicles, results of which can be found in Trafikverket reports (Trafikverket 2014, Trafikverket 2016). Trafikverket has proposed introducing a new class of roads with higher bearing capacity, called BK4, for 74 t vehicles. Currently, the total length of the proposed BK4 road network is just under 8000 km, which corresponds to approximately 8 percent of the total national road network. The limited size of the proposed BK4 road network is due to shortcomings in the infrastructure, but also due to excluding roads where a risk of transport transfer from rail to road has been identified. Therefore, an analysis of the risk of transfer for excluded critical road stretches is advised.

![Figure 1. Existing gross weight curves for BK1, BK2, BK3 and the proposed curve for BK4.](image-url)
One of the main components of the infrastructure that impose restrictions to HCT vehicles access to the road network, are bridges. One possible approach to assess effects of HCT vehicles on bridges is to consider more reference vehicles and update the gross weight curves accordingly. This approach has been used by Trafikverket and the resulting gross weight curve for heavy vehicles is shown in Figure 1. For more information see the Trafikverket report (Trafikverket 2014).

1.3. Environmental aspects of HCT vehicles

The existing European regulations, also in effect in Sweden, on exhaust and noise emissions and fuel consumption were reviewed in the first report of the PBS project (Kharrazi et al. 2015). All the existing regulations with respect to environmental aspects of heavy vehicles are already performance based; thus, proposing new standards for HCT vehicles was deemed to be unnecessary. However, there are concerns about the noise emissions of HCT vehicles since a long heavy vehicle combination is equipped with more tyres. Therefore, a short study was performed on simulation and comparison of noise emissions of a conventional heavy vehicle and an HCT vehicle. The simulation results showed that the noise emission is not a factor that gives either vehicle type a clear advantage over the other. Further description of the study on the noise emission of HCT vehicles and the achieved results are provided in Chapter 7 of this report.

1.4. Safety & maneouvvrability aspects of HCT vehicles

After reviewing the relevant literature and existing PBS schemes and regulations of heavy vehicles in different countries, a candidate set of performance measures for further investigation was identified in the beginning of the PBS project. The candidate measures, which are listed in Table 1, were grouped into four categories based on the practical goals they address (adapted from categorization by Fancher et al. 1989). The four categories are:

**Traction:** The heavy vehicle should be able to start motion, maintain motion and attain a desirable level of acceleration; measures that can be used to assess the vehicle performance with respect to these goals are listed in this category.

**Tracking:** The rear end of the vehicle and all the units within the vehicle combination should follow the path of the front end of the vehicle with adequate fidelity; measures that can be used to assess the vehicle performance with respect to this goal are listed in this category.

**Stability:** The vehicle should be stable, attain directional control and remain upright during manoeuvring; measures that can be used to assess the vehicle performance with respect to these goals are listed in this category.

**Braking:** The vehicle should safely attain a desirable level of deceleration during braking; measures that can be used to assess the vehicle performance with respect to this goal are listed in this category.

The selected measures for further investigation were mainly focused on traction, tracking and stability; since it was reckoned that the existing measures with respect to braking performance in ECE R13 regulations are suitable for the HCT vehicles as well (UNECE 2008). Thus, only braking stability in a turn, which does not exist in the ECE R13, was included in the further investigations. For more information about the candidate measures, see (Kharrazi et al. 2015).
<table>
<thead>
<tr>
<th>Performance measure</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Traction</strong></td>
<td></td>
</tr>
<tr>
<td>Startability (SA)</td>
<td>The maximum grade of the road on which the vehicle can commence from a standing start</td>
</tr>
<tr>
<td>Gradeability (GA)</td>
<td>The maximum grade of the road on which the vehicle can maintain an acceptable speed</td>
</tr>
<tr>
<td>Acceleration capability (AC)</td>
<td>The clearance time of the vehicle accelerating from standing still</td>
</tr>
<tr>
<td><strong>Low speed tracking</strong></td>
<td></td>
</tr>
<tr>
<td>Frontal swing (FS)</td>
<td>Swing out of the vehicle's front corner in a low speed tight turn</td>
</tr>
<tr>
<td>Tail swing (TS)</td>
<td>Swing out of the vehicle's rear corner in a low speed tight turn</td>
</tr>
<tr>
<td>Low-speed swept path (LSSP)</td>
<td>Total road width utilized by the vehicle in a low speed tight turn</td>
</tr>
<tr>
<td><strong>Tracking ability on a straight path (TASP)</strong></td>
<td>Total road width utilized by the vehicle responding to the road unevenness, on a straight path at high speed</td>
</tr>
<tr>
<td><strong>High-speed steady-state offtracking (HSSO)</strong></td>
<td>The maximum distance between path of the rearmost axle of the vehicle and the prescribed path in a steady turn at high speed</td>
</tr>
<tr>
<td><strong>High-speed transient offtracking (HSTO)</strong></td>
<td>The maximum distance between path of the rearmost axle of the vehicle and the prescribed path in a lane change at high speed</td>
</tr>
<tr>
<td><strong>Friction demand of steer tyres (FDST)</strong></td>
<td>Demanded friction at the steer tyres for maintaining the desired path in a low speed tight turn, indicating the proximity of loss of steerability</td>
</tr>
<tr>
<td><strong>Friction demand of drive tyres (FDDT)</strong></td>
<td>Demanded friction at the drive tyres for maintaining the desired path in a low speed tight turn, indicating the proximity of a jackknife</td>
</tr>
<tr>
<td><strong>Steady-state rollover threshold (SRT)</strong></td>
<td>The maximum steady lateral acceleration the vehicle can withstand before rolling over</td>
</tr>
<tr>
<td><strong>Load transfer ratio (LTR)</strong></td>
<td>The fractional change in the load carried on the left and right tyres in a lane change at high speed, indicating the proximity of a lift off</td>
</tr>
<tr>
<td><strong>Rearward amplification (RA)</strong></td>
<td>Amplification of motion (lateral acceleration or yaw rate) in rear units of the vehicle in a lane change at high speed</td>
</tr>
<tr>
<td><strong>Yaw damping ratio (YD)</strong></td>
<td>The rate at which the yaw oscillations of the vehicle settle after a pulse steer input at high speed</td>
</tr>
<tr>
<td><strong>Braking</strong></td>
<td>Directional stability and controllability of the vehicle under heavy braking in a turn</td>
</tr>
</tbody>
</table>
1.5. Representative fleet

To study the candidate performance measures and their relevancy for assessing heavy vehicles performance, a representative fleet of heavy vehicles, including both prospective HCT vehicles and existing conventional heavy vehicles on Swedish roads, was selected; see Table 2.

Table 2. Representative fleet

<table>
<thead>
<tr>
<th>No</th>
<th>Axle configuration*</th>
<th>Dimension</th>
<th>Figure</th>
<th>Vehicle Combination</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>TR4x2-ST3</td>
<td>16.5 m/40 t</td>
<td><img src="image1.png" alt="Image" /></td>
<td>Tractor-semitrailer</td>
</tr>
<tr>
<td>2</td>
<td>TR6x2-ST3</td>
<td>17.1 m/50 t</td>
<td><img src="image2.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>TR4x2-LT2-ST3</td>
<td>24.8 m/60 t</td>
<td><img src="image3.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>TR6x4-LT2-ST3</td>
<td>25.4 m/70 t</td>
<td><img src="image4.png" alt="Image" /></td>
<td>Tractor-link trailer-semitrailer, Bdouble</td>
</tr>
<tr>
<td>5</td>
<td>TR6x4-LT3-ST3</td>
<td>25.4 m/74 t</td>
<td><img src="image5.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>TR4x2-ST3-CT2</td>
<td>25.1 m/60 t</td>
<td><img src="image6.png" alt="Image" /></td>
<td>Tractor-semitrailer-centre axle trailer</td>
</tr>
<tr>
<td>7</td>
<td>TR6x4-ST3-CT2</td>
<td>25.7 m/70 t</td>
<td><img src="image7.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>TR6x4-ST3-DY2-ST3</td>
<td>31.0 m/89 t</td>
<td><img src="image8.png" alt="Image" /></td>
<td>Tractor-semitrailer-dolly-semitrailer, Adouble</td>
</tr>
<tr>
<td>9</td>
<td>TR6x4-LT2-LT2-ST3</td>
<td>33.7 m/90 t</td>
<td><img src="image9.png" alt="Image" /></td>
<td>Tractor-link trailer-link trailer-semitrailer, Btriple</td>
</tr>
<tr>
<td>10</td>
<td>TR6x4-LT3-LT3-ST3</td>
<td>33.7 m/98 t</td>
<td><img src="image10.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>TK6x2-CT2</td>
<td>18.8 m/40 t</td>
<td><img src="image11.png" alt="Image" /></td>
<td>Truck-centre axle trailer</td>
</tr>
<tr>
<td>12</td>
<td>TK8x4-CT3</td>
<td>19.9 m/56 t</td>
<td><img src="image12.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>TK6x2-FT2+2</td>
<td>24.1 m/60 t</td>
<td><img src="image13.png" alt="Image" /></td>
<td>Truck-full trailer</td>
</tr>
<tr>
<td>14</td>
<td>TK6x2-FT2+3</td>
<td>24.1 m/64 t</td>
<td><img src="image14.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>TK8x4-FT2+3</td>
<td>25.1 m/74 t</td>
<td><img src="image15.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>TK6x2-DY2-ST3</td>
<td>24.1 m/64 t</td>
<td><img src="image16.png" alt="Image" /></td>
<td>Truck-dolly-semitrailer</td>
</tr>
<tr>
<td>17</td>
<td>TK8x4-DY2-ST3</td>
<td>25.2 m/74 t</td>
<td><img src="image17.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>TK6x2-CT2-CT2</td>
<td>27.6 m/64 t</td>
<td><img src="image18.png" alt="Image" /></td>
<td>Truck-centre axle trailer-centre axle trailer</td>
</tr>
<tr>
<td>19</td>
<td>TK6x4-CT3-CT3</td>
<td>28.4 m/74 t</td>
<td><img src="image19.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>TK6x4-DY2-LT2-ST3</td>
<td>32.4 m/80 t</td>
<td><img src="image20.png" alt="Image" /></td>
<td></td>
</tr>
<tr>
<td>21</td>
<td>TK6x4-DY2-LT3-ST3</td>
<td>32.4 m/92 t</td>
<td><img src="image21.png" alt="Image" /></td>
<td>Truck-dolly-link trailer-semitrailer, ABdouble</td>
</tr>
<tr>
<td>22</td>
<td>TK8x4-DY2-LT3-ST3</td>
<td>33.5 m/98 t</td>
<td><img src="image22.png" alt="Image" /></td>
<td></td>
</tr>
</tbody>
</table>

* TR=Tractor, TK=Truck, ST=Semitrailer, CT=Centre axle trailer, LT=Link trailer, FT=Full trailer, DY=Dolly

The number following each unit name indicates number of axles. For a full trailer, i+j format is used, where the first digit (i) indicates the number of axles at the front of the trailer and the second digit (j) indicates number of axles at the rear. For the motor vehicles, i.e. tractor and truck, the common format of i+j is used, where i is twice the number of axles and j is twice the number of driven axles.
One of the key issues investigated in the PBS project is the performance of the selected representative fleet with respect to candidate measures, both in summer and winter. Furthermore, the correlation between the candidate performance measures and the drivers’ perception of the vehicle performance; as well as the correlation between heavy vehicles performance during winter and summer, were studied. The results of the correlation studies can be used to assign required performance levels which ensure safe performance both under summer and winter conditions. It should be noted that in another subproject of the Swedish HCT program, called “HCT tyypfordonskombinationer” with the reference number 2015-02327, several HCT vehicles with high efficiency and safe performance, suitable for different transport missions are identified (Fröjd et al. 2017). Thus, in the PBS project the focus was on investigating the candidate performance measures and analysis of the corresponding performance of various heavy vehicles, rather than identifying the potential future HCT vehicles.

All the vehicles in the representative fleet were modelled to simulate the performance of the representative fleet. Moreover, tests with a subgroup of the representative fleet were performed on the test track to further investigate the performance of the vehicles and to gather data for validation of the models. The simulations and experiments on test tracks were conducted under both winter and summer conditions.

Another subject studied in the PBS project is the required level of modelling details for assessing performance of heavy vehicles with respect to the candidate measures. This led to early steps toward development of a PBS tool for evaluation of heavy vehicles performance, which is an important part of an assessment procedure needed for implementation of a PBS-based regulatory framework.

1.6. Report layout

In the coming chapters, first a summary of the test track experiments with the achieved results are provided, which is followed by a chapter on the vehicle models and validation against test track experiments. In Chapter 4, the results of the comparison study on performance of HCT vehicles in winter versus summer are provided, and it is followed by the outcomes of the assessment procedures and required model complexity investigation in Chapter 5. The driving simulator study, which was performed to examine the correlation between drivers’ perceived performance of HCT vehicles with objective measures in a PBS scheme, is discussed in Chapter 6. The short study on noise emissions of the HCT and conventional vehicles and its result are described in Chapter 7. The last chapter discuss the projects outcomes and concludes the report.
2. Test track experiments

During the project, a series of tests were performed by Volvo and Scania. The aim with these tests were two-folded: to investigate the performance of HCT vehicles, as well as to collect data for virtual vehicle model validation. In this chapter a summary of test track results is provided.

2.1. High speed lateral stability

The performed tests were conducted both in summer and winter. Different HCT vehicles were tested, as well as a few conventional heavy vehicle combinations as reference. Most of the performed tests were focused on high speed lateral stability of HCT vehicles, where rearward amplification (RA) and high speed transient offtracking (HSTO) in single lane change manoeuvres were measured/calculated. The obtained results from these tests are summarized in Table 3. It can be seen that the rearward amplification of yaw rate is higher for HCT vehicles in comparison with conventional heavy vehicle combinations. However, the rearward amplification is below 2.0 for all the tested vehicles. The offtracking is equal or below 0.7 m for all tested vehicles, except the last test with a 76 t Nordic combination (truck-dolly-semi-trailer) which was performed on very slippery conditions (wet ice); it resulted in an offtracking of about 2 m. Another observation is that even for vehicle combinations of the same type, but with different dimensions and under different conditions, the performance can be quite different. Both Volvo and Scania have performed a single lane change manoeuvre in winter at speed of 65 km/h with a 74 t AB double combination (tests 6 & 10 in Table 3). However, the rearward amplification of yaw rate is much lower in the Volvo test, RA of 1.3 in comparison with 1.8 for the Scania test. This is due to different snow and friction conditions on the test track, different tyres and lane change style; also, due to the longer wheelbases of the truck and the link-trailer in the Volvo combination.

Table 3. Summary of single lane change tests in summer and winter

<table>
<thead>
<tr>
<th>No</th>
<th>Year &amp; OEM</th>
<th>Vehicle Combination</th>
<th>Manoeuvre</th>
<th>RA of yaw rate</th>
<th>RA of lat. acc.</th>
<th>Offtracking</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2014 Scania</td>
<td>TR6x4-ST2-DY2-ST2-74t, 25.7 m</td>
<td>SLC, 80 km/h</td>
<td>1.8</td>
<td>-</td>
<td>0.7 m</td>
</tr>
<tr>
<td>2</td>
<td>2014 Scania</td>
<td>TR6x4-LT3-ST3-74t, 25.2 m</td>
<td>SLC, 80 km/h</td>
<td>1.3</td>
<td>-</td>
<td>0.5 m</td>
</tr>
<tr>
<td>3</td>
<td>2014 Scania</td>
<td>TR6x4-LT3-LT3-ST3-90t, 30.5 m</td>
<td>SLC, 70 km/h</td>
<td>1.8</td>
<td>-</td>
<td>0.7 m</td>
</tr>
<tr>
<td>4</td>
<td>2014 Volvo</td>
<td>TR6x4-ST3-DY2-ST3-80t, 32 m</td>
<td>SLC, 80 km/h</td>
<td>1.6</td>
<td>1.8</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>2015 Volvo</td>
<td>TR6x4-LT3-ST3-74t, 23.4 m</td>
<td>SLC, 80 km/h</td>
<td>1.8</td>
<td>1.9</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>2015 Scania</td>
<td>TK6x4-DY2-LT2-ST3-74t, 31.4 m</td>
<td>SLC, 65 km/h</td>
<td>1.8</td>
<td>-</td>
<td>0.7 m</td>
</tr>
<tr>
<td>7</td>
<td>2015 Scania</td>
<td>TK6x4-DY2-ST3-57t, 24.3 m</td>
<td>SLC, 70 km/h</td>
<td>1</td>
<td>-</td>
<td>0.3 m</td>
</tr>
<tr>
<td>8</td>
<td>2015 Scania</td>
<td>TR6x4-LT2-ST3-55t, 24.4 m</td>
<td>SLC, 70 km/h</td>
<td>0.7</td>
<td>-</td>
<td>0.1 m</td>
</tr>
<tr>
<td>9</td>
<td>2015 Volvo</td>
<td>TR6x4-LT2-ST3-56t, 24 m</td>
<td>SLC, 70 km/h</td>
<td>0.9</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>10</td>
<td>2015 Volvo</td>
<td>TK6x4-DY2-LT2-ST3-74t, 31.3 m</td>
<td>SLC, 65 km/h</td>
<td>1.3</td>
<td>1.1</td>
<td>0.6 m</td>
</tr>
<tr>
<td>11</td>
<td>2016 Volvo</td>
<td>TK6x4-DY2-ST3-74t, 23.5 m</td>
<td>SLC, 65 km/h</td>
<td>-</td>
<td>-</td>
<td>~2 m</td>
</tr>
</tbody>
</table>
2.2. Roll stability

In 2014, Volvo performed tilt table tests on timber trailers, two of which can be used in 74 t combinations and weighed 42 t each, and the third one weighed 36 t and can be used in a 60 t combination. The purpose of the tests was to check and compare the steady state-rollover threshold (SRT) of timber trailers in the existing combinations on the road network and 74t vehicle combinations. The results are presented in Table 4 (Volvo 2014). As expected the heavier trailers have a lower SRT level, due to the higher centre of gravity (about 10% higher). The difference in SRT levels seem to be correlated to the difference in centre of gravity height, and the lighter trailer has about 10% higher SRT. The 5-axle trailers have the same SRT values, although the load height is higher in the full trailer; this can be due to the higher rigidity of the turn table in the full-trailer, compared with the fifth-wheel in dolly-semitrailer. In general, from roll stability perspective, it is safer to increase the length along with the weight for 74 t vehicles; so that centre of gravity is not too high.

In 2015, Volvo tested the SRT of a high loaded 74 t Bdouble combination on a test track using rollover preventing outrigger wheels. The Bdouble combination had a 2.5-26 m centre of gravity height at the trailers, and without ESP it rolled over at 3.1-3.2 m/s², which is similar to the tilt table results. The combination did not roll over with ESP on, since the ESP function prevented reaching high levels of lateral acceleration (Volvo 2016a).

It should be noted that the roll stability of the combination was also tested in a dynamic lane change manoeuvre, in which the combination did not roll over. The lane change manoeuvres were performed with a 1.75 m/s² lateral acceleration at the front axle at different frequencies. Considering the lateral acceleration rearward amplification of 1.9 (see Table 3), the manoeuvre resulted in 3.3 m/s² in the rear unit in the worst frequency, without rollover. In a dynamic manoeuvre the steady-state value of rollover threshold can be exceeded to some extent, since it is transient.

Table 4. Results of tilt table tests of timber trailers

<table>
<thead>
<tr>
<th></th>
<th>4-axle full trailer, 36 t</th>
<th>5-axle full trailer, 42 t</th>
<th>5-axle dolly-semitrailer, 42 t</th>
</tr>
</thead>
<tbody>
<tr>
<td>SRT (m/s²)</td>
<td>3.61</td>
<td>3.28</td>
<td>3.28</td>
</tr>
</tbody>
</table>
Volvo also performed an additional tilt table test in 2016 with a centre axle trailer. The purpose of this test was to investigate the effect of single versus double mounted tyres on SRT of the trailer. As expected the trailer with single mounted tyres, with its larger track width, had a higher SRT. Moreover, lower pressure in double mounted tyres resulted in even lower value of SRT, see Table 5.

**Table 5. SRT of a centre axle trailer with single and double mounted tyres**

<table>
<thead>
<tr>
<th></th>
<th>Single 355/50 R22.5, 9.0 bar</th>
<th>Twin 295/60 R22.5, 8.5 bar</th>
<th>Twin 295/60 R22.5, 5.5 bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>SRT (m/s²)</td>
<td>4.55</td>
<td>4.15</td>
<td>4.05</td>
</tr>
</tbody>
</table>

### 2.3. Braking stability

Volvo tested the stability of two HCT vehicles during braking at high speed in a curve in winter conditions. The first test was performed in winter 2015 on a 74 t ABdouble combination. The test results showed that the vehicle was stable during braking when the ABS function was working properly. But, malfunctioning of the ABS on any axle groups, resulted in large offtracking (Volvo 2017a). A similar test was conducted with a 76t Nordic combination (truck-dolly-semitrailer) in winter 2016. During this test, the track was very slippery (wet ice) and the maximum braking in curve was performed twice. The first test showed similar satisfactory results as in winter 2015; but, the second repetition of the test resulted in large deviation, about 10 m, between front and rear axles paths (Volvo 2016b).

Additionally, Volvo performed heavy braking in curve at 70 km/h on wet asphalt with a 74 t Adouble combination in 2015. The vehicle remained stable during braking. The stopping distance of the Adouble combination on a wet road surface at different speeds was measured as well, the results are provided in Table 6.

**Table 6. Braking distance of a 74t Adouble on wet asphalt**

<table>
<thead>
<tr>
<th>Initial speed</th>
<th>50 km/h</th>
<th>60 km/h</th>
<th>70 km/h</th>
<th>80 km/h</th>
<th>90 km/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stopping distance of an Adouble (m)</td>
<td>16</td>
<td>25</td>
<td>31</td>
<td>43</td>
<td>56</td>
</tr>
</tbody>
</table>
2.4. Traction and low speed tracking

Volvo also performed traction tests and low speed turns with the 74t Adouble combination in 2015, to investigate the vehicle accessibility and manoeuvrability. The results are summarized in Table 7. It should be noted that for the hill start test, standard constant slope was not used due to space limitations; instead, a country road with progressive slope was used, which was postprocessed into equivalent constant slope for every starting position.

Table 7. Traction and tracking of a 74t Adouble

| Achievable speed during hill climbing on a 5% slope | 45 km/h  |
| Hill start on a 12% slope                          | passed   |
| Acceleration capability on a flat road, 0-50 km/h | 30 s     |
| Acceleration capability on a flat road, 0-80 km/h | 63 s     |
| Low speed swept path in a turn with outer radius of 12.5m | 7.8 m |

Figure 4. Braking in a curve on wet asphalt with a 74t Adouble, Volvo summer test.

Figure 5. Low speed turning with an Adouble, Volvo summer test.
3. Vehicle models

The Volvo vehicle model library (VTM), with some modifications, was used to model the representative fleet, presented in Table 2 in the introduction chapter. The vehicles are modelled as multibody mechanical systems, using SimMechanics toolbox in Matlab-Simulink. The representative fleet consists of 22 vehicles, including HCT vehicles, as well as conventional heavy vehicle combinations allowed on the Swedish roads.

3.1. Tyre model

An important part of the vehicle models, is the tyre model; since the tyre characteristics influence the vehicle performance significantly. Thus, the models were validated against test data collected in both summer and winter, using two tyre models, one for summer conditions and one for winter. Due to the existing variety of tyres and the diversity of road surface condition, especially during winter, choosing one tyre for modelling is not a trivial task. The existing tyre data at VTI, measured at VTI tyre testing facility (Nordström 1993) or gathered in other projects were reviewed, which as expected showed a large diversity, see Figure 6. In this project, an average tyre was selected for each weather condition, see Figure 7. However, there is a need for further investigation of different tyre characteristics and tyre modelling and development of standard tyres for assessment of heavy vehicles performance.

![Sample tyre data for summer condition, high friction](image1)

![Sample tyre data for winter condition, low friction](image2)

Figure 6. VTI Sample tyre data at VTI for various tyres under diverse conditions. Cornering stiffness is the tyre ability to resist deformation while cornering and relates the side force to the slip angle.

![Characteristics of the selected tyres](image3)

Figure 7. Characteristics of the selected tyres for the vehicle fleet simulations in summer and winter.
3.2. Sample validation results

The simulation results for high speed lateral dynamics were compared with test data gathered by Volvo and Scania, presented in previous chapter. Test data for several vehicle combinations, both in summer and winter was used. Some tuning of the tyre cornering stiffness and friction level was allowed to match the test data to validate the models. The results show an acceptable level of accuracy of the models. Some sample results are shown in Figure 8 and Figure 9, where lateral performances of two Bdouble vehicles with different axle configuration in a lane change manoeuvre under summer and winter conditions are plotted. The error in estimated rearward amplification by simulation is bound to 10% for both vehicles. More heavy vehicles and test data were used for validation of the models, plots of which are provided in Appendix A and Appendix B. The error in estimated rearward amplification by simulation varies based on vehicle type and surface condition, but it is bound to 25% for all cases.

Figure 8. Lane change of a Bdouble (TR6x4-LT3-ST3, 25 m, 74 t) in summer, test versus simulation.

Figure 9. Lane change of a Bdouble (TR6x4-LT2-ST3, 24 m, 56 t) in winter, test versus simulation.
4. Heavy vehicles performance, winter versus summer

In this chapter the effect of road surface condition and tyre-road grip on the performance of heavy vehicle combinations is discussed, and correlation between the vehicle performance in winter and summer is explored. From a practical aspect, it is easier and less costly and time consuming to assess the performance of heavy vehicles for summer condition, and use the correlation between the vehicle performance in summer and winter to decide performance levels which ensure safe performance during winter as well, if such a correlation exist. It is also important to note that if a heavy vehicle is to be permitted on a certain road network, features of the roads also play a key role on the required level of performance from the vehicle. In this scope, the candidate performance measures listed in the introduction chapter have been studied with respect to high and low friction, results of which is summarized in this chapter.

4.1. Traction

The traction measures, namely startability, gradeability and acceleration capability were calculated for a series of heavy vehicle combinations. The calculations were performed at Volvo using the detailed powertrain model which is widely accepted and used at Volvo. Less vehicles were considered in this study compared to the reference fleet, since the effect of axle configuration of the towed units on the traction measures is insignificant; the key factor is the normal load on the driven axles. Thus, only one axle configuration is considered for the 9 studied heavy vehicle combinations, presented in Table 8.

The simulation results showed that the effect of tyre-road friction level on the gradeability is insignificant. However, the startability of the heavy vehicles deteriorate significantly with lower friction levels; on average the calculated startability on a low friction surface is only 10% of the startability on a high friction surface, see Figure 10. None of the considered heavy vehicle combinations, not even the conventional vehicles, will be able to start on a grade higher than 5% when the tyre-road friction level is low. This is far from 12% which a heavy vehicle combination should be able to start on during summer conditions. One solution to overcome the startability issue of the heavy vehicles in winter is to allow exceeding the driver axle load limit during a brief time at start up, by axle lift or other means of load transfer to drive axles.

The Volvo model does not handle the low friction for calculation of the acceleration capability; thus, a simpler model was used for this purpose. As expected, the results showed that the acceleration capability decreases in winter conditions. However, effect of low friction on acceleration capability is not as significant as the effect on startability.

Table 8. Heavy vehicle combinations simulated for the traction study

<table>
<thead>
<tr>
<th>No</th>
<th>Vehicle combination</th>
<th>Dimension</th>
<th>Load on driven axles</th>
<th>Figure</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>TR6x4-ST3</td>
<td>16.5 m/40 t</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>TR6x4-ST3-CT2</td>
<td>25.2 m/60 t</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>TR6x4-LT3-ST3</td>
<td>30.9 m/80 t</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>TR6x4-ST3-DY2-ST3</td>
<td>31.5 m/80 t</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>TR6x4-LT3-LT3-ST3</td>
<td>33.8 m/90 t</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>TK6x4-FT1+1</td>
<td>18.7 m/40 t</td>
<td>16</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>TK6x4-DY2-ST3</td>
<td>25.2 m/60 t</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>TK6x4-CT2-CT2</td>
<td>27.3 m/66 t</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>TK6x4-DY2-LT3-ST3</td>
<td>33.8 m/90 t</td>
<td>19</td>
<td></td>
</tr>
</tbody>
</table>
4.2. Low speed tracking

The effect of tyre-road friction, as well as the turn angle on the swept path in a low speed tight turn was investigated. As used in many regulations, turning in a roundabout with outer radius of 12.5 m was simulated. In the Australian PBS scheme, a 90˚ turn is used (NTC 2008), while in the European Directive 97/27/EC, it is stated that a vehicle combination should be able to move within an area described by an outer circle of 12.5 m radius and an inner radius of 5.3 m. In Swedish regulations an inner circle of 2 m is used instead, to allow for manoeuvring of 25.25 m vehicle combinations.

A study by Pecchini and Giuliani (2013) suggests that more angles should be studied. Long vehicle combinations will not reach their steady state offtracking when driving in a roundabout, which makes the angle a test parameter to investigate in a PBS framework. Here 5 exit angles ranging from 90˚ to 210˚ were used, see Figure 11. The longitudinal speed was set to 1 m/s. The reference trajectory is given as input to the driver model in the simulation. The driver model is a PID controller acting on the vehicle longitudinal and lateral path errors. All the 22 vehicles in the reference fleet were simulated in the roundabout manoeuvre, and the resulting offtracking are plotted in Figure 12. The achieved results indicate a dependency of the offtracking on the simulated turn angle, specifically for longer vehicle combinations. Note that here offtracking is plotted instead of the swept path; the track width should be added to the offtracking values to get the swept path.

![Startability (%)](image)

**Figure 10. Effect of tyre-road friction level on the startability of heavy vehicle combinations.**

![Illustration of exit angles](image)

**Figure 11. Illustration of the considered exit angles in the roundabout manoeuvre.**
All the turning manoeuvres were also simulated with a tyre-road friction of 0.25, to analyse the effect of road surface condition on the swept path. The relative difference between the low speed offtracking in winter and summer conditions is plotted in Figure 13. The relative difference is well below 5% for most of the vehicle combinations, and below 10% for the rest. Thus, it was concluded that the inclusion of low tyre-road friction in the low speed tracking measures is not necessary. Here, only the results for swept path/offtracking were presented, comparable results are expected for the other low speed tracking measures, tail swing and frontal swing.

Figure 12. Offtracking of 22 vehicle combinations in a roundabout manoeuvre on high friction surface.

Figure 13. Relative difference, in percentage, between low speed offtracking in summer and winter.
4.3. High speed tracking

The effect of wintry road conditions was investigated for only one of the high speed tracking measures, namely High Speed Transient Offtracking (HSTO); since, it was anticipated that the effect of wintry road conditions would be most significant for this measure. It should also be noted that the high speed steady-state offtracking and the tracking ability on straight path are anticipated to be highly correlated; since, the difference between the two measures is the level of lateral acceleration the vehicle combination is exposed to.

To investigate the correlation between the HSTO of vehicle combinations in winter and summer, Single Lane Change (SLC) manoeuvres were simulated for the vehicle combinations in the reference fleet, with the selected tyre models for both high and low friction. A single lane change manoeuvre can be simulated using an open loop sinusoidal steer input, or as a closed loop manoeuvre with a driver model in the loop that decides the steering input for following a defined path. As the first step, a closed loop and an open loop SLC manoeuvre on dry asphalt were simulated for a subgroup of the reference fleet, to investigate the correlation between the two methods of simulating a lane change.

For the open loop SLC, a sinusoidal steer input with frequency of 0.4 Hz at speed of 80 km/h was used; the amplitude was adjusted for each vehicle to result in a 3m lateral displacement of the front axle, so that the vehicles perform similar lane changes. For the closed loop single lane change manoeuvre, a simple PID driver model for following the desired path was used. The desired path was defined according to the ISO 14791 standard (ISO 2002), with a lateral acceleration of 2 m/s² at the front axle and a frequency of 0.33 Hz. This results in a 3m lane change which is comparable with the simulated open loop SLC. The achieved results are shown in Figure 14. In addition to the offtracking, rearward amplification of lateral acceleration and yaw rate, as well as load transfer ratio (LTR) are plotted in the figure, since all these measures can be investigated in a SLC manoeuvre. More information about these high speed stability measures are provided in section 4.5.

Figure 14 shows that the results of the open loop and closed loop manoeuvres are highly correlated for all the relevant performance measures. Care should be taken before generalizing this result, due to its dependency on the utilized driver model. Nevertheless, in the continuation of this investigation, open loop SLC manoeuvres were used to avoid dependency of the results on the driver model.

![Figure 14. Correlation between the performance of vehicles, closed loop SLC vs. open loop SLC.](image-url)
A 3m open loop single lane change manoeuvre was simulated for all the 22 vehicle combinations in the reference fleet, with three steer input frequency of 0.3, 0.35 and 0.4 Hz and with speed of 80 km/h for high friction and 70 km/h for low friction (μ=0.25). The resulting HSTO for high friction versus low friction is plotted in Figure 15. The offtracking values for high and low friction are highly correlated. The steer input frequency does not have a significant effect on the results.

In the next step, the input steering frequency was kept constant as 0.35 Hz, but different values of friction, ranging from 0.2 to 0.35, were considered to simulate different winter road conditions, see Figure 16. The offtracking is significantly larger for all the vehicles in winter compared with summer, and it increases as the friction decreases. The correlation is evident for all simulated friction levels, and it is almost linear. This winter-summer performance correlation can be used to require a certain level of performance on summer, which also ensures stable performance in winter.

It should be noted that in the performed simulation, the vehicle combinations are compared in a single lane change with 3 m lateral displacement using a sinusoidal steer input. Other approaches can be used, such as giving a sinusoidal lateral acceleration input with a fixed amplitude. Here the sinusoidal steer input is used due to its easier implementation.

Figure 15. Correlation between offtracking of the vehicle combinations in the reference fleet in winter versus summer, during a 3 m SLC manoeuvre with different steer input frequency.

Figure 16. Correlation between offtracking of the vehicle combinations in the reference fleet in winter versus summer, during a 3 m SLC manoeuvre for different friction levels.
The influential infrastructure features during a lane change manoeuvres are the lane width and the road surface friction. The nominal values of these parameters on the Swedish road network, according to road design guidelines and road maintenance requirements are as followings (Trafikverket 2012):

- The road friction should be at least 0.35 for major roads and 0.25 for minor roads.
- The lane width is 3.5-3.75 m for highways, 3.0-3.75 m for major roads and 2.75-3.25 m for minor roads.

Considering the linear correlation depicted in Figure 16, the low friction offtracking is about 1.45 times more than the high friction offtracking for a surface friction level of $\mu=0.35$. Thus, the offtracking in summer should be below 0.7m, in a single lane change at 80 km/h with lateral displacement of 3m, to ensure an offtracking below 1.0m in winter, which means the vehicle stays within the lane on major roads in winter as well, considering a lane width of 3.5m and the vehicle width of 2.5m.

The values of the simulated high speed transient offtracking for vehicle combinations in the reference fleet are plotted in Figure 17. Most of the simulated vehicles have an offtracking below 0.7 m on high friction, however many of the combinations with centre axle trailers have larger offtracking, including the conventional combination tractor-semitrailer-centre axle trailer which has an offtracking of 0.8 m. Thus, if it can be assumed that the existing fleet on Swedish roads are safe and allowing occupying up to 3.6 m lane width during a lane change, an offtracking of 0.8 m can be set as the limit.

It should be noted that the provided offtracking values are for the specific simulated configuration of the vehicle combinations; different offtracking values can be achieved by changing the vehicle configuration, such as number and position of the axles and position of the articulation joints.

![High Speed Transient Offtracking](image)

*Figure 17. Offtracking of the vehicle combinations of the reference fleet in a 3m SLC manoeuvre.*

### 4.4. Low speed stability

The friction demand of steer tyres is a measure aimed to estimate how much friction is needed to complete a tight turning manoeuvre without losing the steerability of the vehicle. The friction demand of the drive tyres estimates how much friction is needed to propel the vehicle and to prevent a jackknife of the towed units in such a manoeuvre. Simulating the same roundabout manoeuvre used for swept path, the friction demands of the steer and drive tyres for all the 22 vehicle combinations in the reference fleet were calculated. It should be noted that the purpose is to assess how much peak friction is required for the vehicle to complete the manoeuvre. Thus, the friction demand is calculated as the ratio of planar forces and normal forces, and it is not normalized by friction coefficient.
Figure 18 and Figure 19 show the calculated friction demand of the steer tyres, using tyre models for high and low tyre-road frictions respectively. Unexpectedly, the friction demand of the steered axle is lowered in the snow condition ($\mu=0.25$) compared to the dry asphalt. A similar observation can be drawn for the drive axles by observing Figure 20 and Figure 21. This phenomenon can be explained by the fact that the friction demand is not only dependent on the friction, but also on other tyre properties such as cornering stiffness. This is illustrated in Figure 22, where the friction demand on the steered axle is plotted for a parameter sweep in friction and stiffness for the Adouble combination (TR6x4_ST3_DY2_ST3). The plot illustrates that the stiffness of the tyres plays an important role in the measure with an almost linear dependency. This is aligned with the intuition that stiffer tyres will generate higher tyre forces for the same displacement and that most of the required friction forces are used to oppose other tyre forces within the combination rather than maintaining the path of the vehicle. Also, worth noticing is that the vehicle configuration has a higher impact on the friction demand than the total length of the vehicle combination. For instance, the friction demand of steer tyres is highest for the combinations with 4-axled trucks, due to resistance from the rear axle group to traveling through a tight turn. Furthermore, the results show that the friction demand on snow does not depend on the manoeuvre angle, implying that the peak force utilization occurs early in the manoeuvre on snow.

Figure 18. Friction demand of steer axles on dry asphalt

Figure 19. Friction demand of steer axles on snow
Figure 20. Friction demand of drive axles on dry asphalt.

Figure 21. Friction demand of drive axles on snow.

Figure 22. Friction demand as a function of tyre stiffness and friction for the Adouble combination.
In conclusion, the simulation results suggest that the interpretation of the friction demand measures is ambiguous and that in addition to the friction level, the tyre stiffness has also a significant impact on the value. This indicates that calculating the friction demand on high friction as a means for assessing the performance on low friction is not an appropriate approach, due to the dependency of the measure on both cornering stiffness and friction level. However, it can be argued that the measure can be calculated for winter condition directly to ensure that a heavy vehicle combination would be able to manoeuvre through a tight turn in winter conditions.

4.5. High speed stability

The load transfer ratio was investigated in the same lane change manoeuvre used for high speed transient offtracking study, described in section 4.3. As expected the rollover risk is not an issue during winter due to lower levels of lateral acceleration, and the load transfer ratio is generally lower in winter compared to summer, see Figure 23. Thus, it can be concluded that testing the vehicle roll stability during summer condition is sufficient.

![Load Transfer Ratio](image)

**Figure 23. Load transfer ratio of vehicle combinations in the reference fleet in a 3 m SLC manoeuvre.**

Rearward amplification of yaw rate was also evaluated in the same single lane change. Again, the effects of steer input frequency and tyre-road friction level on the correlation between performance in summer and winter was investigated, see Figure 24 and Figure 25.

It can be seen that the rearward amplification values for winter and summer are highly correlated. Although the values for winter and summer are not too different, the difference gets exaggerated for vehicles with poor performance which will result in swing out and instability in winter. Thus, the yaw rate rearward amplification in summer condition should be below a certain value, to ensure that the vehicle will not be prone to swing outs in winter. According to the obtain simulation results, this limit value is between 2.1 and 2.5 in a single lane change at 80 km/h with a lateral displacement of 3m. All the simulated vehicle combinations have a yaw rate rearward amplification below 2.1, except for the vehicle combination with double two-axled centre axle trailers, see Figure 26. Thus, considering the performance of the current fleet and the correlation between winter and summer performance, a limit value of 2.2 is suggested.
Figure 24. Correlation between yaw rate RA of vehicle combinations in the reference fleet in winter versus summer, during a 3 m SLC manoeuvre with different steer input frequency.

Figure 25. Correlation between yaw rate RA of vehicle combinations in the reference fleet in winter versus summer, during a 3 m SLC manoeuvre for different friction levels.

Figure 26. Yaw rate RA of vehicle combinations in the reference fleet in a 3 m SLC manoeuvre.
With respect to yaw damping ratio, the performance of the vehicle combinations during winter and summer are similar. However, the simulation results in Figure 27, indicate that vehicles with very low damping ratio in summer, below 0.1, will be prone to undamped oscillation in winter. As shown in Figure 28, all the simulated combinations have a yaw damping above 0.2, except the combinations with double centre axle trailers. Thus, considering the performance of the current fleet and the correlation between winter and summer performance, a limit value of 0.2 is suggested.

**Figure 27.** Correlation between yaw damping ratio of vehicle combinations in the reference fleet in winter versus summer, during a 3 m SLC manoeuvre for different friction levels.

**Figure 28.** Yaw damping ratio of vehicle combinations in the reference fleet in a 3 m SLC manoeuvre.

### 4.6. Braking

In the PBS project, braking stability in turn has been investigated by Volvo on test track in winter conditions, where two HCT vehicles were tested. The test results indicate that the vehicles stay stable during braking in a turn on snow, when the ABS function works properly. But, malfunctioning of the ABS on any axle groups, as well as extreme slippery conditions, results in large offtracking. No simulation has been performed with respect to the braking stability in the PBS project due to lack of detailed brake models. An issue which should be further investigated in future is the effect of the braking delay on stability of long HCT vehicles during braking in a turn. It should be noted that in Volvo test results no considerable benefit was observed with using Electronic Braking System (EBS).
5. Assessment procedure

In a PBS based regulatory framework, compliance assessment of the vehicles with the standards is a key component. The vehicles can be assessed either via numerical modelling or physical testing. Numerical modelling is usually preferred due to its cost effectiveness. However, using numerical modelling in a legislation raise questions of model accuracy and model complexity. Thus, in the PBS project the required level of modelling details for assessing performance of heavy vehicles with respect to the candidate measures was investigated, results of which is presented in this chapter.

5.1. Traction

Three levels of model complexity and their impact on the three traction measures, namely startability, gradeability and acceleration capability, were investigated. The most comprehensive model is an OEM developed model, which is widely accepted internally at Volvo and is used for longitudinal dynamics simulations. The second level is a model taken from Kati et. al. (2014) which makes use of an engine map, considering effects of rotating parts, but neglects time delays in gear shifting. The third and simplest model, derived in the PBS project, is based on first principal modelling. The derived models are not simulation models, but simple expressions dependent on the most influencing components such as the maximum torque that the engine can produce. The derivation of these simple expressions is explained in the following sections.

5.1.1. Simple expression for startability

The most contributing components for the startability measure is the tyre and the powertrain and its ability to produce a high level of torque and to distribute this through the clutches/torque converters to the wheels. The maximum torque, $T_{eng}$, that the engine can produce as a function of engine rotational speed, $\omega$, can be considered known from the supplier. The transmission ratio of the first gear, $R_{tm,1}$, and the final drive ration, $R_{fd}$, can also be considered known as well as the tyre radius, $R_{whl}$. A maximum propelling force that can be achieved by the powertrain, given ideal clutch conditions without torque amplification, is then given by:

$$F_{PT max} = \max_{\omega} \frac{T_{eng}(\omega)}{R_{tm,1}} \frac{R_{fd}}{R_{whl}}$$

(1)

The tyres will, in some conditions, limit this propelling force by the friction $\mu$. Further on, it will be assumed that the change of the load transfer due to the slope is neglected for both total load, $Mg$, and the load on the driven wheels $N_{D}$. A simple expression for the maximum slope that the vehicle combination can commence is then given by the fraction of the maximum propelling force and the normal load, as:

$$S_{\%} = \left[ \min (N_{D} \mu F_{PT max}, D) \right] \frac{1}{Mg}$$

(2)

where the hard brackets indicate the integer part of the term and $\eta$ is an efficiency factor for the powertrain.

5.1.2. Simple expression for gradeability

Gradeability is very similar to startability but with the difference that the speed needs to be maintained. For gradeability with the maintained specific speed, $v_{ref}$, a similar expression as (2) can be derived. Here the speed is higher than for the startability measure; thus, an optimal gear needs to be found for this speed and air drag and rolling resistance of the tyres needs to be considered. An optimal gear can be found by maximizing the propelling torque for the given set speed, $v_{ref}$, according to:

$$i = \min_{i} T_{eng}(\omega_{i}) R_{tm}(i)$$

(3)
where the engine speeds $\omega_i$ for the gears, $i$, is given by:

$$\omega_i = \frac{v_{ref}R_fgR_{tm}(i)}{R_{whi}}$$  \hspace{1cm} (4)

The fraction of the maximum propelling force and the normal load, including the rolling resistance and the air drag, is given by:

$$G\% = \left[ \frac{100}{\eta} \left( \min(N_D \mu \frac{F_{PT \, max}}{Mg}) \eta - C_{rr} - 0.5 \rho_{air} C_d A_{front} v_{ref}^2 \right) \right]_{\text{INTEGER}}$$  \hspace{1cm} (5)

where $\eta$ is the same powertrain efficiency term as in (2), and $\rho_{air}$, $A_{front}$, $C_d$ and $C_{rr}$ are the density of air, effective frontal area, air drag coefficient and the rolling resistance coefficient.

### 5.1.3. Simple expression for acceleration capability

Acceleration capability is a measure of how fast a vehicle combination can clear for example a rail crossing from a stand still. The performance measure is the required time to complete the 100 meters. This measure depends heavily on the powertrains maximal capabilities, and not only at a certain speed as for the startability and gradeability measures. Here, a range of speeds from zero up to the speed that the vehicle combination reaches at the travelled 100 meters needs to be considered. This is largely due to the strong rotational speed dependence of the engine torque as well as the quantization effects of the transmission. A speed dependent maximum torque at the wheels can be obtained as:

$$F_{PT \, max}(v) = \max_i T_{eng} \left( \frac{v R_{tm}(i) R_{fg}}{R_{whi}} \right) R_{tm}(i) R_{fg}$$  \hspace{1cm} (6)

The travelled distance from this force can be derived by integrating the produced acceleration twice according to:

$$\dot{s} = v$$  \hspace{1cm} (7)

$$\ddot{s} = \frac{\min(F_{PT \, max}(v), \mu N_D) \eta}{M}$$  \hspace{1cm} (8)

Then the measure is obtained from the implicit equation:

$$s(t_{acc \_cap}) = 100$$  \hspace{1cm} (9)

### 5.1.4. Comparison results

The traction measured for the 9 vehicle combinations in Table 8 were calculated with models with different complexities; two different engine alternatives were considered. The achieved results are depicted in Figure 29.

It can be noticed that the general match between all three models is quite good. For the startability measure, the relative error is about 10-15% for most of the vehicle combinations with a higher relative error for the heavier combinations peaking at 30%. It should also be noted that the clutch component of the OEM model, used for calculation of startability, has been improved since this comparison study was performed. The improved OEM model shows that a clutch overloading can reduce the estimated startability for heavier combinations and decrease the accuracy of the simple expressions even further. However, for new heavy vehicles, clutch overloading will not be an issue due to improved torque transfer by larger final drive ratio or improved gearbox with crawler gears which will provide a gear ratio which is almost twice as large as a standard gearbox.

For the gradeability measure, the match between the simple expressions and OEM model is very good. This implies that the simplifications made are justified and the accuracy of the simple expression is within the rounding error of the measure. With the intermediate model, the measure is overestimated.
for almost all the cases with both large absolute and relative errors. This speaks against intuition that the accuracy increases with the model complexity.

Figure 29. Calculated traction measures with models with different complexity
The intermediate and simple models’ parameters have been changed to match the OEM models, but only the basic parameters are used by the simpler one. An explanation of the large discrepancy between the OEM and the intermediate models could be that parameters not used in the simple model may have a large impact if they are improperly tuned. This illustrates the importance of transparency and simplicity of the models used in a legislation. The acceleration capability measure shows approximately 20% relative error for the two least complex models compared to the OEM model for both engine cases.

To summarize, the performed comparison suggests that the simple expression could potentially be used for assessment of heavy vehicles performance with respect to traction measures. Furthermore, it can be concluded that accuracy and precision are not necessarily implied by model complexity. The simplicity of the models used to assess the performance measures would decrease the risk of inaccurate and erroneous results due to incorrect parameters and other mistakes in the model.

5.2. Low speed tracking

A simple model for calculation of the swept path in a low speed turning manoeuvre, described in section 4.2, was derived in the PBS project. The simple model generates the trajectory of the wheels, often called tractrix. Many tractrix equations can be found in the literature. For example, in Erkert et al. (1989) simplified tractrix equations are derived in polar coordinates and compared to test track measurements. Another example is given by Sayers (1991) using a general multi-body framework. Here, a similar but simplified approach is taken where all assumptions are explicit and traceable in the derivation.

Assuming that effects of the track width is negligible, two measurement points are introduced for each unit, see Figure 30, one at the steered axle and one at the rear axle. They are denoted by the Cartesian coordinates of a global reference frame \((x_1, y_1)\) and \((x_2, y_2)\) respectively.

![Figure 30. Schematic picture of a vehicle unit.](image)

The trajectory of the first point, given by the steering input and the speed of the vehicle, is assumed to be smooth, and that the time derivatives of the coordinates are well defined. Considering only kinematic motion, the following constraints on the two points and their motion can be stated:

1. The distance between the points will always remain constant, and equal to the length \(L\).
2. The lateral slip at the rear axle is zero, i.e. there is no motion perpendicular to the direction of the line between the points and the second point.

These constraints can be formalized as:

\[
\begin{align*}
(x_2 - x_1)^2 + (y_2 - y_1)^2 &= L^2 \\
(y_2 - y_1)x_2 - (x_2 - x_1)y_2 &= 0
\end{align*}
\]  \quad (10)

The polar coordinates for the distance between the points can be introduced according to:

\[
\begin{align*}
x_2 - x_1 &= r \cos(\beta) \\
y_2 - y_1 &= r \sin(\beta)
\end{align*}
\]  \quad (11)
Observe that the first constraint reduces to \( r = L \), while the second one becomes:

\[
L \sin(\beta) \left( \dot{x}_1 - L \sin(\beta) \dot{\beta} \right) - L \cos(\beta) \left( \dot{y}_1 + L \cos(\beta) \dot{\beta} \right) = 0 \tag{12}
\]

Rearranging the equation above and using the coordinate transformation will result in:

\[
\begin{aligned}
\dot{\beta} &= \frac{1}{L} (\dot{x}_1 \sin(\beta) - \dot{y}_1 \cos(\beta)) \\
x_2 &= x_1 + L \cos(\beta) \\
y_2 &= y_1 + L \sin(\beta)
\end{aligned}
\tag{13}
\]

For multiple of units, equation (13) can be used repetitively, assuming that the interaction between the units in a joint is only one way from the leading vehicle unit. In other words, it is assumed that a trailing unit does not affect the motion of the leading unit. However, it should be considered that the zero lateral motion point and the hinge point will be different. Denoting the length between the leading point and the hinge point for the \( i \)th vehicle unit by \( L_{hi} \) and the distance between the leading point and the point with zero slip by \( L_n \), equation (13) can be rewritten as:

\[
\begin{aligned}
\dot{\beta} &= \frac{1}{L_i} (\dot{x}_{i-1} \sin(\beta_i) - \dot{y}_{i-1} \cos(\beta_i)) \\
x_i &= x_{i-1} + L_{hi,i} \cos(\beta_i) \\
y_i &= y_{i-1} + L_{hi,i} \sin(\beta_i)
\end{aligned}
\tag{14}
\]

where \((x_i, y_i)\) now describes the global coordinates of the hinge points of the combination and \( \beta_i \) the absolute angle of the \( i \)th unit. The first point coordinates \((x_1, y_1)\) is assumed to be differentiable and given by the reference trajectory of the manoeuvre. The independent variable does not necessary need to be time, and should ultimately describe the evolution of the reference trajectories of \((x_1, y_1)\). However, using time has the advantages of making the derivative to be interpreted as speeds.

The offtracking of all the vehicles in the reference fleet were calculated in a roundabout at low speed, using the simple expressions derived here, as well as with the developed multibody dynamics models of the vehicles described in Chapter 3. Different turn angles in the roundabout were considered. The resulting offtracking values were compared and the relative error of the simple expression with respect to the vehicle models are illustrated in Figure 31. The relative difference between the two are less than 4% for all vehicle combinations and turn angles.

![Figure 31. Relative difference in percentage between the tractrix expression and the multibody dynamics models of the 22 vehicles in the reference model.](image-url)
This result suggests that the swept path can be computed by simple tractrix equations, using only geometrical distances of units, their wheel bases and distances to hinge points. This enables a very robust and transparent assessment of the low speed swept path measure.

5.3. High speed tracking and stability

To study the required level of model complexity for high speed performance measures, the developed high-fidelity models described in Chapter 3 were compared with a simpler yaw-plane single track linear model. The early analysis showed that inclusion of the tyre relaxation length has a significant effect, and that the validity of the single track linear model deteriorate considerably without it. Thus, the tyre relaxation length was included in the linear model as well.

The studied performance measures are maximum rearward amplification of yaw rate and lateral acceleration, lateral load transfers, yaw damping and high speed transient offtracking. The rearward amplifications can be calculated by linear frequency domain analysis, or by simulating single lane change manoeuvres at different frequencies; both approaches were used in this investigation. Damping ratio, high speed transient offtracking and load transfer ratio were calculated in a single lane change manoeuvre. It should be noted that the single track linear model is two dimensional and cannot be used for direct simulation of lateral load transfer. Therefore, the load transfer ratio for the linear models was estimated by post processing of the simulation data, using the track width, centre of gravity height and lateral acceleration.

The analysis was performed on three vehicle combinations, a truck-hauling two centre axle trailers, denoted as Double CAT, and Nordic and Adouble combinations. The effect of the payload height on the validity of the simpler models was investigated, by considering two different payload heights of 1.5 m and 4 m in the high-fidelity models. The relative error of the single track linear models with respect to the high-fidelity models are plotted in Figure 32. The results illustrate that the estimation error for the yaw rate rearward amplification by the simple model is less than 15% for most cases, so a 2D model might be sufficient for calculation of this measure. However, the estimation error for damping ratio can be more than 20% and the estimation errors for HSTO and LTR are substantial. Thus, a three-dimensional model is needed for calculation of these measures.

![Figure 32. Relative error of the single track linear model with respect to the high-fidelity models, plotted for two payload heights and three vehicle combinations.](image-url)
5.4. Steady-state rollover threshold

Steady-state rollover threshold (SRT) of a heavy vehicle is the maximum lateral acceleration that the vehicle could sustain in steady-state turning without rolling over. One of the commonly used numerical models for calculation of SRT is the calculation method in the ECE R111 regulation (UNECE 2001). However, the tilt table tests performed by Volvo, described in Chapter 2, indicates that the ECE R111 calculation method overestimates the SRT; one of the possible reasons for this overestimation is the tyre lateral deformation which is not taken into account in the ECE R111. Therefore, a modified analytical model for calculation of SRT, based on the ECE R111 is proposed here.

The proposed model considers the main factors that influence the rollover threshold, namely the height of centre of gravity, the track width and all factors that affect the vehicle roll stiffness, similar to ECE R111. Additionally, the proposed model incorporates the tyre lateral stiffness which has a considerable influence on the effective track, and consequently on the steady state rollover threshold. Linear parameters are used for suspension and tyre characteristics in the proposed model. Same approach as in ECE R11 is used to address the SRT calculation of roll-coupled units and effects of fifth wheel connection. For roll-coupled units, the fifth wheel is replaced by a representative roll stiffness and a representative track width.

In the following section the equations for calculating the SRT, according to the proposed model, are explained. It should also be noted that the proposed model is being considered for publication as an ISO standard.

5.4.1. Vehicle roll stiffness and effective tyre lateral stiffness

First the roll stiffness of each axles with respect to the ground plane should be calculated; to do so, the suspension roll stiffness and the roll stiffness from the tyre normal stiffness are added up. The equivalent suspension roll stiffness of the i:th axle, $K_{\varphi ESi}$, with respect to ground plane can be calculated from suspension roll stiffness with respect to the axle roll centre, $K_{\varphi Si}$, as:

$$K_{\varphi ESi} = K_{\varphi Si} \left( \frac{H_S}{H_S - H_{RCi}} \right)^2$$

(15)

where $H_S$ and $H_{RCi}$ are the height of sprung mass centre of gravity and suspension roll centre above the ground, respectively.

The roll stiffness from the tyre normal stiffness, $K_{\varphi Ti}$, is calculated by:

$$K_{\varphi Ti} = \frac{C_{ZTi} b_i^2}{2}$$

(16)

where $C_{ZTi}$ is the tyres normal stiffness at each side of the i:th axle and $b_i$ is the track width. Note that if an axle has dual mounted tyres, the track with will be calculated based on the distance of the points centrally located between the contact centres of the inner and outer dual wheels, $b_C$, and the dual tyre spacing, $S_{DT}$. Dual tyre spacing is the distance between the contact centres of the inner and outer dual wheels, see Figure 33:

$$b_i = \sqrt{b_C^2 + S_{DT}^2}$$

(17)

Adding up the two above terms, the roll stiffness of an axle, $K_{\varphi i}$, can be calculated as:

$$K_{\varphi i} = \frac{K_{\varphi ESi} K_{\varphi Ti}}{K_{\varphi ESi} + K_{\varphi Ti}}$$

(18)

Then the vehicle roll stiffness, $K_{\varphi V}$, which is sum of the individual axle roll stiffnesses, will be:

$$K_{\varphi V} = \sum_{i=1}^{n} K_{\varphi i}$$

(19)

If the vehicle consists of roll-coupled units connected via fifth wheel, a fifth wheel representative roll stiffness, $K_{\varphi fw}$, should be introduced based on the normal load on the fifth wheel, $N_{fw}$.
\[ K\varphi_{fw} = 4 N_{fw} \]  
Consequently, the vehicle roll stiffness will be:

\[ K\varphi_V = \sum_{i=1}^{n} K\varphi_i + K\varphi_{fw} \]  

Next the vehicle effective tyre lateral stiffness, \( C_{yT} \), should be calculated by summing up the tyres lateral stiffness at each side of all the axles, \( C_{yTi} \):

\[ C_{yTe} = \sum_{i=1}^{n} C_{yTi} \]  

The vehicle effective track width, \( b_v \), is defined as:

\[ b_v = \frac{\sum_{i=1}^{n} (b_i N_i)}{N_t} \]  
where \( N_i \) is the normal load on each axle, and \( N_t \) is the total normal load on all axles. Note that if the vehicle consists of roll-coupled units connected via fifth wheel, a fifth wheel representative track width, \( b_{fw} \), should be introduced:

\[ b_{fw} = \frac{\sum_{i=1}^{n} b_i}{n} \]  

Consequently, the vehicle effective track width will be:

\[ b_v = \frac{\sum_{i=1}^{n} (b_i N_i)}{N_t} + \frac{b_{fw} N_{fw}}{N_t} \]  

\[ b_i = \frac{b_i^2 + S_{dr}^2}{S_{dr}} \]

\[ S_{dr} \]

\[ b_v = \frac{\sum_{i=1}^{n} (b_i N_i)}{N_t} + \frac{b_{fw} N_{fw}}{N_t} \]

**Figure 33. Vehicle dimensions used in calculation of SRT.**

### 5.4.2. Calculation of SRT

After calculating the roll stiffness of each axle and the total roll stiffness and effective track width of the vehicle, the wheel lift-off at each axle can be estimated. The wheel lift-off happens when the overturning moment, \( M_o \), becomes as large as the restoring moment, \( M_R \). Each of these moments can be calculated based on the lateral acceleration, \( a_y \), as:

\[ M_R = N_i \left( \frac{b_i}{2} - \frac{a_y N_i}{C_{yTi}} \right) \]  

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\[ M_D = a_Y \left( D_{K\phi_i} N_t H_{CG} \left( \frac{D_{K\phi_i} (N_t - W_U) H_S^2}{K\phi_i - D_{K\phi_i} N_t H_S} \right) \right) \] (27)

\[ H_{CG} \text{ and } W_U \text{ are height of centre of gravity of the complete vehicle above the ground and total weight of the unsprung mass of the vehicle, respectively. } D_{K\phi_i} \text{ is the distribution of the vehicle roll stiffness between the individual axles expressed as a percentage of the vehicle roll stiffness:} \]

\[ D_{K\phi_i} = \frac{K\phi_i}{K\phi_V} \] (28)

Note that the restoring moment, \( M_D \), has a term dependent on the tyre lateral stiffness: \( a_Y N_t/C_{YTII} \), which is equal to the tyre lateral deformation. This term does not exist in the ECE R111 and is the main applied modification. The reason behind inclusion of this additional term is that in a turn, the tyre vertical force will not be acting at the tyre centre point and will be closer to the roll centre due to the tyre lateral deformation.

Thus, the lateral acceleration at which the first wheel lift-off happens, \( a_{yl} \), is equal to:

\[ a_{yl} = \min_i \left( \frac{N_i h_i}{D_{K\phi_i} N_t H_{CG} + \left( \frac{D_{K\phi_i} (N_t - W_U) H_S^2}{K\phi_i - D_{K\phi_i} N_t H_S} + \frac{N_t^2}{C_{YTII}} \right)} \right) \] (29)

While, the maximum theoretical lateral acceleration, \( a_{ym} \), at overturning moment is:

\[ a_{ym} = \frac{N_t h_t}{N_t H_{CG} + \left( \frac{(N_t - W_U) H_S^2}{K\phi_i - N_t H_S} + \frac{N_t^2}{C_{YTII}} \right)} \] (30)

Linear interpolation between the lateral acceleration at which the first wheel lift-off happens and the maximum theoretical lateral acceleration at overturning, gives the steady state rollover threshold, \( a_{SRT} \):

\[ a_{SRT} = a_{ym} - \left( a_{ym} - a_{yl} \right) \frac{N_i}{N_t} \] (31)

Examples of calculated steady state rollover threshold based on the proposed method, in comparison with tilt table results and values based on the ECE R111, are provided in Table 9. The proposed method provides more accurate results than ECE R111.

**Table 9. SRT calculation by different methods (m/s²)**

<table>
<thead>
<tr>
<th></th>
<th>5-axle full trailer, 42 t</th>
<th>5-axle dolly-semi-trailer, 42 t</th>
<th>6x2 Truck, 25 t</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tilt table test</td>
<td>3.28</td>
<td>3.28</td>
<td>4.16</td>
</tr>
<tr>
<td>ECE R111</td>
<td>3.72</td>
<td>3.84</td>
<td>4.54</td>
</tr>
<tr>
<td>Proposed method with ( C_{YT} = 200 \text{ kN/m} )</td>
<td>3.22</td>
<td>3.31</td>
<td>3.80</td>
</tr>
<tr>
<td>Proposed method with ( C_{YT} = 400 \text{ kN/m} )</td>
<td>3.45</td>
<td>3.56</td>
<td>4.16</td>
</tr>
</tbody>
</table>
6. **Driving simulator study**

A driving simulator study, recruiting professional truck drivers, was performed to evaluate and compare the performance of HCT and conventional vehicles in realistic driving conditions, and most importantly, to investigate the correlation between driver subjective evaluation of the vehicle performance and the objective performance measures. In this chapter, the performed simulator study and the achieved results are presented.

### 6.1. Driving simulator

The driving simulator study was performed in VTI’s driving simulator, Sim IV, located in Gothenburg. The Sim IV is equipped with a hexapod with six degrees of freedom (roll, pitch, yaw, surge, sway and heave) which is mounted on a sled with two extra degrees of freedom permitting significant linear movement along both longitudinal and lateral directions. It has three LCD displays for rear-view mirrors and a visual system comprising nine projectors. The visual system gives the driver a 210° forward field of vision. Vehicle and environment sound is provided by the vehicle speaker and a few complementing speakers, see Figure 34 (Jansson et al. 2014).

Before the main simulator study of the PBS project, a pre-study was performed in the Sim IV to tune the motion cueing algorithm of the Sim IV for improved driving experience with heavy vehicles. Main objective with the pre-study was to investigate whether Sim IV can provide a realistic driving experience with respect to the modifications of the vehicle model. This pre-study was supported by the competence centre ViP (Virtual Prototyping and Assessment by Simulation, www.vipsimulation.se), with contribution from the PBS project.

Tuning of the motion cueing algorithm was done together with a small group of professional test truck drivers, in an informal setting. Then the pre-study was conducted which consists of comparison of a baseline truck with 4 modified versions of it, each with its unique parameter set aimed at altering dynamic behaviour. The applied parameter changes resembled hardware changes which affect the vehicle’s handling characteristics and are probable in reality. The focus was on lateral vehicle dynamics. The driving simulator capability in representing the relative differences between the baseline and modified vehicles and the perceived handling of the vehicles by drivers were investigated by comparing the drivers perceived changes in the vehicle handling with the applied parameter changes and their expected effects.

Ten test truck drivers from Volvo participated in the pre-study, for which both subjective and objective data were gathered. The objective data consisted of the logged vehicle states during the simulator study and the subjective data was collected by questionnaires and an informal interview. Two questionnaires were used; first questionnaire focused on rating the overall realism of the driving simulator, as well as the usefulness of such driving simulator studies for vehicle dynamics testing. The second questionnaire focused on the perception of differences in the vehicle behaviour caused by changes in the vehicle model parameters.

![Figure 34. The VTI Sim IV. (Pictures: VTI)](image-url)
The drivers average rating of the usefulness of Sim IV for vehicle dynamics testing was 4.3 in a 5-grade scale, 1 being not useful at all and 5 very useful. This means that the test drivers think that such a motion based driving simulator is a useful tool for vehicle dynamics testing. Furthermore, the answers to the second questionnaire showed that Sim IV has the capability to provide the drivers with satisfactory feedback, and in most cases the drivers could correctly deduce the vehicle changes associated with each parameter set. The achieved results confirmed that Sim IV can be used for the simulator study in PBS project on driver perception of HCT vehicles performance. For more information on the pre-study refer to the ViP report (Augusto et al. 2017).

6.2. Driving scenario

To study truck drivers’ perception of HCT vehicles lateral performance and stability, and to compare it with their perception of the performance of conventional vehicles, a realistic driving scenario was developed.

The driving scenario consisted of driving on a model of road 180 between the cities of Borås and Alingsås, which is a rural curvy and uneven road with allowed speed of 80 km/h in major part of the road. The road model has been developed within the ViP Known-roads project, in which the actual road curvature and roughness has been measured and implemented (Nåbo et al. 2015). Furthermore, a fictional almost straight road section was added to the end of the modelled road, so that two overtaking events on straight road could be added to the driving scenario. In the implemented events the drivers were forced to overtake a slow-moving car. In one of the events they were also forced to drive back rather urgently to the original lane due to the meeting traffic. Additionally, they were asked to perform two double lane change (DLC) manoeuvres between cones at the end of the scenario. It should be also noted that each driver started the study with a 5-min driving training to get familiar with the driving simulator. They were instructed to drive freely during the training session and to perform DLC manoeuvres between cones. The driving was performed with an activated cruise control, set at 80 km/h, so that the driver could focus on the manoeuvring of the vehicle rather than keeping the desired speed.

6.3. Tested vehicle combinations

Each participant in the simulator study drove a pair of vehicles consisting of an HCT vehicle and a conventional vehicle. The three chosen vehicle pairs are:

- Tractor-semitrailer & Adouble
- Nordic & ABdouble
- Truck-centre axle trailer & Truck-double centre axle trailers

The objective was to investigate the performance of these vehicles with respect to high speed tracking and stability performance measures, namely high speed transient offtracking, yaw rate rearward amplification and load transfer ratio. The vehicle combinations were parameterized in a way that the vehicles in one of the pairs, the Nordic and ABdouble, had a similar performance with respect to lateral dynamics, although the ABdouble was longer and heavier.

Table 10. The lateral performance of the chosen vehicles for the simulator study

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>40 t, 16.5 m</td>
<td>80 t, 30 m</td>
<td>40 t, 19 m</td>
<td>74 t, 28 m</td>
<td>64 t, 25 m</td>
<td>83 t, 32 m</td>
</tr>
<tr>
<td>Yaw rate RA</td>
<td>1.08</td>
<td>1.84</td>
<td>1.65</td>
<td>1.65</td>
<td>1.31</td>
<td>2.12</td>
</tr>
<tr>
<td>HSTO (m)</td>
<td>0.29</td>
<td>0.92</td>
<td>0.88</td>
<td>0.95</td>
<td>0.49</td>
<td>1.29</td>
</tr>
<tr>
<td>LTR (%)</td>
<td>39</td>
<td>60</td>
<td>57</td>
<td>52</td>
<td>52</td>
<td>89</td>
</tr>
</tbody>
</table>
The modelled vehicles used in the driving simulator study were compared in an offline simulation with a single sine steer input with frequency of 0.3 Hz, resulting in a lateral acceleration of 1.5 m/s² at the front axle; the results are shown in Table 10. As explained, the Nordic and ABdouble combinations have similar performance, despite different length and weight.

6.4. Test drivers and collected data

55 professional truck drivers participated in the study, out of which only two were female drivers. The drivers were aged between 25 and 77 years old, with an average age of 47 years old. On average they drove heavy vehicles 3 days per week; only one of the drivers had driven HCT vehicles before.

The drivers were divided into three groups, each group drove one of the vehicle pairs. In each group, half of the drivers started the study by driving the conventional vehicle and then the HCT vehicle, while the other half started with driving the HCT vehicle.

The drivers were asked to fill in a questionnaire after each drive, so that their subjective evaluation can be compared with objective measures. The questionnaire included questions such as how easy it is to control the vehicle, how easy it is to keep the vehicle on the desired path, and how they rate the roll stability of the vehicle. A 7-grade scale was used in the questionnaire, which is provided in Appendix C. The drivers’ answers to the questionnaire were compared with the objective measures of interest, namely yaw rate rearward amplification, high speed transient offtracking and load transfer ratio.

Furthermore, each driver was asked to fill in an additional questionnaire about the simulator realism at the end of the study, provided in Appendix D.

6.5. Simulator study results

In this section the results of the driving simulator study are discussed. First the questionnaire results on the realism of the driving experience are provided. Then the correlation between drivers’ subjective evaluation of the vehicle performance and the objective measures are investigated, and finally a comparison of the performance of the driven HCT and conventional vehicles is presented.

6.5.1. Realism of the driving experience

The average rating of the realism of the driving experience in the simulator by the truck drivers was 5.2 in a 7-grade scale, 1 being not realistic at all and 7 very realistic. The drivers were also asked to rate the realism of the overtaking events in the driving scenario, which received an average rating of 5.8. Thus, it can be included that the driving experience and the encountered events were perceived as realistic by the drivers. The ratings for realism of other aspects of the driving experience by the drivers are provided in Table 11.

<table>
<thead>
<tr>
<th>Average rating 7-grade scale</th>
<th>Roads</th>
<th>Surrounding</th>
<th>Braking</th>
<th>Acceleration</th>
<th>Sound</th>
<th>Suspensions</th>
<th>Cabin vibrations</th>
<th>Steering feel</th>
<th>Manoeuvrability</th>
<th>Speed perception</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.0</td>
<td>4.6</td>
<td>4.7</td>
<td>4.8</td>
<td>4.7</td>
<td>4.8</td>
<td>4.9</td>
<td>4.7</td>
<td>4.7</td>
<td>4.7</td>
<td>4.7</td>
</tr>
</tbody>
</table>

6.5.2. Subjective evaluation versus objective measures

Using the logged vehicle states in the simulator, offtracking, yaw rate rearward amplification and load transfer ratio of the vehicles during the performed manoeuvres by the truck drivers were calculated. A sample of logged data for the ABdouble combination during the DLC manoeuvres and the overtaking events, which is used for the calculation of the yaw rate RA during the driving scenario, is shown in Figure 35.
Figure 35. Logged yaw rate of the first and last units of the ABdouble combination when a test driver is overtaking or driving between the cones. The yaw rate peaks used for calculation of the maximum rearward amplification during the manoeuvres are shown with * marks in the plots.

Figure 36. Sample offtracking for the truck with double centre axle trailers when a test driver is overtaking or driving between the cones, calculated from the logged position of the first and last axles.
Figure 36 shows an example of the calculated offtracking for the truck with double centre axle trailers combination during the overtaking events and the double lane change manoeuvres. Another example for the calculated load transfer ratio of the Adouble combination, while driving through the curvy section of the road 180 in the driving scenario, is depicted in Figure 37.

In addition to the objective measures which were calculated using the logged driving data in the simulator, data on subjective assessment of the vehicles performance was gathered using questionnaires. The drivers were asked to rate (in a 7-grade scale) different aspect of the vehicle performance during the driving scenario with questions such as:

- How easy it is to keep the vehicle on the desired path on the curvy parts of the road.
- How they perceive the roll stability of the vehicle on the curvy parts of the road.
- How they perceive the trailers, stable or oscillatory, during overtaking and lane changes.
- How easy it is to control the vehicle.

There was also one concluding question, asking the drivers to compare the controllability of the two vehicles they drove.

For the curvy sections of the road, the correlation between the offtracking of the vehicles with the drivers’ ratings of the difficulty of keeping the vehicle on the desired path was investigated, as well as the correlation between the load transfer ratio and the drivers’ ratings of the roll stability of the vehicle. The results are depicted in Figure 38, which show a strong correlation for both cases, with correlation coefficients of 0.88 and 0.89.

Figure 38. Correlation between offtracking and load transfer ratio with drivers’ ratings of the vehicle performance while driving through curves, a lower rating by drivers means better performance.
Figure 39. Correlation between yaw rate RA and offtracking with drivers’ ratings of trailer oscillations during the DLC manoeuvres, a lower rating by drivers means better performance.

Figure 40. Correlation between yaw rate RA and offtracking with drivers’ ratings of trailer oscillations during overtaking event, a lower rating by drivers means better performance.

Figure 41. Correlation between yaw rate RA and offtracking during the DLC manoeuvres with drivers’ ratings of the vehicle controllability, a lower rating by drivers means better performance.

Next, the correlations between the yaw rate rearward amplification and offtracking with drivers’ ratings of the vehicle performance during the DLC manoeuvres and the overtaking events were studied. The results show that there is an evident correlation between the studied objective measures and the driver ratings of the vehicle performance, see Figure 39 and Figure 40. The strongest correlation is between the yaw rate rearward amplification and the driver ratings of the trailer oscillatory behaviour during the DLC manoeuvres. This can be explained by the fact that the DLC is a more severe manoeuvre compared to the overtaking performed by the drivers. Thus, the vehicle is excited more severely resulting in more sensible oscillations by the drivers. Drivers rating of the trailer oscillations during the overtaking has a stronger correlation with the offtracking than the yaw rate RA of the vehicle. One possible explanation is that the drivers are more focused on avoiding encroaching into the opposite lane and hitting other vehicles during a lane change, which is what the offtracking measure describes.
Figure 42. Correlation between drivers’ ratings of the vehicles controllability with combined yaw rate RA and offtracking measures, a lower rating by drivers means better performance.

The correlations between the yaw rate RA and offtracking with the drivers’ ratings of the vehicle controllability is illustrated in Figure 41, which shows similar pattern as in previous figures. Note that in all the three plots of correlation between the rearward amplification and the vehicle perceived performance by the drivers, the ABdouble combination seems to be rated worse by the drivers with respect to the actual RA value. This can be explained by the fact that although the ABdouble combination has rather low RA values, comparable with the Nordic combination, but it has relatively larger offtracking due to its configuration and length. This highlights that the perceived performance and controllability of the vehicle by the drivers is related to both offtracking and rearward amplification. In Figure 42, the correlation between drivers’ perceived controllability of the vehicles and an objective measure combining both RA and offtracking values (RA+5 times offtracking) is depicted showing a very strong correlation with coefficient of 0.98.

6.5.3. Comparison of HCT and conventional vehicles by the drivers

The drivers were asked to compare the controllability of the vehicles they drove by the question: “how easy/difficult it is to drive the HCT vehicle in comparison with the conventional vehicle”. It was a 7-grade scale question with 1 being much easier and 7 much more difficult. The results are presented in Table 12. The drivers’ ratings show that an HCT vehicle is not necessarily more difficult to drive than a conventional vehicle. For instance, for the ABdouble and Nordic pair, which were parameterized to have similar performance with respect to lateral dynamics measure, the drivers’ ratings also indicate that they are almost equally controllable. For the other two pairs, the drivers have rated the HCT vehicle to be slightly more difficult to drive, compared with the conventional vehicle. This is aligned with the fact that in those pairs, the HCT vehicle has significantly larger rearward amplification, offtracking and load transfer ratio.

To conclude, the driving simulator study results confirm that a PBS scheme is a better means to evaluate performance of heavy vehicles than length or weight. Furthermore, performance measures are strongly correlated with the perceived performance of the vehicles by the drivers.

Table 12. Comparison of controllability of the HCT and conventional vehicles by the 54 truck drivers

<table>
<thead>
<tr>
<th></th>
<th>Adouble vs. Tractor-Semi</th>
<th>ABdouble vs. Nordic</th>
<th>Truck-CAT vs. Truck-double CAT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average rating, 1-7 scale</td>
<td>4.8 (slightly more difficult)</td>
<td>3.7 (almost the same)</td>
<td>4.9 (slightly more difficult)</td>
</tr>
</tbody>
</table>

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7. Noise Emissions

All the existing regulations with respect to environmental aspects of heavy vehicles are already performance based; thus, proposing new standards for HCT vehicles was deemed to be unnecessary. However, there are concerns about the noise emissions of HCT vehicles, since a long heavy vehicle combination is equipped with more tyres. Therefore, a simulation study on noise emissions was performed in the PBS project, which is presented in this chapter.

By means of a Vehicle Noise Simulation (VNS) tool, the noise emissions of an HCT vehicle and a conventional heavy vehicle were compared. The considered conventional vehicle is a Nordic combination which is one of the most common vehicle combinations for transport of timber and other goods in Sweden. The HCT vehicle is an ABdouble combination, see Figure 43. The noise emissions during different cases of pass-by test were simulated, including 50 and 80 km/h, and the shortest distances of 7.5 and 30 m, from the vehicle centre to the observer (microphone). The input data consisted of noise levels for tyres measured in an earlier project (Blokland et al. 2015).

![Figure 43. The conventional (top) and the HCT (bottom) vehicles in the noise emission simulations.](image)

7.1. The vehicle noise simulation tool

The VNS tool has been developed by Dr Piotr Mioduszewski, Technical University of Gdansk, Poland, by request from VTI. An older version of the tool has been used in studies at VTI during the 1990’s. However, the old program was based on an operating system no longer used in modern computers. Therefore, a new version of the VNS tool, adapted to Windows, was developed for the purpose of this study.

The conventional and HCT vehicles under study were created in the VNS tool, as illustrated in Figure 44. Note that at the front of the vehicles, there are two axles; this is due to the fact that the first axle represents the power unit (engine, fans, exhaust, etc), and not tyres. The power units were modelled as tyres, since there was no other provision in VNS to add a power unit source of noise. The second axle is the actual steering axle.

The VNS calculates the noise emission versus distance along the road, when the studied vehicle passes-by the roadside location of the microphone. The noise calculation starts when the vehicle front is 20 m away from the roadside point closest to the microphone, and continues until the front of the vehicle is 50 m ahead of this point; i.e. for a total of 70 m of vehicle travel. A longer distance can be
chosen if desired. The time history of the noise was calculated, which is SPL (sound pressure level) versus distance travelled. In addition, some overall noise levels were calculated, which are:

- $L_{\text{max}}$: The maximum noise level during the pass-by (measured with time constant $F$), in dB(A)
- $L_{\text{eq}}$: The equivalent noise level during the 70 m travelled distance, in dB(A)
- $SEL$: Sound Exposure Level, normalized to one second period, in dB(A)

The latter two are “averages” based on the accumulated acoustic power during the pass-by test. There is also a fourth measure -- $SENEL$ which is the Single Event Noise Exposure Level -- but this is nominally equal to the $L_{\text{eq}}$, and thus not presented separately in this study.

The two following speeds and driving conditions were chosen for the simulations. The latter is assumed to be a typical driving behaviour going through a village, when it is difficult to keep a constant speed:

- Cruising at constant speed at 80 km/h
- Moderate acceleration at an average speed of 50 km/h

In practice, the noise emitted from the side of the vehicle that is furthest from the microphone is partly screened by the vehicle body and by tyres on the nearest side. This effect may be substantial on cars (having relatively low ground clearance), and for microphone positions high above ground, but for these trucks with relatively open structures, any such screening should be minor and was neglected here.

The VNS tool assumes that there is no ground absorption of sound when it propagates from the sources to the microphone. The ground is assumed to be acoustically hard such as asphalt or well packed gravel or soil. This is no problem at 7.5 m distance, unless the road surface is porous, but at 30 m there is often loose soil, grass and bushes, or even a ditch, which may provide some absorption. Such absorption will essentially have the same effect for the two vehicles studied here; thus, it does not affect the comparison.

![Figure 44. The heavy vehicles created in the VNS tool.](attachment:figure44.png)

7.2. Tyre noise data

The following tyres, which are among common truck tyres in the market, were selected for the simulation due to the existence of recently measured noise levels for them (Blokland et al. 2015):

- For the steering axle: Michelin Xline Energy Z, 315/70 R 22.5
- For the drive axles (twin mounting): Michelin Xline Energy D, 315/70 R 22.5
- For the trailer axles: Michelin Xline Energy T, 385/55 R 22.5
Note that the trailer tyres are much wider, and may carry about 1000 kg more load per tyre, than the others. The drive axle tyres were the same for the outer and the inner pairs. However, to take into account the fact that the outer tyres partly screen the inner pairs, the inner tyres were assigned 1 dB(A) lower noise level than the outer tyres. This is an expert judgement based on earlier experience. The drive axle tyres are classified as M+S tyres; thus, they are likely to be used in Sweden all year round.

The noise levels representing the three selected tyres were taken from a report of a study made in a so-called NordTyre research programme (Blokland et al. 2015), where the noise level ($L_{\text{max}}$) of 32 truck tyres from the market were measured on a number of road surfaces, including one ISO 10844 surface. The noise levels on these four surfaces at a coast-by-speed of 70 km/h were averaged arithmetically to give a noise level as input to this study. Using a speed exponent of 3.3, typical of the data in (Blokland et al. 2015), the noise levels at 70 km/h were extrapolated to 50 and 80 km/h, which were the test speeds in the simulation.

The noise levels were increased in the cases where it was considered that the drive axle tyres were subject to significant torque, as a noise level correction for torque on the tyres, in relation to free-rolling tyres:

- 50 km/h, conventional vehicle: + 3 dB(A) for the drive axle tyres
- 50 km/h, HCT vehicle: + 4 dB(A) for drive axle tyres (higher load to pull requires more torque)
- 80 km/h, conventional vehicle: + 1 dB(A) for the drive axle tyres
- 80 km/h, HCT vehicle: + 2 dB(A) for drive axle tyres (higher load to pull requires more torque)

The noise levels were further adjusted for directionality as follows:

- Steering axle tyres, Michelin Xline Energy Z: No adjustment, supposed to be omnidirectional
- Drive axle tyres, Michelin Xline Energy D: +2 dB at front and rear directions, + 1 dB at 45, 135, 225 and 315 degrees relative to the rolling direction
- Trailer axle tyres, Michelin Xline Energy T: Same as for the drive axles. This is motivated by the tyres being extra wide.

These adjustments are judged to be very conservative. It is known that tyres emit noise differently in different directions; almost always with higher levels in the forward and rearward directions than to the sides. As there is too little data published about this, very conservative additions were considered.

### 7.3. Power unit noise

Both vehicles are assumed to have the same engine power. However, the HCT vehicle will need a higher degree of utilized power as it has more mass to pull. This is accounted for by assuming that the power unit of the HCT vehicle emits 2 dB(A) more noise at both speeds of 50 and 80 km/h driving conditions. Thus, it has been assumed that the power unit source emits:

- 77 dB(A) for the driving of the conventional vehicle, at both 50 and 80 km/h
- 79 dB(A) for the driving of the HCT vehicle, at both 50 and 80 km/h

Since there is no option in VNS for a power unit source, it was simulated as “tyres” noise. Thus, the overall 77 dB(A) for the conventional vehicle power unit is shared as 74 dB(A) on the left “tyre” source and 74 dB(A) for the right “tyre” source. A similar approach is used for the HCT vehicle.

The power unit noise appears to have little or no influence on the overall noise of the vehicles. Even if it has been underestimated by a couple of dB here, it would not affect the results at 80 km/h, and just marginally affect the results at 50 km/h. But even in the latter case, it will not influence the comparison of the vehicles significantly.
7.4. Simulation results

The simulation results are shown in Table 13, followed by diagrams in Figure 45 and Figure 46. It should be noted that the vertical scales are determined automatically by the VSN software; therefore, when comparing results, one must be aware of the incompatible vertical scales.

The differences between the conventional and HCT vehicles are small, in the range of 1.1 – 1.7 dB(A), but systematic and consistent for all studied cases. The noise difference is marginally higher, the higher the speed is. As expected, the HCT vehicle emits more noise; however, fewer HCT vehicles are needed to carry the same load as conventional vehicles and this can balance the higher noise level per vehicle. Assuming 25 % fewer HCT vehicles than the conventional vehicles on road, will result in a noise reduction of 1.25 dB(A), in terms of daily L_{eq} and according to common calculations in traffic noise prediction models (10xlog(3/4)). This almost exactly balances the increase in per-vehicle-noise emission according to the simulation results. Consequently, for noise exposure along the road, expressed by L_{eq} values, the choice of vehicle has no significant effect (< 0.5 dB(A)).

At hindsight, for the 30m source-microphone distance, it might have been better to extend the studied distance of travel from the chosen “-20 to +50 m”, to “-50 to +80 m”. However, it is unlikely that it would have changed the comparison of the vehicles significantly.

**Table 13. Noise emission simulation results**

<table>
<thead>
<tr>
<th>Vehicle combination</th>
<th>Mic. distance, m</th>
<th>Speed, km/h</th>
<th>Driving type</th>
<th>Noise level, dB(A)</th>
<th>Noise level diff, dB(A)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Based on the maximum pass-by noise level (L_{max})</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conventional</td>
<td>7.5</td>
<td>50</td>
<td>Moderate accel.</td>
<td>84.4</td>
<td></td>
</tr>
<tr>
<td></td>
<td>7.5</td>
<td>80</td>
<td>Constant speed</td>
<td>88.9</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>50</td>
<td>Moderate accel.</td>
<td>73.3</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>80</td>
<td>Constant speed</td>
<td>78.0</td>
<td></td>
</tr>
<tr>
<td>HCT</td>
<td>7.5</td>
<td>50</td>
<td>Moderate accel.</td>
<td>85.5</td>
<td>1.1</td>
</tr>
<tr>
<td></td>
<td>7.5</td>
<td>80</td>
<td>Constant speed</td>
<td>90.3</td>
<td>1.4</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>50</td>
<td>Moderate accel.</td>
<td>74.6</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>80</td>
<td>Constant speed</td>
<td>79.5</td>
<td>1.5</td>
</tr>
<tr>
<td><strong>Based on the equivalent pass-by noise level (L_{eq})</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conventional</td>
<td>7.5</td>
<td>50</td>
<td>Moderate accel.</td>
<td>80.9</td>
<td></td>
</tr>
<tr>
<td></td>
<td>7.5</td>
<td>80</td>
<td>Constant speed</td>
<td>85.5</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>50</td>
<td>Moderate accel.</td>
<td>72.4</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>80</td>
<td>Constant speed</td>
<td>77.1</td>
<td></td>
</tr>
<tr>
<td>HCT</td>
<td>7.5</td>
<td>50</td>
<td>Moderate accel.</td>
<td>82.2</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td>7.5</td>
<td>80</td>
<td>Constant speed</td>
<td>87.2</td>
<td>1.7</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>50</td>
<td>Moderate accel.</td>
<td>73.7</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>80</td>
<td>Constant speed</td>
<td>78.7</td>
<td>1.6</td>
</tr>
<tr>
<td><strong>Based on the sound exposure level (SEL)</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conventional</td>
<td>7.5</td>
<td>50</td>
<td>Moderate accel.</td>
<td>87.9</td>
<td></td>
</tr>
<tr>
<td></td>
<td>7.5</td>
<td>80</td>
<td>Constant speed</td>
<td>90.5</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>50</td>
<td>Moderate accel.</td>
<td>79.4</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>80</td>
<td>Constant speed</td>
<td>82.1</td>
<td></td>
</tr>
<tr>
<td>HCT</td>
<td>7.5</td>
<td>50</td>
<td>Moderate accel.</td>
<td>89.2</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td>7.5</td>
<td>80</td>
<td>Constant speed</td>
<td>92.2</td>
<td>1.7</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>50</td>
<td>Moderate accel.</td>
<td>80.8</td>
<td>1.3</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>80</td>
<td>Constant speed</td>
<td>83.7</td>
<td>1.6</td>
</tr>
</tbody>
</table>
Figure 45. Noise emission results for moderate acceleration at 50 km/h.

Figure 46. Noise emission results for constant speed at 80 km/h.
7.5. A note regarding maximum noise levels

It should be noted that these vehicles do not frequently travel in urban areas, where noise exposure from road traffic due to high volumes of traffic reach harmful levels. Therefore, the $L_{eq}$ level for these vehicles is of limited interest, as it would hardly contribute to too high $L_{eq}$ levels. Instead, it is the maximum level $L_{\text{max}}$ which is the most relevant to consider in this case. The EU regulations or recommendations deal only with $L_{\text{den}}$ values, where “den” stands for Day-Evening-Night, and levels during night time are subject to 10 dB penalty and evening levels are subject to 5 dB penalty on top of the $L_{eq}$ during daytime. In Sweden, however, $L_{eq}$ during 24 h are used in noise standards. In addition, $L_{\text{max}}$ values are considered in the guidelines for existing homes near roads, and this is unique for Sweden. In the “Target values” (Riktvärden) practiced in Sweden, $L_{\text{max}}$ above 70 dB(A) in a garden or patio location at a home is considered as undesirable. Such levels should not be exceeded more than 5 times per “average maximum-hour” during the day-evening time period, 06-22, see further (NV 2017). There are complicated standards also for maximum levels during night time, to avoid awakening effects. In these standards, it is assumed that 70 dB(A) at the facade outside the bedroom corresponds to 45 dB(A) in the bedroom (NV 2017).

Since the standards consider both the maximum level and the number of times when that level is exceeded, it is very complicated to say what the effect of the HCT versus conventional vehicles is. It is more likely that the 70 dB(A) level is exceeded by the HCT vehicles, see for example the diagrams for 30 m distance in Figure 45 and Figure 46. On the other hand, if the conventional vehicles also emit maximum levels above 70 dB, the number of such events would be lower for HCT vehicles.

To summarize, considering the above discussions with respect to $L_{eq}$ and $L_{\text{max}}$, noise emission is not a factor that gives either vehicle type a clear advantage over the other.
8. Discussion

In the PBS project, infrastructure, environmental and safety aspects of HCT vehicles have been investigated. The infrastructure aspects, which has been focused on road network categorization with respect to 74t vehicles, was mostly performed by Trafikverket, results of which can be found in Trafikverket reports (Trafikverket 2014, Trafikverket 2016).

All the existing regulations with respect to environmental aspects of heavy vehicles are already performance based; thus, proposing new standards for HCT vehicles was deemed to be unnecessary. However, to address the concerns about the noise emissions of HCT vehicles, a short simulation study was performed on comparison of noise emissions of a conventional and an HCT vehicle. The study showed that the resulting noise emission for a certain transport task is very similar for both vehicles.

During the PBS project extensive research were performed on safety aspects of HCT vehicles resulting in a proposal of a PBS scheme, using test track experiments, offline simulations and driving simulator studies. The performed studies have been described in the previous chapters, here a summary of the achieved results is provided.

The driving simulator study showed that there is a strong correlation between drivers’ perceived performance and controllability of the vehicles with the performance measures investigated in the study, namely high speed stability and tracking measures. Furthermore, the study results indicated that controlling and driving an HCT vehicle does not need more effort than driving a conventional heavy vehicle, if the vehicles have similar performance values. Thus, it was confirmed that using a PBS scheme for assessing heavy vehicles is a better approach for ensuring their safe performance than limiting their length and weight.

One of the main objective of the PBS project was to investigate the applicability of PBS in Sweden with attention to winter road conditions. Thus, the safety aspects which should be considered with respect to winter conditions were investigated, resulting in proposals for safe performance levels. Moreover, the required complexity of models for accurate assessment of heavy vehicles with respect to the performance measures in the proposed PBS scheme were identified. The results are summarized through Table 14 to Table 19.

Table 14. Proposal for a PBS scheme, traction measures

<table>
<thead>
<tr>
<th>Winter vs. summer</th>
<th>Performance level</th>
<th>Assessment procedure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Startability</td>
<td>12%, to be coherent with European regulations.</td>
<td>Simple expressions can be used to assess vehicles performance, see section 5.1.1.</td>
</tr>
<tr>
<td>Performance level</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gradeability</td>
<td>Effect of low friction on this measure is not significant; it is enough to assess under summer conditions.</td>
<td>Maintain 70 km/h on a 1% grade, suggested by Transportstyrelsen.</td>
</tr>
<tr>
<td>Performance level</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Acc. capability</td>
<td>Effect of low friction on this measure is not significant; it is enough to assess under summer conditions.</td>
<td>Instead of this measure, a criterion on engine power can be assigned, to be coherent with European regulations 5 kW/t can be used.</td>
</tr>
<tr>
<td>Performance level</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td></td>
<td>Simple expressions can be used to assess vehicles performance, see section 5.1.3.</td>
</tr>
</tbody>
</table>
### Table 15. Proposal for a PBS scheme, low speed tracking measures

<table>
<thead>
<tr>
<th>Winter vs. summer</th>
<th>Performance level</th>
<th>Assessment procedure</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Frontal swing</strong></td>
<td>Effect of low friction on this measure is not significant; it is enough to assess under summer conditions.</td>
<td>The European regulations for heavy vehicles manoeuvrability does not include frontal swing. Thus, it can be excluded from the PBS scheme for HCT vehicles.</td>
</tr>
<tr>
<td><strong>Performance level</strong></td>
<td>0.8 m (1m for vehicles with retractable axles in the lifted position, or loadable axles in the unladen condition), to be coherent with European regulations.</td>
<td>Simple expressions can be used to assess vehicles performance, see section 5.2.</td>
</tr>
<tr>
<td><strong>Assessment procedure</strong></td>
<td>Simple expressions can be used to assess vehicles performance, see section 5.2.</td>
<td></td>
</tr>
</tbody>
</table>

### Table 16. Proposal for a PBS scheme, high speed tracking measures

<table>
<thead>
<tr>
<th>Winter vs. summer</th>
<th>Performance level</th>
<th>Assessment procedure</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Tracking ability on a straight path</strong></td>
<td>Maximum offtracking of 0.4 m on a straight road is suggested by Transportstyrelsen. The road characteristics is not defined in Transportstyrelsen proposal. The nominal values of road crossfall in Swedish road design guidelines which is 2.5-3% on a straight road should be considered. Further investigation is required.</td>
<td></td>
</tr>
<tr>
<td><strong>Performance level</strong></td>
<td>This measure is anticipated to be highly correlated with the other two high speed tracking measures; Thus, it can be excluded from the PBS scheme for HCT vehicles.</td>
<td></td>
</tr>
<tr>
<td><strong>Assessment procedure</strong></td>
<td>A 3D model including the roll dynamics is needed, inclusion of tyre relaxation length is important, see section 5.3.</td>
<td></td>
</tr>
</tbody>
</table>

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Table 17. Proposal for a PBS scheme, low speed stability measures

<table>
<thead>
<tr>
<th>Friction demand of steer tyres</th>
<th>Winter vs. summer</th>
<th>Simulation results suggest that the interpretation of the friction demand measure on high friction is ambiguous, and in addition to the friction level, the tyre stiffness has also a significant impact on the friction demand, see section 4.4.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Performance level</td>
<td>Due to ambiguity of this measure on high friction, either it should be assessed on low friction directly, or another criterion such as regulating the load portion on the steer axle should be considered.</td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td>Not investigated.</td>
<td></td>
</tr>
<tr>
<td>Friction demand of drive tyres</td>
<td>Winter vs. summer</td>
<td>Simulation results suggest that the interpretation of the friction demand measure on high friction is ambiguous, and in addition to the friction level, the tyre stiffness has also a significant impact on the friction demand, see section 4.4.</td>
</tr>
<tr>
<td>Performance level</td>
<td>Due to ambiguity of this measure on high friction, either it should be assessed on low friction directly, or another criterion such as regulating the load portion on drive axles should be considered.</td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td>Not investigated.</td>
<td></td>
</tr>
</tbody>
</table>

Table 18. Proposal for a PBS scheme, high speed stability measures

<table>
<thead>
<tr>
<th>Steady-state rollover threshold</th>
<th>Winter vs. summer</th>
<th>Rollover stability is not worsened under winter conditions due to lower levels of lateral acceleration.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Performance level</td>
<td>Transportstyrelsen has suggested 3.5 m/s². Further tests and comparison of conventional and HCT vehicles is required.</td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td>A modified version of the calculation method in ECE R111 is proposed. The proposed method includes the effect of tyre lateral stiffness, see section 5.4.2.</td>
<td></td>
</tr>
<tr>
<td>Load transfer ratio</td>
<td>Winter vs. summer</td>
<td>Rollover stability is not worsened under winter conditions, due to low levels of lateral acc.</td>
</tr>
<tr>
<td>Performance level</td>
<td>This measure is anticipated to be highly correlated with steady state rollover threshold and rearward amplification of lateral acceleration. Thus, it can be excluded from the PBS scheme for HCT vehicles. However, there is still a question about including lateral acceleration RA in the PBS scheme, in addition to yaw rate RA.</td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td>A 3D model with roll dynamics is needed, inclusion of tyre relaxation length is important, see section 5.3.</td>
<td></td>
</tr>
<tr>
<td>Yaw damping ratio</td>
<td>Winter vs. summer</td>
<td>The yaw rate RA values for winter and summer are similar. However, the difference gets exaggerated for vehicles with poor performance, resulting in swing out in winter.</td>
</tr>
<tr>
<td>Performance level</td>
<td>The yaw rate RA in summer condition should be below a certain value to ensure that the vehicle will not be prone to swing outs in winter. Simulation results suggest a limit value of 2.2. The proposed value of 2.4 by Transportstyrelsen might be high, see section 4.5.</td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td>A 3D model with roll dynamics is needed if high accuracy is demanded; however, a 2D model can provide estimates with an error below 20%. Inclusion of tyre relaxation length is important, see section 5.3.</td>
<td></td>
</tr>
<tr>
<td>Yaw damping ratio</td>
<td>Winter vs. summer</td>
<td>The slippery winter condition does not have a significant effect on yaw damping in normal ranges. However, vehicles with very low damping in summer will lose stability in winter.</td>
</tr>
<tr>
<td>Performance level</td>
<td>The simulation results suggest a limit value of 0.2. The proposed value of 0.15 by Transportstyrelsen might be low, see section 4.5.</td>
<td></td>
</tr>
<tr>
<td>Assessment procedure</td>
<td>A 3D model including the roll dynamics is needed, inclusion of tyre relaxation length is important, see section 5.3.</td>
<td></td>
</tr>
</tbody>
</table>
Table 19. Proposal for a PBS scheme, braking measures

<table>
<thead>
<tr>
<th>Winter vs. summer</th>
<th>Experiments by Volvo on test track suggests that HCT vehicles maintain stability while braking in a turn with a functioning ABS: no considerable benefit was observed with usage of EBS. Under extreme slippery conditions (wet ice) the vehicle would have large offtracking while braking in a turn.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Performance level</td>
<td>A braking delay under 0.6 for the whole vehicle is suggested by Transportstyrelsen. Further investigation on the effect of braking delay is required.</td>
</tr>
<tr>
<td>Assessment procedure</td>
<td>Not investigated.</td>
</tr>
</tbody>
</table>

8.1. Outlook

There is still a need for further investigation on some of the performance measures, as can be seen in Table 14 to Table 19. For instance, the effect of braking delay on stability of the heavy vehicles during braking in a turn should be studied and more tests on steady-state rollover threshold of heavy vehicles should be performed. There is also a need for development of a PBS tool to be used by the authorities for assessing the HCT vehicles performance, as well as by the hauliers and manufacturers for developing and selecting safe and efficient HCT vehicles. Development of such a tool initiated in the PBS project and a beta-version of an open access PBS tool has been developed, which is described in a Chalmers report, see (Jacobson et al. 2017). Additionally, during the project, Transportstyrelsen has been working on development of a web-based tool for assessment of the 74ton HCT vehicles. The web-tool needs to be extended to include more performance measures, especially with respect to longer HCT vehicles. Moreover, the included assessment models/algorithms in Transportstyrelsen web-tool needs to be validated and improved considering the outcomes of the PBS project.

Another important aspect which needs further investigation is the effect of tyre characteristics on the performance of the HCT vehicles. As stated in Chapter 3, due to the existing variety of tyres and the diversity of road surface condition, especially during winter, choosing one tyre for assessing HCT vehicles is not a trivial task. Therefore, there is a need for defining and modelling standard tyres to be used in assessment of HCT vehicles with respect to a Swedish PBS scheme. To do so, the existing range of truck tyres in the market should be analysed, compared and categorized so that one or more representative tyres can be selected and modelled as standard tyres.

The issues of double versus single mounted tyres for HCT vehicles should also be studied further. To study the effect on lateral stability of HCT vehicles, Volvo performed some tests on a 54t double centre axle trailer combination with single and double mounted tyres, commissioned by Trafikverket. The test results showed that rearward amplification is lower with double mounted tyres, however, the difference depends on the tyre air pressure (Volvo 2017b). Further studies with other HCT vehicles should be performed before Volvo test results can be generalized. Moreover, a more holistic view, considering other aspects, such as rollover stability which is better for vehicles with single mounted tyres due to larger track width, should be applied.

8.2. Project publications

More information on the outcomes of the project can be found in the following publications, listed in order of publication year:

• Kharrazi, S., “Performance of high capacity vehicles – winter versus summer”. In proceedings of the International Symposium on Heavy Vehicle Transport Technology (HVTT14), Rotorua, New Zealand, 2016.
• Bruzelius, F., Kharrazi, S., “Low speed performance based standards for Nordic countries”. Submitted for journal publication
References


Volvo (2014). “Rollover stability calculation on three timber trailers according to ECE Regulation 111, and comparison with testing results”. Volvo Engineering report, ER-655480.

Volvo (2016a). “The timber B-double combination FH-1825/ST-DRAG was tested with 74 ton GCW at AstaZero. The main purpose was to evaluate the roll stability and lateral stability for the combination having a high center of gravity payload. A steering robot was used”. Volvo Engineering report, ER-661174.


Volvo (2017a). “Brake testing in winter conditions on a 74 ton AB-double combination showed good stability at panic braking, but loss of ABS-function on any axle group quickly resulted in unmanagable situations”. Volvo Engineering report, ER-665621.

Appendix A. Model validation against test data, summer tests

Figure 47. TR6x4-ST2, 14 m, 40 t, summer

Figure 48. TR6x4-ST2-DY2-ST2, 25.7 m, 74 t, summer
Appendix B. Model validation against test data, winter tests

Figure 49. TR6x4ST3, 17.2 m, 38 t, winter

Figure 50. TR6x4LT2ST3, 24.4 m, 55 t, winter
Figure 51. TK6x4DY2ST3, 24.3 m, 57 t, winter

Figure 52. TK6x2DY2LT2ST3, 31.3 m, 74 t, winter
### Appendix C. Questionnaire after driving each vehicle

1) Hur lätt var det att hålla hela ekipaget rätt placerad och på rätt kurs på det kurviga vägavsnittet?

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<th>Mycket lätt</th>
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2) Hur upplevde du fordonskombinationens krängstabilitet på det kurviga vägavsnittet?

<table>
<thead>
<tr>
<th>Mycket stabilt</th>
<th>Inte alls stabilt</th>
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3) Upptäckte du att fordonskombinationens släp/trailers var stabil eller slängig vid omkörningar?

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<th>Mycket stabil</th>
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4) Hur var det att köra om de långsamma bilarna?

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<thead>
<tr>
<th>Väldigt lugnt</th>
<th>Mycket stressigt</th>
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</table>

5) Upptäckte du att fordonskombinationens släp/trailers var stabil eller slängig vid körning mellan konerna?

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<thead>
<tr>
<th>Mycket stabilt</th>
<th>Mycket slängig</th>
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6) Sammanfattat, hur lätt var det att kontrollera/styra fordonskombinationen i de olika körsituationerna?

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<tr>
<th>Mycket lätt</th>
<th>Mycket svårt</th>
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Eventuell kommentar:

__________________________________________________________________________________

__________________________________________________________________________________

__________________________________________________________________________________

(SVARA BARA OM DET HAR VARIT DIN ANDRA KÖRNING) Sammanfattningsvis, hur var det att kontrollera/styra den här fordonskombinationen jämfört med den du körde först?

<table>
<thead>
<tr>
<th>Mycket lättare</th>
<th>Lika</th>
<th>Mycket svårare</th>
</tr>
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<tbody>
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</tbody>
</table>
Appendix D. Questionnaire on realism of the driving experience

1. Hur realistisk tycker du körningen i simulatorn var totalt sett?
   
   1 2 3 4 5 6 7
   Inte alls realistisk ☐ ☐ ☐ ☐ ☐ ☐ ☐ Mycket realistisk

2. Hur realistiska upplevde du situationen då du fick möte vid omkörningen?
   
   1 2 3 4 5 6 7
   Inte alls realistisk ☐ ☐ ☐ ☐ ☐ ☐ ☐ Mycket realistiska

   Vad var det som var realistiskt eller orealistiskt? _______________________________________
                                           _______________________________________
                                           _______________________________________

3. Hur realistiskt upplevde du:

<table>
<thead>
<tr>
<th>Inte alls realistisk</th>
<th>Mycket realistisk</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 2 3 4 5 6 7</td>
<td></td>
</tr>
</tbody>
</table>

   Vägmiljön?
   Omgivning (t ex diken, terräng, hus)?
   Inbromsningar?
   Accelerationer?
   Ljudet (från vindbrus, motor, däck)?
   Lastbilens fjädringsrörelse (vertikalt och krängning)?
   Vibrationer i lastbilshytten?
   Styrkänsla och motstånd i ratten?
   Manövrerbarhet och köregenskaper vid styrning?
   Hastighetsupplylevelsen?

   Eventuell kommentar och förbättringsbehov: _______________________________________
                                           _______________________________________
                                           _______________________________________

4. Kände du dig åksjuk under körningen?

   1 2 3 4 5 6 7
   Inte alls åksjuk ☐ ☐ ☐ ☐ ☐ ☐ ☐ Mycket åksjuk

The Swedish National Road and Transport Research Institute (VTI), is an independent and internationally prominent research institute in the transport sector. Its principal task is to conduct research and development related to infrastructure, traffic and transport. The institute holds the quality management systems certificate ISO 9001 and the environmental management systems certificate ISO 14001. Some of its test methods are also certified by Swedac. VTI has about 200 employees and is located in Linköping (head office), Stockholm, Gothenburg, Borlänge and Lund.